REPORTS OF
THE 6TH INTERNATIONAL CONGRESS ON ACOUSTICS

III

D ELECTROACoustics
E ARCHITECTURAL ACOUSTICS

ICA

TOKYO, AUGUST 21-28, 1968
REPORTS
of
THE 6TH INTERNATIONAL
CONGRESS ON ACOUSTICS

Editor in Chief
Dr. Y. KOHASI
ACOUSTICAL SOCIETY OF JAPAN

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The Use of Digital Systems in Acoustical Measurements

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Introduction

The digital technology and system knowledge have in the past decade expanded to a level far beyond any estimate made just a few years ago. Still new areas are being taken up to evaluation for possible use of digital systems. One of the reasons for this is the hardware technology, that is the development within the electronic components with integrated circuit blocks, and another reason is the development of the software part, that is the computer programming part which controls the process and organizes the system. Sometimes it can even be hard to find a sharp limit between the hardware and software parts. They are both important to the complete system.

If now this technology has to be implemented on the science of acoustics, it will be worth mentioning some of the reasons which could make today's acoustical scientists look for digital systems. The use of acoustical analysis in respect to maintainability analysis of a given type of equipment will need automatic data logging and evaluation. The higher demand for accuracy during noise analysis (sonic booms) has demanded higher rate of speed, and the wish to minimize the influence of human errors due to the higher speeds will require automatic logging.

The major point which seems to ask for digitalization is therefore speed. Slower requirements will still be able to rely on paper recordings and human evaluation. Higher speeds will bring higher system costs, so a compromise between these two will normally be the result.
**System Approach**

The different requirements and the compromise between cost and speed therefore has demanded a system approach where flexibility is of major importance. The degree of speed will par example define the tape medium (paper or magnetic) or direct on-line connection to the computer. In B & K we have found that the system shown in figure 1, meets the above mentioned requirement of speed and flexibility.

The Parallel analyzer is of the analog frequency analyzer type, with 50 band-pass filters, which can be scanned successively by a digital multiplexer circuit up to 40-50 complete cycles per second.

The multiplexer output is the analog representation of the filter now being scanned. Therefore a counter is needed to give the channel number information to the computer. For some use also a time reference may be needed. The analog signal coming out of the multiplexer, is fed to the A/D-converter, which converts the signal to max. five decimal digits, presented in the BCD- or the 8421-code. Codechange is easily obtained by changing a matrix card.

On the input side of the Buffer and codeconverter is now 9 digits in parallel ready to be transmitted to the tape medium. The paper puncher and paper tape will in this example represent the tape medium and no big change will be necessary in changing to magnetic tape or on-line type of operation. The buffer will be necessary to hold the 9 digit information during the read-out sequence, which is sequential digit by digit. No change in the 9 digits is allowed during this period. The codeconverter will change the code to the proper code for the punch-out converter, which in turn will control the mechanical punch-out arms in the puncher.

The flexibility is obtained when one tries to consider, that some application may want to get the digital display only. The display unit can be connected right after the A/D-converter. The time-unit can be interchangeable with other units (range no. information etc.).

**Tape Organization**

Fig. 2 shows as an example how the block-format on the tape may be organized. One block represents max. 9 digits, which might be defined as:
The Use of Digital Systems in Acoustical Measurements

1. Range No. Shows the range of input selector
2. 1st Digit of Channel no.
3. 2nd Digit of Channel no.
   etc., as shown in Fig. 2

Using this kind of block definition, where every place in the block has the same meaning every
time, will easily make the tape useful for the computer. The code shown on figure 2 is the
standardized Flexowriter code which is useful on most computer read equipment. Space and car-
return signals are necessary only if the tape is to be read directly to printer or typewriter to make
a readable tabulation on the paper.

Software
Until now I have only talked about the hardware part of the system. But to use the computer to
analyse, or to evaluate, the results, it is also necessary to work on the software side of the
problem, that means to program the computer so it "knows" the meaning of the digits in the block
format. Therefore figure 1 also shows the connection of the typewriter and typewriter converter
to the tape puncher. In this way the program and the data tape can be fed to the computer using
part of the same equipment. Also comments in plain language can be printed on the tape. The
computer program will later select between data and plain language, and only use the data.

Synchronization
Because speed of data handling was one of the prime reasons which resulted in the figure 1 system,
a few comments on this aspect is worth mentioning. The A/D converter and the paper puncher is
normally working on different speed ranges. Therefore some kind of synchronization between the
units is needed. The A/D-converter cannot accept any change in its input before the digitizing
procedure is finished, and the puncher cannot accept any new 9 digit information from the buffer
before the mechanical puncher arms have finished their punching. Every unit will give a ready
signal when a process is ended. If now this signals are used as control signals this will make the
system follow the slowest unit. In this case the slowest unit is the paper puncher, (max. 100
punched characters per second). If this speed is to be exceeded the flexibility of the system
allows us to change the paper tape punch unit with the much faster magnetic tape units whereby
the speed can be accelerated to more than 100000 characters per second. And then if this speed still is not enough, on-line arrangements can be made directly to the computer.

Conclusion
We have seen the possibility to build up a digital system, which can be used to log and evaluate acoustical data in an automatic mode. The computer may be connected directly or may be fed with the tape at a later time. The future will probably show how special computers take over the entire system using Fast Fourier Transform arithmetics on samples taken directly at the acoustical waveform.
On the Measurement and Interpretation of Cross-Power-Spectra.

Jens Trampe Broch, Dipl. Ing. E.T.H.

When it is desired to find the relationship, if any, between data observed at one point in a physical system and data observed at another point in the system use can be made of the methods of correlation techniques. One such method is mathematically formulated in the so-called cross-correlation-function:

\[ \gamma_{xy}(\tau) = \lim_{T \to \infty} \frac{1}{T} \int_{0}^{T} f_x(t) f_y(t+\tau) \, dt \]  

(1)

where \( f_x(t) \) is the magnitude of the process measured at the point x at an arbitrary instant of time, \( t \), and \( f_y(t+\tau) \) is the magnitude of the process measured at the point y at a time \( \tau \) later.

In general the cross-correlation function turns out to show a fairly complicated dependency of \( \tau \) and to obtain meaningful data some sort of frequency analysis of the correlation relationship is necessary.

For a long time people involved in turbulence research have utilized correlation measurement techniques, and analog electronic correlators, which actually were nothing but electronic multipliers without time delay circuits, have been commercially available on the market for some time. It has been, however, an expressed desire to obtain time delay units for these multipliers so that the complete correlation could be measured, both as a function of time and space. Dependable time delay units are, on the other hand, not so easy to build and very few such units have been successfully constructed according to analog principles. There are ways out of this. One method, which is very elegant but rather expensive, is to utilize digital sampling principles.

Another method, which in the authors opinion looks at least equally promising at present, is the use of analog cross-spectral-density measurements with very narrow band frequency analyzers.

Mathematically the cross-spectral-density function is obtained by taking the Fourier transform of the cross-correlation function:
\[ w_{xy}(f) = \int_{-\infty}^{\infty} w_{xy}(\tau) e^{-j2\pi ft} d\tau \]  
(2)

where \( w_{xy}(f) \) is the (complex) cross-spectral density function.

From the theory of Fourier integrals it is known that when equation (2) is valid then \( \gamma_{xy}(\tau) \) can also be found by inversion:

\[ \gamma_{xy}(\tau) = \int w_{xy}(f) e^{j2\pi ft} df \]  
(3)

writing

\[ w_{xy}(f) = |w_{xy}(f)| e^{-j\phi_f} \]  
(4)

and considering the fact that \( \gamma_{xy}(\tau) \) is always a real quantity one obtains:

\[ \gamma_{xy}(\tau) = \int |w_{xy}(f)| e^{j(2\pi ft - \phi_f)} df = \int |w_{xy}(f)| \cos(2\pi ft - \phi_f) df \]  
(5)

Formula (5) can also be written:

\[ \gamma_{xy}(\tau) = \int |w_{xy}(f)| \cos(2\pi ft) df + \int |w_{xy}(f)| \sin(2\pi ft) df \]  
(6)

Equations (1) and (6) form the basis for analog cross-spectral density measurements. This will be clear from the following: An ideal analog frequency analyzer will allow only that part of the signal to be measured which has frequency components within a narrow frequency band, \( \Delta f \), see Fig. 1.

Assuming that no attenuation or amplification of these frequency components take place in the analyzer, and that both analyzer channels used have equal phase shifts then the cross-correlation between the two measurement channels is given by the expression:

\[ \gamma_{x_{af}y_{af}} = \frac{1}{2} \int |w_{xy}(f)| \cos(2\pi ft) df + \frac{1}{2} \int |w_{xy}(f)| \sin(2\pi ft) df \]  
(7)

When \( \Delta f = 0 \):

\[ \gamma_{x_{af}y_{af}} = 2 \int |w_{xy}(f)| \cos(2\pi ft) df + 2 \int |w_{xy}(f)| \sin(2\pi ft) df \]  
(8)

Setting \( \tau = 0 \) and utilizing equations (1) and (8) one obtains:

\[ 2 \int |w_{xy}(f)| \cos(2\pi ft) df = \lim_{T \to \infty} T \int f_{x_{af}}(t) \cdot f_{y_{af}}(t) dt \]  
(9)

Rearranging equation (9) and setting \( 2 |w_{xy}(f)| \cos(\phi_f) = C_{xy}(f) \) gives:

\[ C_{xy}(f) = \lim_{\Delta f \to 0} \lim_{T \to \infty} \int f_{x_{af}}(t) \cdot f_{y_{af}}^*(t) dt \]  
(10)

Furthermore, again utilizing equation (1) and (8), and setting \( \tau = \text{at} (90^\circ \text{phase shift}) \) gives:

\[ Q_{xy}(f) = \lim_{\Delta f \to 0} \lim_{T \to \infty} \int f_{x_{af}}(t) \cdot f_{y_{af}}^*(t) \Delta t \]  
(11)

where \( Q_{xy}(f) = 2 |w_{xy}(f)| \sin(\phi_f) \) and \( f_{y_{af}}^*(t) \) is equal to \( f_{x_{af}}(t) \) shifted \( 90^\circ \) in phase.

The reason for introducing \( 2 |w_{xy}(f)| \) instead of \( |w_{xy}(f)| \) is that in physically realizable systems only positive frequencies are involved, while \( w_{xy}(f) \) was introduced for mathematically convenience.
where both positive and negative frequencies where considered, see equation (3). $C_{XY}(f)$ is denoted as the co-spectral density function while $Q_{XY}(f)$ is the quadrature (quad) spectral density function.

Fig. 2 shows the principle of analog cross-spectral density measurements and it is seen that the main operations necessary are filtering, the shifting in phase of one of the filtered signals by $90^\circ$ and a multiplication of the two signals. The shifting in phase is possible in the new Brøel & Kjær Tracking Filter Type 2020 which has been designed with a view to also allow for cross-spectral density measurements. A typical measuring arrangement utilizing two Tracking Filters is shown in Fig. 3.

In the practical measurement of cross-spectral density functions certain important facts have to be taken into account which may not be quite obvious from the theoretical derivations. One of the most important of these is the analyzer bandwidth/signal correlation time relationship. The mathematical Fourier transform, equation (2), actually presupposes a continuous frequency analysis with infinitely narrow band filters ($e^{-j2\pi ft}$). Such analog filters are neither practical (because of the infinite analyzing time required) nor do they exist.

Commercially available analog filters have very definite bandwidths and the output of such a filter is therefore also self-correlated (auto-correlated) over very definite time intervals only. These time intervals may be regarded as the "memory" of the filter and it is clear that the multiplications required to obtain the cross spectral density function must be performed within the "memory time" of the filter. When the "memory" is not perfect the cross-spectral density function obtained will be in error.
How large the error will be depends, of course, upon the "memory" of the filters and the time delay between the two signals being multiplied. Theoretically the auto-correlation function for the output of a box-shaped narrow-band filter is a Sin(x)/x function as plotted in Fig. 4. In practice, however, no filter has the exact theoretical box-shape, and some experiments have been made to investigate the influence of filter bandwidths in practice. The results of these experiments will be presented orally at the conference.

From Fig. 4 the theoretical error in a cross-spectral density measurement can be read when the bandwidth, Δf, of the analyzing filters and the delay time, τ, between $S_{xx}(\omega)$ and $S_{yy}(\omega)$ are known. For instance, for the cross-spectral density measurement to be correct to within some 5% the following relation is obtained (Fig. 4):

$$E \cdot Δf \approx 0.2$$  \hspace{1cm} (12)

In addition to this the "normal" statistical error in power measurements on random functions

$$E = \frac{1}{\Delta f \cdot T}$$  \hspace{1cm} (13)

where T is the averaging time used in the multiplication (see equations (10) and (11)), should be considered. In general it can be said that the more accurate the measurements are to be the longer is the measurement time required.

For a given measurement time the two relationships (12) and (13) are conflicting in that a narrow bandwidth minimizes the delay-time error but increases the statistical error. In practice, therefore, both kind of errors must be considered and a suitable compromise chosen.

Further discussions and practical examples of the measurement and use of cross-spectral density functions will be presented orally at the conference and published in the Jøel & Kjær Technical Review.
A New Tool for
Real-Time Audio Spectrum Analysis

Heinz Blüsser

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During the last few years we have found an ever increasing demand for an automatic plotting spectrum analysis system for noise and vibration measurements. The most important elements of such a system are a set of parallel third-octave band filters spanning at least the center frequency range from 50 Hz to 10 KHz and a set of parallel level detectors, both rms and peak. The rms detectors should have two different integration time constants. The data readout should be both analog and digital: oscilloscope display, X-Y plot, numerical print out and digital magnetic tape recording. The system must be fully programmable for real-time data processing by a computer.

We have developed a new instrument, called "Real-Time Audio Spectrum Analyzer", which fulfills all these requirements. It utilizes 24 parallel channels, each 1/3 octave wide, to continuously monitor signals over any eight-octave range between 1.8 Hz and 18 KHz. Where coverage of virtually the entire range is required, an additional 12 channels can be added in a separate instrument. In either case, each channel is monitored once every 25 milliseconds, and the resulting spectrum is displayed on a CRT. An additional marker on the CRT indicates the displayed amplitude range: a selectable 40 dB "window" within the overall measurement range of 140 dB as shown in Fig. 1. A digital readout is also provided on the front panel for monitoring the output level of any channel.
Real-Time Audio Spectrum Analysis

The particular channel is selected by pushbutton. The readout is calibrated in dB above 1 microvolt independent of the range setting. Using a 1"-Condenser microphone with a nominal sensitivity of 5 mV/µbar, the readout is directly in dB or sound pressure level.

The 40-dB display can be in terms of either peak or rms values so that transients as well as longer-term phenomena can be evaluated. In addition, either of two time constants can be selected for rms measurements to match the character of the phenomenon being measured. Time constants of 0.1 and 1 second are standard. These agree with the requirements for "fast" and "slow" dynamic response as specified in IEC recommendation 179 for Precision Sound Level Meter; however, time constants up to 15 seconds can be provided when longer integration time is required.

The peak and rms detectors used in each of the channels must be extremely linear. For example, to measure signals with a crest factor of 5 over a dynamic range of 40 dB with an accuracy of ± 1 dB, one needs a ratio of at least 74 dB between the maximum peak voltage and the maximum drift or offset voltage, i.e. with a maximum output swing of 50 volts the offset voltage has to be smaller than 10 millivolts.

Fig. 2 shows a simplified block diagram of the rms detector. The high linearity is achieved by using a linear rectifier and overall feedback. Operational amplifier A1 and the associated circuitry work as linear half wave rectifier for output signals up to 50 volts. The following operational amplifier A2 takes the average of the rectified signal. Without the third amplifier A3 the circuit would be an average responding detector. But the third operational amplifier A3 inverts the DC signal and feeds it back to the summing point of operational
amplifier A1, i.e. the rectifier is biased by a voltage which changes its value proportional to the AC input signal. Thus the DC output signal is proportional to the rms value of the AC input signal. For tone burst signals with crest factors up to 5, accuracy is ±1 dB with respect to a steady sinewave having the same rms value. For narrow band noise the deviation is less than ±0.2 dB.

If the resistors are disconnected from the summing point of operational amplifier A2, the immediate information is stored in the capacitor. Then the DC output voltage changes its value by less than 100 microvolts per second; this corresponds to a change of less than 1 dB in 2 hours at full scale and less than 1 dB in 1 minute at -40 dB.

This storage capability plus analog and digital outputs make the analyzer compatible with a wide variety of recording and processing instrumentation. A spectrum can be held on the CRT for slow scanning by an internal scanner. The resulting analog output is suitable for recording on an X-Y recorder. Or the spectrum can be printed out as it is shown in Fig. 3. The whole spectrum, which corresponds to that of Fig. 1, was printed out in 1.2 seconds by a fast printer. The first two digits in each line of Fig. 3 indicate the scanned channel, the next four digits indicate the band level with a resolution of a tenth of a dB. With rapid scanning in the non-storage mode, the digital output permits accumulation of large amounts of data on high-speed digital recorders with a maximum
Real-Time Audio Spectrum Analysis

scanning rate of 1 millisecond per channel. All functions of the analyzer, such as selection of range, channel, and display mode, can be controlled remotely. So a truly automated spectrum analysis system with immediate data processing by a computer can be assembled without further modifications of the analyzer.

The rugged construction of the analyzer, its relative small size and low power consumption of approximately 100 W, made possible in part through the use of both linear and digital integrated circuits, permit its use in a wide variety of environments including those encountered in airborne and shipboard use.

All these features make the analyzer well suited for many applications. Among acoustical applications are aircraft noise analysis for determination of "Effective Perceived Noise Level" (EPNL), fast frequency response testing of sound systems using broadband noise, determination of airborne and impact sound protection in architectural acoustics, and analysis of noise generated by various machines such as automobiles, jet engines, lathes, etc. Underwater acoustical applications include studies of marine sound and sound propagation. The availability of filters down to 2 Hz center frequency make the analyzer especially suited for vibration measurements to analyze the effects of shock and vibration testing. Wave form analysis and characterization of earth tremors are two additional applications of the analyzer. Thus the analyzer closes the gap between usual spectrum analysis systems and very expensive digital data processing systems.

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<td>4</td>
<td>1</td>
<td>0</td>
<td>7</td>
</tr>
<tr>
<td>3</td>
<td>1</td>
<td>0</td>
<td>1</td>
</tr>
<tr>
<td>2</td>
<td>1</td>
<td>0</td>
<td>5</td>
</tr>
<tr>
<td>1</td>
<td>1</td>
<td>0</td>
<td>5</td>
</tr>
</tbody>
</table>

Fig. 3 Spectrum print out
Low Frequency Measurements using Capacitive Transducers

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The measurement of slowly varying air-pressures e.g. sonic booms, raises the problem of transducers and electrical systems which are stable and sensitive enough to convert the pressure variations into electrical signals which can be detected by indicating instruments or stored on tape for later analysis.

A typical shock-wave produced by a supersonic aircraft at altitudes above 20,000 feet has a nature at ground level as shown in Fig. 1. The peak pressure change $p$ may be as high as 2-5 lbs. foot (1000-2500 ubar) and several orders higher at low altitudes. The duration $t$ varies from 0.03 to 0.4 seconds. A Fourier-integral of the N-wave shows that to get a good transformation of the signal the system must have a linear frequency response in the range from 2-3 decades below the lowest and above the highest fundamental frequency $1/t$. That is $0.01$ Hz to $3$ kHz to cover the entire range.

---

**Fig. 1**
Idealized shock wave

**Fig. 2**
Fourier integral of N wave
Low Frequency Measurements using Capacitive Transducers

As transducer one of the best choices is a condenser microphone but as a standard microphone normally has an air equalization hole, which gives a lower limiting frequency of 1–3 Hz, something must be done to lower it. One way is to tighten the microphone absolutely, but to sensitive devices this will cause problems when it is exposed to very low pressure e.g., by air transport another possibility is to connect the volume behind the diaphragm to a larger volume with constant pressure. This will give an increase in sensitivity to the microphone around 10% at frequencies lower than that determined by the air equalization hole. A new type of microphone, Fig. 3, can be modified for low frequency operation by mounting a silicone-rubber ring which closes the equalization hole. The lower limiting frequency will then be in the order of 0.01 to 0.02 Hz.

![Rubber ring](Image)

**Fig. 3**

Low frequency microphone

To detect the slowly varying capacitance changes of the microphone ordinary preamplifiers must have a very high input impedance which is difficult to obtain and difficult to maintain under humid conditions. To make the microphone act as a low impedance, the capacitance of it can be measured at a frequency which is well above the frequency range of the microphone itself. This may be done in several ways. The microphone may be compared to a well known capacitor in a bridge circuit. The advantages of this is that the circuit can be made very stable and independent of the frequency of the generator, on the other hand it is difficult to detect the signals from such a bridge over a wide dynamic range.

A very often used solution is a circuit as shown in Fig. 4 where $C_1$ is the microphone and $C_2$ a rather large condenser, including cable. This circuit can be used in two ways, the impedance around the resonant points can be detected at a constant frequency, or the circuit can be used as a frequency determining part of an oscillator the output signal of which can then be detected in a F.M. detector. The main advantage of this system is that the microphone and a coil adaptor can be connected through a two-wire cable to detector and power supply. The disadvantage is that it is difficult to make the system sensitive as the quality factor of the circuit will be rather low.
Low Frequency Measurements using Capacitive Transducers

![Circuit Diagram](image)

**Fig. 4**
Series circuit with parallel capacitor

A parallel circuit can be used in the same ways as the series circuit but requires that signal-detection is close to the microphone. The main features of this system is that it is possible to make it very sensitive and get a large dynamic range and a good stability. A system using this principle was built and will be discussed more thoroughly.

It was decided that the system should fit direct to standard 1" condenser microphones with a capacitance ranging from 50-80 pF therefore the microphone adaptor should have a diameter of 15/16" and a length in the order of 3". To avoid transmission of high frequencies in cable, the adaptor should contain the oscillator and the detector, to be able to drive long cables it should also contain a preamplifier. Tuning of the circuit should be done by an external voltage.

The limitation in space forces the use of simple circuits and small components. A schematic drawing of the system is shown in Fig. 5.

![Schematic Drawing](image)

**Fig. 5**
Schematic drawing of system
The quality factor of the tuned circuit is determined by the phase shift, that gives usable linearity and the max. capacitance variation of the microphone which is about 6 pF for linear operation. L is about 2.5 H and capacitance of microphone and tuning capacitance diodes is 105 pF. Quality factor is in order of 15. The generator is a 10 Mc crystal controlled oscillator with a single transistor and it is driven to bottom to give a constant voltage, which is in order of 5 V_{rms}. The preamplifier is a monolithic circuit. The tuning voltage is applied through a diode to compensate for thermal capacitance variations of the capacitance diodes.

The adaptor containing these items is connected through a 5 wire cable to a power supply which also contains the rest of the system. The signal is amplified in an output amplifier and part of the output signal is applied to a Miller-integrator with three different time constants, the output from the integrator is used to tune the microphone circuit. The time constant is set to give the system a lower limiting frequency of 0.01-0.1 and 1 Hz. To obtain this with a reasonable condenser the biggest resistor should be 20 Mohm, and to avoid current off-set it is necessary to use a differential pair of field effect transistors in the input of the integrator.

In the fourth position the circuit can be tuned by a potentiometer to make the electrical system D.C. sensitive. It is possible to apply a 1 kHz signal to the tuning voltage and in this way produce a capacitance variation which can be detected at the output and in this way test the entire system. The meter indicates overload and balance in the D.C. mode.

<table>
<thead>
<tr>
<th>Data obtained</th>
<th>System Alone</th>
<th>System with microphone</th>
</tr>
</thead>
<tbody>
<tr>
<td>Capacitance range</td>
<td>50 - 80 pF</td>
<td>0.01 Hz - 18 kHz</td>
</tr>
<tr>
<td>Frequency range</td>
<td>DC - 200 kHz</td>
<td>0.8 μbar peak</td>
</tr>
<tr>
<td>Linear range</td>
<td>±6 pF</td>
<td>±2000 μbar</td>
</tr>
<tr>
<td>Noise: 0.01 Hz - 200 kHz</td>
<td>0.2 μbar peak</td>
<td>0.15 μbar peak</td>
</tr>
<tr>
<td>2 Hz - 200 kHz</td>
<td>0.15 μbar peak</td>
<td>0.4 pF/day, 0.015 pF/°C</td>
</tr>
<tr>
<td>2 Hz - 20 kHz</td>
<td>30 μbar/day, 1 μbar/°C</td>
<td>Output of adaptor</td>
</tr>
<tr>
<td>Stability in DC-mode</td>
<td>±12 V ±10 mA</td>
<td>0.15 μbar peak</td>
</tr>
<tr>
<td>Output of amplifier</td>
<td>±12 V ±10 mA</td>
<td>0.4 pF/day, 0.015 pF/°C</td>
</tr>
<tr>
<td>Power</td>
<td>+15 V, 20 mA</td>
<td>-15 V, 22 mA</td>
</tr>
</tbody>
</table>

It is possible to cover other ranges than described. The microphone here is made for a polarization voltage of 200 volts. As this is not applied, the microphone may be made more sensitive and the quality factor of the tuned circuit may be higher in order to be able to detect lower levels. For detection of higher levels a less sensitive microphone may be used.
Reciprocity Calibration of Microphones in a Nitrogen-Filled Reverberation Chamber

Takayuki Nakajima
Electrotechnical Laboratory

Introduction

The author has studied a method for performing diffuse-field calibration\textsuperscript{*} of microphones based upon the reciprocity principle. Reciprocity calibration was carried out in a nitrogen-filled reverberation chamber statistically, during which many frequency samples and space samples were processed in a digital computer. Diffuse-field correction\textsuperscript{**} for MR 103\textsuperscript{***} (or equivalent) has been obtained in a nearly equal precision, compared to those in the pressure response and in the free-field correction.

Principle of Reciprocity Calibration in a Reverberation Chamber

Three microphones are mounted in pairs, and the pairs later interchanged, on devices which can revolve in a chamber. They operate as a source and a receptor, keeping an adequate distance for negligible direct-sound transmission between the two. In the two-port network, which consists of Source—Acoustic Medium—Receptor, the transfer admittance between a source and a receptor, that is Reciprocity parameter\textsuperscript{(1)} of the field, is determined by medium constants, chamber volume and reverberation time at a given frequency.

\textsuperscript{*} The diffuse-field response $M_f$ at a given single frequency is:

$$M_f = \left[ \frac{1}{4\pi} \int_0^{2\pi} \int_0^\pi M_f^2(\theta, \phi) \sin\theta \, d\theta \, d\phi \right]^{\frac{1}{2}}$$

where $M_f(\theta, \phi)$ is the free-field response due to sound incident from the direction $(\theta, \phi)$.

\textsuperscript{**} The diffuse-field correction of a microphone at a given frequency is the diffuse-field response level minus the pressure response level.

\textsuperscript{***} Microphone supporting device is a long cylinder whose diameter is equal to that of MR 103.
Reciprocity Calibration of Microphones in a Nitrogen-Filled Reverberation Chamber

By measuring the ratio of the driving voltage to the open-circuit voltage of the above network with the pair shown in Meas. 3 of Table 1, an equation containing two unknowns in the form of the product: diffuse-field response of source B times diffuse-field response of receptor C, is derived. Another equation containing the ratio of the same unknowns is led from measured voltage ratios in Meas. 1 and Meas. 2. The combination of the two equations leads to the diffuse-field response, \( \langle M_d \rangle \), as follows:

\[
\langle M_d \rangle = \frac{1}{\omega_c} \left( \frac{V_e}{\langle R_0 \rangle \langle R_f \rangle} \right)^{\frac{1}{2}}, \quad \langle J_d \rangle = \frac{6}{\pi \log_{10} 10} \frac{1}{\rho f} \frac{1}{c} \langle T \rangle^{\frac{1}{2}}
\]

where \( \omega_c \); capacitance of transducer, \( \rho \); density of medium, \( c \); velocity of sound, \( T \); reverberation time, \( V_e \); effective chamber volume, \( f, \omega \); frequency, \( \omega = 2\pi f \), \( \langle \ldots \rangle \); showing the values determined by statistical means, \( J_d \); reciprocity parameter.

Facilities and Equipments

A new reverberation chamber in which the medium is replaced by nitrogen was designed to carry out the experiment above 4000 Hz. Sound absorption is low and stable in purified dry nitrogen compared with that in air. In the chamber, there is a facility which causes the source and receptor microphones to move around spirally and an

Table 1 Pairing of Transducers, A, B and C, in Reciprocity Calibration.

<table>
<thead>
<tr>
<th>Source</th>
<th>Receptor</th>
<th>Voltage Ratio</th>
</tr>
</thead>
<tbody>
<tr>
<td>Meas. 1</td>
<td>A</td>
<td>B</td>
</tr>
<tr>
<td>Meas. 2</td>
<td>A</td>
<td>C</td>
</tr>
<tr>
<td>Meas. 3</td>
<td>B</td>
<td>C</td>
</tr>
</tbody>
</table>

*Pitch of spiral rotation, 45 mm. Speed is 0.25-4 rpm, variable. The interval between source and receptor, 95 cm, and the distance from the nearest walls, 60 mm.
adequate number of independent space sample points in the field can be obtained.

**Measurements**

The voltage ratio measurement is made in two operations. In the first operation, the transfer switch is thrown to the upper position shown in Fig. 1. This energizes the source by impressing the voltage in pure tone form across its terminals. The random amplitude voltages which appear at the terminals of the receptor as a function of time are rectified and recorded on punched paper tape after they have been converted to digital code by digital-voltmeter. The sampling speed is 4.2 times per second, and 800 independent pressure samples are obtained. In the second operation, the transfer switch is thrown to the lower position, impressing a voltage across the insert resistor. The indicator reading, $V_0$, of the digital-voltmeter produced by the known attenuation, $A$, is recorded. From the recorded samples: $s_n$, $n=1,2,...,N$, the voltage ratio, $<R>$, is

$$<R> = A + \left\{10 \log_{10} \left( \frac{\sum_{n=1}^{N} s_n^2}{N} \right)^{\frac{1}{2}} - 10 \log_{10} V_0 \right\}$$

In order to measure the reverberation time, about 1000 points are sampled and recorded on punched tape from frequency response curve. The frequency range is $+100-150$ Hz. From statistical analysis on the relations between reverberation time and frequency response in reverberation room, it is derived that the average occurrence times per unit frequency of crossing of a given level in frequency response is in proportion to reverberation time at the frequency, and that the distribution of occurrence times of crossing of many levels follows the Rayleigh distribution. (Ref. Fig. 2)

From the above mentioned space samples and frequency samples and other input data—medium constants, measured values in coupler calibration, etc.—, a computer program provides for the printout of diffuse-field correction for laboratory standard microphones. The results are given in Table 2, and they are plotted in Fig. 3. The diffuse-field correction for MR 103 obtained from free-field correction and directivity patterns is shown in the same figure.

* The independent space samples are obtained from every point which is more than a half-wavelength apart.
Reciprocity Calibration of Microphones in a Nitrogen-Filled Reverberation Chamber

<table>
<thead>
<tr>
<th>kHz</th>
<th>Correc. (dB)</th>
<th>kHz</th>
<th>Correc. (dB)</th>
</tr>
</thead>
<tbody>
<tr>
<td>4.00</td>
<td>1.4</td>
<td>7.00</td>
<td>3.6</td>
</tr>
<tr>
<td>4.25</td>
<td>1.6</td>
<td>7.50</td>
<td>3.9</td>
</tr>
<tr>
<td>4.50</td>
<td>1.7</td>
<td>8.00</td>
<td>4.1</td>
</tr>
<tr>
<td>4.75</td>
<td>1.9</td>
<td>8.50</td>
<td>4.2</td>
</tr>
<tr>
<td>5.00</td>
<td>2.1</td>
<td>9.00</td>
<td>4.1</td>
</tr>
<tr>
<td>5.25</td>
<td>2.3</td>
<td>9.50</td>
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</tr>
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<td>2.9</td>
</tr>
<tr>
<td>6.00</td>
<td>2.9</td>
<td>12.00</td>
<td>2.3</td>
</tr>
<tr>
<td>6.50</td>
<td>3.3</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Table 2 Diffuse-Field Correction for MR 103 (or Equivalent).

The two corrections obtained from the two methods are in very good agreement with each other. The precision of resulting values which is represented with triple median-error, 3 \( \mu \), (96 percent probability) is expressed by standard deviations, \( \delta \), of terms shown in suffixes, as follows:

\[
3y_d = 2D24 \sqrt{3 \left( \frac{\sigma_{(R)}^2}{2} + \frac{\sigma_{(T)}^2}{4} + \frac{\sigma_A^2}{2} + \frac{\sigma_P^2}{4} \right) + \frac{\sigma_{(R)}^2}{2} + \frac{\sigma_{(T)}^2}{4} + \frac{\sigma_A^2}{2} + \frac{\sigma_P^2}{4}}
\]

where \( N \): independent space sample points, \( B \): polarizing voltage, \( A \): standard attenuator, \( P \): pressure response

The precision of the values given in Table 2 is estimated about \( \pm 0.3 \text{ dB} \) up to 8 kHz, \( \pm 0.5 \text{ dB} \) above 9 kHz from the estimated values of each term of error:

\( \sigma_{(R)} \approx 0.19 \text{ dB from } N=800; \sigma_{(T)} \approx 0.5 \text{ dB, and the rest, } 0.05 \text{ dB up to } 8 \text{ kHz}; \)

\( \sigma_{(R)} \approx 0.8 \text{ dB, and the rest, } 0.1 \text{ dB above } 9 \text{ kHz.} \)

Conclusion

The diffuse-field calibration of laboratory standard microphones, such as the MR 103, was studied experimentally in a nitrogen-filled reverberation chamber which was specially designed for the statistical method of measurements. The resulting diffuse-field correction for MR 103 (or equivalent) is presented in both tabular and graphic form.

Acknowledgement

The author is grateful to Dr. Eiichi Matsui and Mr. Syozo Hattori of the Acoustics Section, Electrotechnical Laboratory, for useful suggestions and support in the work.

Comparison Calibration of Microphones in a Reverberation Room

HATTORI Shozo and MINAGAWA Moriyasu
Electrotechnical Laboratory

INTRODUCTION
The available data should be examined statistically to calibrate acoustical instruments in a reverberation room, in which the sound field properties have functions of an ergodic ensemble. The properties in a room have been analyzed statistically, then, on the basis of this analysis, the microphone comparison calibration method has been developed.

STATISTICAL ANALYSIS
A brief outline of statistical analysis is as follows.

(1) When a sound source is excited at a single frequency, sound pressure \( p \) at a receiving point follows the Rayleigh-distribution and \( p^2 \) is distributed exponentially.

In case of a multi-tone excitation, of which frequency components are all independent and \( n \) in number, distribution \( \bar{p}^2 \) follows \( \chi^2 \) distribution with \( 2n \) degrees of freedom, where \( \bar{p}^2 \) means the average of \( p^2 \) over a relatively long time so as to cancel out the beat effect. Setting \( Z = 20 \log (\bar{p}^2 / \langle \bar{p}^2 \rangle) \), where the bracket \( \langle \rangle \) denotes an ensemble average, and transforming the distribution of \( \chi^2 \) to the distribution of \( Z \), the probability function of \( Z \) is

\[
W(Z) = 0.23 \cdot \frac{n^\alpha}{\Gamma(n)} \cdot 0.07 \cdot \exp(-n \cdot 0.05),
\]

and the standard deviation of \( Z \), i.e. S.D. of SPL is

\[
\sigma_Z = 4.3 \sqrt{\psi'(n)}, \quad \text{(dB)}
\]

where \( \psi'(n) \) is polygamma function.
(2) In case of a warble-tone sound source, we may regard the warble tone as a multi-tone whose frequency components are nearly $2\alpha f / \alpha$ in number, where $\alpha f$ is the modulation swing and $\alpha$ is the modulation frequency. This is because amplitudes of components are all nearly equal, for modulation indices of 10 or higher.

(3) In the reverberation room, however, the frequency components of the multi-tone are correlated with each other, and the normalized frequency-correlation of the squared modulus of a frequency response in a room is shown as

$$\rho(\delta) = \frac{1}{1 + (2\pi \zeta \delta)^2} \quad (3)$$

where $\zeta = T/13.8$, $T$ = reverberation time.
Comparison Calibration of Microphones in a Reverberation Room

Consequently, the statistical independence of each of the components diminishes and S.D. of SPL increases.

(4) The nomogram for determining S.D. of SPL is drawn up for the above frequency-correlations, and is shown in Fig. 1. When frequency intervals of the multi-tone are closed in order of \( \Delta f \lesssim 3.3 \), S.D. of SPL depends only upon the frequency width \( 2\Delta f \), to which the random band noise excitation is applied. There is good agreement between the theory and the experiments as shown in Fig. 2, in which S.D. for the experimental results is computed by samples of 10.

METHOD OF COMPARISON CALIBRATION OF MICROPHONES

Comparison calibration of microphones were carried out in the Electrotechnical Laboratory reverberation room which is 210 m\(^3\) in volume and has non-parallel walls.

One of the MR-103 condenser microphones were used as a standard microphone whose diffuse-field reponse was measured by the reciprocity method in the nitrogen-filled reverberation chamber.

Figure 3 gives an arrangement for the calibration system, and Table 1 lists the characteristics of the warble-tone used. Since the signal of the saw-tooth oscillator was used for a external modulation signal, the modulation frequency and the warble band could be changed easily and arbitrary. In order to average the output of the microphones over a relatively long time, a level meter having a time constant of 30 sec. was

---

Table 1

<table>
<thead>
<tr>
<th>Frequency (kHz)</th>
<th>Modulation Band (Hz)</th>
<th>S.D. (dB)</th>
</tr>
</thead>
<tbody>
<tr>
<td>8</td>
<td>2 kHz</td>
<td>±200 Hz</td>
</tr>
<tr>
<td>4</td>
<td>1.5</td>
<td>±200 Hz</td>
</tr>
<tr>
<td>2</td>
<td>1.0</td>
<td>±100 Hz</td>
</tr>
<tr>
<td>1</td>
<td>0.5</td>
<td>±50 Hz</td>
</tr>
<tr>
<td>0.5</td>
<td>0.2</td>
<td>±25 Hz</td>
</tr>
</tbody>
</table>

Fig. 3 Arrangement for comparison calibration of microphones.
Comparison Calibration of Microphones in a Reverberation Room

used. The standard and the unknown microphones were set up on the turntable as shown in Fig. 3, and the positions of these microphones and the sound source were chosen over $\lambda$ apart from the walls.

Five MR-103 microphones were used as unknown microphones and calibrated in order to confirm the accuracy of this method. Following $X$ is regarded as the error of this method, because the following $R_{ps}$ and $R_{ps}$ have been measured by the absolute pressure calibration within an accuracy of 0.1 dB.

$$X = (R_{dx} - R_{ds}) - (R_{px} - R_{ps}) \quad \text{(dB)}$$

where $R_{dx}$ and $R_{ds}$ are diffuse-field response levels of the unknown and the standard microphone, respectively, and $R_{px}$ and $R_{ps}$ are pressure response levels of the unknown and the standard microphone, respectively. The positions of the microphones were moved and the calibrations were carried out on five sampling points per one unknown microphone.

The resulting $X$ and S.D. of $X$ are shown in Fig. 4 and Fig. 5.

CONCLUSION

Standard deviation of SPL in a reverberation room is derived theoretically from the values of the warble band, the modulation frequency and reverberation time by using nomogram, and the accuracy of the comparison calibration is esitimated. When further accuracy is required, an average over many sampling points of the calibration may be taken. For instance, by averaging five sample-values, the accuracy of this method can be held within $\pm 0.5$ dB.

The Free Field and Pressure calibration of Condenser Microphones using Electrostatic Actuator.

Gunnar Rasmussen  Electronic Engineer
A/S Brüel & Kjaer, Denmark

Free field reciprocity calibration of condenser microphones is a very time consuming and demanding task. It is therefore common practice to make pressure calibrations and add corrections for the diffraction thereby obtaining the free field response of the microphone.

An important factor and one often overlooked in this procedure is that the microphone diaphragm when used in a free field will look into another acoustical impedance than the impedance of the coupler volume used for the closed chamber reciprocity calibration technique. If therefore a free field calibration of a microphone and a closed coupler calibration of the same microphone is carried out and the pressure response $\frac{e_p}{\rho}$ is subtracted from the free field calibration $\frac{e_p}{\rho_f}$, this correction $\frac{e_p}{\rho} - \frac{e_p}{\rho_f}$ can only be transferred to another microphone if the impedance of this microphone diaphragm is the same as the impedance of the microphone used for the original calibration. It is shown in this paper that this problem is only significant around and above the diaphragm resonance. Unfortunately, this is also the range where the closed coupler reciprocity calibration has severe limitations, and calibration in this range is by some people considered more or less an art. The electrostatic actuator offers great advantages for frequency response calibration under those circumstances.

The electrostatic actuator makes it possible to apply a fictive pressure to the diaphragm. This may be done while the diaphragm is looking into its normal free field radiation impedance. It is also possible to use the electrostatic actuator in a closed coupler volume, thereby getting the difference between the closed coupler volume pressure response $\frac{e_p}{\rho}$ and the pressure response under free field load conditions $\frac{e_p}{\rho}$.

It was the purpose of this work to investigate the correlation between the two methods under closed coupler conditions, to check possible influence of the electrostatic actuator on the calibration.

*) Discussed in a paper presented to WG 13 T.C. No. 29 under I.E.C. by Professor F. Ingerslev.

— D - 25 —
to present typical data for some microphones and give a typical curve, which may be used either way deducting the closed coupler response $E_{00}$ from the actuator response $E_{2020}$ or vice versa.

The force $F$ applied by the electrostatic actuator is inversely proportional to the square of the distance between the diaphragm and the actuator $d$ and proportional to the polarization voltage $E_0$ and the variation imposed on the polarization voltage $e$. $F = \frac{E_0^2 \cdot e}{d^2} \cdot K$

It is important that the actuator is acoustical transparent so the diaphragm is loaded in the same way as without actuator. This is most easily obtained for microphones with no cavity surrounding the diaphragm. The actuator should cause no change in the microphone response. Typical values of $E_0$ and $d$ are 800 V and 0.5 mm. The force attracting the diaphragm to the back plate inside the microphone is 200 V over 0.02 mm. That means less than 0.65% change in diaphragm position due to the actuator.

This may be counteracted by raising the polarizing voltage of 200 V to 201.28 V and referring the calibration to 200 V polarizing voltage.

The pressure response was measured under different load conditions by the set-up shown in Fig. 1. The load on the microphone diaphragm is changed by plotting the frequency response while radiating into a free field and then placing a 1/4 wave length tube over the microphone with actuator as shown in Fig. 2. For a 1/4 wave length between the diaphragm and the rigid end surface of the tube will the pressure at the diaphragm approach zero hence presenting a very low radiation impedance to the diaphragm, which is allowed to swing freely with maximum volume displacement.

The frequencies used was accurately adjusted using the electronic counter and the 1/4 wave length tube tuned to proper length by a micrometer moving a piston as the end surface of the tube.
The output signal was filtered and could be read on an expanded precision meter scale. There is good agreement between the closed coupler reciprocity method and the 1/4 wave length actuator tube as also reported earlier privately by Mr. K. Rasmussen, The Acoustical Laboratory of The Technical Highscool in Copenhagen. See Fig. 3. It is worth noting that the calibration in closed couplers is very tedious and time consuming if carried out at higher frequencies due to wave motions in the coupler volumes.

![Frequency response curve](image)

**Fig. 3**

The impedance $Z_r$ of Fig. 4 through which the force creating the diaphragm motions looks into the microphone may be deduced from the curves, see Fig. 3. By assuming that it is mainly a mass reactance $m_r$ and then calculate $m_r$ from the change in resonance frequency of the diaphragm we get for $f_0 = \frac{1}{2\pi \sqrt{m_r k_0}}$ the change in $m_a$ in % to be $(\frac{f_0 - f_{02}}{f_{02}})$, $m_a \approx 56 \text{ kg/m}^2$ where $f_0$ is the diaphragm resonance looking into essential zero rayls $f_{02}$ the resonance for 42 rayls load and $m_a$ a typical value for the diaphragm mass of a B & K 4132 Microphone.

![Electromechanical equivalent circuit](image)

**Fig. 4**
This may also be calculated from the impedance in a plane progressive wave $Z_s$ which for a shallow column of air of depth is $Z_s = -iQ_c \cot \varphi d$ where $Q_c$ is 42 rayls and $\omega$ angular frequency, $c$ the speed of sound in air. For small values of $\varphi d$ we have

$$Z_s = -iQ_c \frac{\varphi d}{c} + i\omega \frac{d^2}{3}$$

Here is the mass reactance $Z_r = i\omega \frac{d^2}{3}$ and the actual mass $m_r$ added to the diaphragm under free field conditions is $m_r = \frac{Z_r}{i\omega} = \frac{d^2}{3} = \frac{8.18 \cdot 0.27 \cdot 1.29 \cdot 10^3 \cdot 10^6}{3 \pi} \sim 55.6 \, m\text{g/m}^2$

for a diaphragm of 18 mm in diameter and a reduction factor of $e^{-27}$ for the diaphragm movement.

The compliant reactance $C_r$ is large compared to $C_a$ that means the stiffness $\frac{C}{L}$ of the air is small compared to that of the diaphragm.

$$C_r \frac{1}{i\omega Z_r} = \frac{d^2}{3} \cdot \frac{8.18 \cdot 0.27}{3 \pi \cdot 1.29 \cdot 10^3 \cdot 10^6 \cdot 3.40^2} \sim 0.29 \cdot 10^{-6} \, m \, m^2 / 0.29 \, \mu F$$

The mass reactance cannot be ignored and the order of magnitude is large enough to require consideration in transferring diffraction corrections from one microphone to another. One may hope that future standardization efforts will take also the unique calibration possibilities offered by the electrostatic actuator into consideration. The typical correction curve obtained for B & K 4132 and 4131 is shown in Fig. 5.

![Correction curve](image)

It is obvious that no problem exist in the use of a microphone when the correction for the acoustical load is known. Under normal circumstances would the pressure response for the microphone looking into a free space be most valuable allowing correction for free field diffraction to be added with no correction for the mechanical impedance of the microphone needed.

With the miniature condenser microphones available to-day it seems reasonable to choose a small microphone for the measurement of sound pressure levels in confined spaces at high frequencies, where the only applications may be found, in which the pressure response under zero impedance load conditions seems of any practical relevance.
DÉTERMINATION THÉORIQUE DE LA RÉPONSE D'UN ÉCOUTEUR DYNAMIQUE

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I° INTRODUCTION

La connaissance de la courbe de réponse des écouteurs en fonction de la fréquence est souvent primordiale dans bien des recherches et sa détermination précise est parfois délicate. Il est connu que l'emploi de coupleurs permet l'obtention de cette caractéristique avec une précision et une fidélité très acceptables. Cependant, on sait aussi que la forme et le volume de ces coupleurs ont une influence primordiale sur l'allure des courbes obtenues. Pour cette raison, depuis bien des années, divers auteurs ont essayé de réaliser des coupleurs particuliers possédant des caractéristiques acoustiques aussi voisines que possible de celles de l'oreille humaine. Dans le cas où l'écouteur considéré est du type électroélectrique, il est possible de déterminer approximativement sa courbe de réponse à partir de considérations théoriques faisant intervenir certaines analogies électro-mécano-acoustiques ce qui peut être très utile pour le constructeur.

II° RAPPEL DE NOTIONS THÉORIQUES CLASSIQUES

Lorsque le conducteur mobile de l'écouteur se déplace à une vitesse $v$ dans l'entrefer d'un aimant où l'induction est $B$, il y a création d'une force contre-électromotrice $e$ :

$$ e = B l v $$

si $l$ désigne la longueur du conducteur mobile.

En désignant par $Z_e$ l'impédance électrique de ce conducteur lorsqu'il est immobile, la loi d'OHM généralisée permet d'écrire :
DÉTERMINATION THÉORIQUE DE LA RÉPONSE D’UN ÉCOUTEUR DYNAMIQUE

\[ U = Z_e I + R I v \]  \hspace{2cm} (2)

en désignant par \( U \) la différence de potentiel à ses bornes et par \( I \) l’intensité du courant qui le traverse.

D’autre part, le conducteur mobile est soumis à une force \( F \) (loi de LAPLACE) :

\[ F = R I v \]  \hspace{2cm} (3)

ce qui permet de définir l’impédance mécanique du système :

\[ Z_m = \frac{F}{v} \]  \hspace{2cm} (4)

et les relations 3 et 4 permettent d’écrire :

\[ v = \frac{R}{Z_m} I \]  \hspace{2cm} (5)

et, en portant cette valeur dans la relation 2, on obtient :

\[ U = \left[ Z_e + \frac{R^2 I^2}{Z_m} \right] I = Z I \]  \hspace{2cm} (6)

où \( Z \) représente l’impédance totale du système. On voit que tout se passe comme si l’impédance \( Z_e \) était augmentée de la quantité :

\[ Z_m = \frac{R^2 I^2}{Z_m} \]  \hspace{2cm} (7)

appelée impédance motionnelle, \( Z_e \) étant formée d’une résistance \( r \) et d’une inductance \( L \) en série.

Il est possible de calculer \( Z_m \) à partir de la détermination de \( Z_M \), ce qui peut se faire à l’aide d’une analogie du type admittance.

**III° LA PRESSION ACOUSTIQUE DANS LA CAVITÉ**

Lorsque l’écouteur est placé sur un coupleur de volume \( V \), il faut tenir compte de la capacité acoustique relative à ce coupleur :

\[ C_a = \frac{V}{\Delta C^2} \]  \hspace{2cm} (8)

et l’impédance mécanique de l’écouteur est alors associée avec l’impédance relative à cette capacité. Si le coupleur est cylindrique de base \( S \), on a :

\[ C_m = \frac{C_a}{S^2} \]  \hspace{2cm} (9)

où \( C_m \) est sa capacité mécanique à laquelle correspond l’impédance mécanique \( Z_c \) :

\[ Z_c = \frac{1}{\omega C_m} = \frac{s^2}{\omega C_a} \]  \hspace{2cm} (10)

en négligeant l’impédance mécanique relative au microphane à condensateur formant la base de la cavité.

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En définitive, on aboutit à deux systèmes couplés dont les impédances sont :

a) impédance électrique totale de l'écouteur :
   \[ Z = Z_e + \frac{R^2 1^2}{Z_m} \]

b) impédance mécanique du coupleur :
   \[ Z_c = \frac{S^2}{\omega C_a} \]

Les analogies électromécaniques permettent alors de déterminer la pression acoustique \( p \) développée dans le coupleur :

\[ p = \frac{B 1 S^2}{\omega C_a \left[ B^2 1^2 + Z_e Z_m \right]} \]  \( (11) \)

IV° CONCLUSIONS

En écrivant :

\[ Z_e = \sqrt{R^2 1^2 + \omega^2} \]

\[ Z_m = \sqrt{R_m^2 + \left[ R_m - \frac{k}{\omega} \right]^2} \]  \( (12) \)

on peut représenter les courbes de variation de \( Z_e \) et de \( Z_m \) en fonction de la fréquence \( f \) (figure 1), ce qui permet aussi d'avoir une idée de la courbe de variation du produit \( Z_e Z_m \) également en fonction de \( f \) (figure 2).

On peut ainsi découper trois zones appelées I, II, et III à l'intérieur desquelles le produit \( Z_e Z_m \) est :

dans I : inversement proportionnel à \( f \)
dans II : pratiquement indépendant de \( f \)
dans III : proportionnel à \( f^2 \).

Dans ces conditions, il est possible d'avoir une idée de la variation de \( p \) en fonction de \( f \) en utilisant la relation 11 :

zone I : le produit \( Z_e Z_m \) est supérieur au produit \( R^2 1^2 \) et \( p \) est pratiquement indépendante de \( f \).

zone II : le produit \( Z_e Z_m \) est inférieur au produit \( R^2 1^2 \) et \( p \) devient inversement proportionnelle à \( f \).

zone III : le produit \( Z_e Z_m \) est vite très supérieur au produit \( R^2 1^2 \) et \( p \) devient graduellement inversement proportionnelle à \( f^3 \).

Ces conclusions sont traduites par la figure 3 où la pression \( p \) est :
- constante dans la zone I,
- décroît théoriquement de 6 dB par octave dans la zone II,
- décroît théoriquement de 18 dB par octave dans la zone III.

Lorsque des fuites apparaissent entre l‘écouteur et le coupleur, la fréquence de résonance mécanique $f_m$ se produit à une fréquence plus élevée par suite de la diminution de la capacité mécanique du système et la zone I tend à s‘élargir ; la zone III commence donc à une fréquence nettement plus élevée et la décroissance dans les fréquences aiguës audibles est alors moins accentuée.

Les résultats obtenus sur deux écouteurs électrodynamiques classiques ont confirmé le bien fondé de cette théorie qui demanderait à être étudiée plus minutieusement.

**Figure 1 :** Forme générale des courbes de variation de $Z_e$ et de $Z_m$ en fonction de la fréquence $f$.

**Figure 2 :** Forme générale de la courbe de variation du produit $Z_eZ_m$ en fonction de la fréquence $f$.

**Figure 3 :** Forme générale de la courbe de variation de la pression $p$ développée dans un coupleur par un écouteur dynamique.
On the Transmission Characteristics of the 20 cc Coupler

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Introduction
The exact analysis of transmission characteristics of a 20 cc coupler which is widely used for pressure calibration of one-inch standard condenser microphones is reported in the paper. "The method of solving wave equations by infinite matrices" (abbr. the method of infinite matrices) used for the purpose has been worked out for exact analysis on free-field correction of a standard microphone mounted on a semi-infinite rod.¹ Some effects which are due to wave-motion of medium in the coupler and to diaphragm impedance of microphones are analyzed. Effects due to heat conduction of coupler walls and to capillary tubes are already known.²

Assumptions for the analysis are (1) that vibrations are symmetrically rotational, (2) that losses in the medium can be neglected, and (3) that the back plate of a microphone is flat and smaller than the diaphragm.

Analysis
In the method of infinite matrices, the coupler is divided into three cylinders and two diaphragms, as depicted in Fig. 1.

Next, impedance infinite matrices of each part must be derived. Using the method of separation of variables under the boundary condition that radial particle velocity at the circumferential wall of a cylinder is zero, a solution of the wave equation of the medium can be expanded in terms of normalized Dini series. Pressure and

Fig. 1 Schematic cross-section of a 20 cc coupler divided into three cylinders and two diaphragms.

\( V_i \): pressure infinite matrix at port \( i \),

\( I_i \): velocity infinite matrix at port \( i \),

\( V_0 \): infinite matrix representing electrostatic force on the source diaphragm.
velocity matrices are defined as one-column infinite matrices in which are arranged coefficients of the series on pressure and on axial particle velocity at an end surface of the cylinder, respectively. Then, the following relationships at ports 2 and 3 in Fig. 1, for instance, can be written as

\[
\begin{align*}
V_2 &= Z_D I_2 + Z_T I_3 \\
V_3 &= Z_T I_2 + Z_D I_3
\end{align*}
\] (1)

where \( Z_D \) and \( Z_T \) are open-circuit impedances of the cylinder, and they are represented by infinite matrices derived from the definitions of open-circuit impedances and the solution of the equation.

As a diaphragm of a microphone is fastened along a boundary circle, both displacement and pressure acted on it can be expanded in terms of normalized Fourier-Bessel series. Pressure and velocity matrices are also defined as one-column infinite matrices in which coefficients of the series are arranged. From a solution of the equation of motion, pressure and velocity matrices at port 8, for instance, can be written as

\[
V_8 = Z_M \cdot I_8
\] (2)

where \( Z_M \) is an infinite matrix representing the diaphragm impedance.

Then, conversion infinite matrices are sought which represent relationships between pressure or velocity matrices at both sides of surfaces dividing the coupler. In order to get relationships between pressures or axial particle velocities at adjoining end surfaces of cylinders of different radii, coefficients of both normalized Dini series are compared using their orthogonality. In the relation between velocity matrices, the boundary condition, that axial particle velocity at the flange of a larger cylinder, i.e. \( a_1 > \rho > a_2 \), is zero, is taken into consideration. Defining the resulting infinite matrix as \( D \) conversion matrix, the relationships at ports 3 and 4, for instance, are given by the formulas

\[
\begin{align*}
V_3 &= D \cdot V_4 \\
I_4 &= -\frac{a_2}{a_1} D^T \cdot I_3
\end{align*}
\] (3)

where \( D^T \) is transposition of the matrix \( D \). The relations, \( \frac{a_2^2}{a_1^2} D D^T = E \) and \( D D^T \neq D D^T \), show that \( D \) conversion matrix has no inversion uniquely, and that both pressure and velocity matrices are converted unidirectionally.

In order to get relationships between pressures or velocities at an end surface of a cylinder and on a diaphragm of a microphone, coefficients of both normalized Dini and normalized Fourier-Bessel series are also compared using their orthogonality. Defining the infinite matrix as \( F \) conversion matrix, the following relations at ports 7 and 8, for instance, are given as

\[
\begin{align*}
V_8 &= F \cdot V_7 \\
I_7 &= -\left( \frac{b}{a_2} \right)^2 F^T \cdot I_8
\end{align*}
\] (4)
On the Transmission Characteristics of the 20 cc Coupler

where $F^T$ is transposition of the matrix $F$. They have the same characteristics as a conversion matrix. The second equation satisfies the boundary condition that axial particle velocity at the flange of the cylinder, i.e. $a_2 > \varphi > b$, is zero.

Now, the transmission characteristics of a 20 cc coupler is defined as volume displacement of the receptor diaphragm using the coupler normalized by that using the ideal one, when unit electrostatic force acts on the source diaphragm. Volume displacement of the receptor diaphragm can be obtained by integration of displacement over its area facing on the back plate. Then, coefficients are rearranged in a one-row infinite matrix defined as the SUM matrix. Expanding the unit electrostatic force in terms of a normalized Fourier-Bessel series, a one-column infinite matrix, $\frac{1}{\pi b^2} \text{SUM}^T$, can also be obtained.

Utilizing the above results and applying network theorems, the transmission characteristics of a 20 cc coupler is

$$
\left( \frac{i \omega}{\pi b^2} \text{SUM} Z_{HR} F \left[ E + Z_{DP} \left( \frac{i \omega}{\pi b^2} \text{SUM} Z_{HR} F \right)^{-1} \right] Z_{T72} \right) \left( \frac{i \omega}{\pi b^2} \text{SUM} \right) \left[ E + \left( \frac{i \omega}{\pi b^2} \text{SUM} Z_{HR} F \right) \left( \frac{i \omega}{\pi b^2} \text{SUM} Z_{HR} F \right)^{-1} \right] \frac{1}{\pi b^2} \text{SUM}^T
$$

$$
\left( \frac{i \omega}{\pi b^2} \text{SUM} Z_{HR} F \left[ E + Z_{DP} \left( \frac{i \omega}{\pi b^2} \text{SUM} Z_{HR} F \right)^{-1} \right] Z_{T72} \right) \left( \frac{i \omega}{\pi b^2} \text{SUM} \right) \left[ E + \left( \frac{i \omega}{\pi b^2} \text{SUM} Z_{HR} F \right) \left( \frac{i \omega}{\pi b^2} \text{SUM} Z_{HR} F \right)^{-1} \right] \frac{1}{\pi b^2} \text{SUM}^T
$$

(5)
On the Transmission Characteristics of the 20 cc Coupler

where the subscripts S and R represent source and receptor, respectively. \( Z_{D2} \) and \( Z_{D7} \) are open-circuit driving-point impedances of the coupler at ports 2 and 7, respectively. \( Z_{T2} \) is open-circuit transfer impedance from the port 2 to the port 7. Expressions in brackets of the numerator show that the effects due to diaphragm impedance depends on the ratio of the driving-point impedance and the diaphragm impedance.

**Numerical results**

For numerical calculations, infinite matrices are truncated into finite ones, using the method of reduction confirming convergence by changing the order of matrices. It was sufficient for practical purpose that the order of all matrices was 10.

The wave-motion of the coupler is shown in Fig. 2. The abscissa is represented by the product of wave number \( k \) and radius \( a \) of the larger cylinder. Maximum value is only 0.13 dB at \( ka = 1.1 \). Below the frequency, wave-motion of air used in the coupler coincides with that of hydrogen, but there are slight differences in wave-motion caused by difference in diaphragm motion above the frequency. In the strict sense, it shows that wave-motion cannot be separated from the effect of diaphragm impedance.

Figure 3 shows diaphragm impedance corrections. At low frequencies, correction has positive value, because both driving-point and diaphragm impedances are stiffness reactance. When air is used, the curve shows that driving-point impedance changes from stiffness to mass reactance at about 2.2 kHz. When hydrogen is used, correction remains in positive value, because both impedances change from stiffness to mass reactance at about 8 kHz.

Theoretical values are compared with experimental ones, Fig. 4. For theoretical values, difference in the transmission characteristics between air and hydrogen was used. For experimental purposes, the difference in attenuation of electrical terminals between the source and the receptor microphones corrected by differences in effects due to heat conduction and in the ratio of the specific heats of two gases was taken up. At 2 kHz and below, both values agreed with each other. But there are some differences above that frequency. It seems that the differences are caused by degeneration of convergence for theoretical values.

**Conclusion**

It is concluded that wave-motion and effect due to diaphragm impedance of a 20 cc coupler have been precisely analyzed, and that numerical results have sufficiently agreed with experimental ones. Now, all corrections of the coupler are theoretically obtainable. It is not necessary to obtain wave-motion of the coupler experimentally. The method of infinite matrices is available for analysis on a field containing some discontinuous boundaries.

**Reference**

2) For example, American Standard Method for the Calibration of Microphones, Sl.10-1966.
The Free Field Calibration of a Sound Level Meter

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If a microphone is placed in a sound field the diffractions of the sound waves on the microphone produce an appreciable change in the resulting sound pressure acting on the microphone diaphragm. The ratio of the RMS output voltage from the microphone to the RMS sound pressure existing in the free field at the microphone location with the microphone removed from the sound field is called the Free-field correction. The Free-field correction depends on the orientation of the microphone with respect to the direction of propagation of the sound and on the external dimensions of the microphone (in particular those of the front and of fitted protective grids). The Free-field corrections of the microphone used in the following measurements are shown in fig. 1, 2 and 3.

With hand held sound level meters where the microphone is mounted in solid connection with the rest of the instrument, not only the microphone, but the entire apparatus, whose dimensions compared with those of the microphone are

Fig. 1

Fig. 2

Fig. 3
The Free Field Calibration of a Sound Level Meter

great, is placed in the sound field and, because of its size, will have some influence on the measuring results. In the following are shown some results of measurements carried out on sound level meters with different size and shape.

Description of the measuring set-up
The measuring set-up is illustrated in fig. 4. The measurements were carried out on wooden dummies on which the microphone could be placed. Each of the figures shows difference (in dB) of the RMS output voltage from the microphone when placed on the dummy and the corresponding voltage measured with the microphone alone, placed in the same position in the sound field and the same angle of incidence. The curves therefore show the influence of the sound level meter body. The measurements were carried out in an anechoic room (8 x 10 x 12 meters) with the inside covered by 1,5 meter rock-wool wedges. The wooden dummies used are shown in fig. 5 and are referred to in the following text by the numbers I, II, III, and IV.

The measurements were carried out at 0°, 30° and 90° angle of incidence. On account of the fact that none of the used sound level meter dummies have rotational symmetry the measurements at 30° and 90° of incidence were carried out with the dummy in 3 positions, 1 with the side, 2 with the front (side of the indicating instrument) and 3 with the back toward the sound source. The curves therefore also show the spread of the influence from the sound level meter body if it is turned 360° around its own axis.

For the measurements a 1" microphone, type 4131, was used, partly mounted with normal protecting grid and partly mounted with a correction grid (Random incidence Corrector, UA 0055) which gives the microphone a better omnidirectional sensitivity (fig.2). In addition a ½" microphone, type 4134, was used (fig.3).

Fig. 4 - Measuring set-up

Fig. 5. Size and shapes of the sound level meter (dimension in mm).
Fig. 6. Influence of the sound level meter body.

Results and discussion

From fig. 6 it is seen that the diffraction caused by the cone-shaped sound level meter is relatively small in view of dimensions but too great to be overlooked for free field measurements. By comparing the curves from the sound level meter sizes I, II and III it is seen that the cone-shape with the smallest angle naturally is the best one, but that the improvement does not justify the, in other respect, unpractical shape. A real improvement is obtained when the microphone is placed at a certain distance in front of the sound level meter. The spread that arises when the sound level meter is turned around its own axis will in this case also be minimum.
Fig. 7. Man's influence on the sound field.

A man's influence on the sound field by examination of the diffraction on hand held sound level meters it will be innatural not to examine the disturbance of the sound field that will be produced by a man holding the instrument in his hands. By comparing the curves fig. 7 it is easily seen that the disturbance produced by the observer is much greater than that produced by the sound level meter. It is seen that the disturbance are greatest in the range 300 - 700 Hz. In fig. 8 is shown the disturbance that an average man will produce in a free sound field measured at 400 Hz, 1600 Hz and 6300 Hz. It is seen that the distance between the microphone and the observer must be approx. 1 meter or more if a disturbance of ± 1 dB shall be maximum.

The shown set-up with the sound level meter with the observer behind, in a free sound field will hardly occur in practical measurements, but can be looked upon as a worst case. With measurements in a diffuse sound field or with measurements of noise with a certain bandwidth the influence from the observer will be small and the accuracy of measurements will therefore be better.
Impulse Noise Measurements

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It is the intention of this paper to show the importance of having high crest factor capability in instruments for impulse noise measurements. For measurements of alternating variables such as an acoustical signal the most useful quantity as a measure of the signal strength is the root mean square value. In some earlier articles (1) the squaring properties of such RMS instruments have been considered, but this paper will look more into the averaging process.

By measurements of stationary random or nonrandom signals, it is normally assumed that the averaging time is long compared to the longest period in the envelope of the signal. This assumption could be valid for repetitive pulses with reasonable repetition rate, but how should single impulses, or pulses with very low repetition rate be evaluated? Several authors (2) have worked on this problem and although there has been some discrepancies these works are now resulting in an IEC proposal for an Impulse Sound Level Meter (German standards for such an instrument already exists).

These standards prescribe that the mean square of the signal should be produced with an RC averaging time constant of 35 ms. The square root of the peak voltage on the averaging condenser should then be used as a measure of the impulse sound pressure in that the instrument in other respect should fulfill the Precision Sound Level Meter requirements.

These standards also prescribe some test signals of single tone bursts and repetitive tone bursts and the corresponding indications with tolerances whereby the averaging process can be checked.
Let us consider first the ideal circuit. The prescribed check signals consists of tone bursts of a 2 kHz sinusoidal signal. As the RC timeconstant is long compared to a half period of the signal, the sinewave can be replaced with a squarewave with same r.m.s. value without introducing essential differences. As the squaring circuit is symmetrical, the input can just as well be considered as a rectangular pulse of same duration as the tone burst. The output of the squaring circuit \( u = ke^2 \) will thus also be a rectangular pulse and the voltage \( v \) on the condenser will rise and fall exponentially with timeconstant \( RC \) according to the differential equation: \( ke^2 - v = iR = RC \frac{dv}{dt} \).

As the indication should be proportional to the square root of \( v \) peak, a fictitious instantaneous indication \( E = \sqrt{v} \) is introduced: \( e^2 - E^2 = RC \frac{dE^2}{dt} \).

This means that \( E^2 \) will rise and fall exponentially, while \( E \) will rise faster than the exponential and fall exponentially with timeconstant \( 2 \, RC \) as shown on fig. 2.

As the rising curve is of special interest to the following this curve is also shown in log-log scales on fig. 3.

In this ideal circuit the root extraction is done in the meter scale. Let us now consider a circuit where the root extraction is done in the squaring circuit itself. This is done by feeding back the voltage on the smoothing condenser to the squaring circuit, thereby changing the size of the parabola as shown on fig. 4 (see also (1)). Actually the parabola is approximated by a polygon, but as the errors are less than 5% and in most cases will average out to less than 1 or 2% the curve will in the following be considered as a true parabola. In fig. 5 is shown such a circuit.
Impulse Noise Measurements

For this circuit: \[ i = \frac{h}{V} e^2 - \frac{v}{R} = C \frac{dv}{dt} \]

Again a fictitious instantaneous indication \( E \) is introduced: \[ E = \sqrt{h \cdot \frac{R}{2}} \]

One thereby gets: \[ e^2 - \frac{e^2}{2} = \frac{RC}{2} \frac{de^2}{dt} \]

This is exactly the same differential equation as for the first circuit except for the factor \( \frac{1}{2} \) on the time constant, and the rise and fall curves will therefore be the same if the time constants are adapted each other.

However, this equality between the two circuits is correct only as long as the parabolic approximation of the squaring circuit is correct. This is most certainly not the case during the first part of the rise where the parabola is very small. In fig. 6 is shown the variable parabola in log-log scale as a set of straight lines with slope 2 and it is shown how at the high end these lines bend over to a common line with slope 1, parallel to the line for the instantaneous indication \( E \). If the "knee" is at \( F_e \) where \( F \) is the crest factor of the parabola, then for this part of the rise:

\[ i = \frac{h}{V} F_e^2 \frac{e}{F_e} - \frac{v}{R} = C \frac{dv}{dt} \]

which gives:

\[ F_e - E = RC \frac{de}{dt} \]

On fig. 7 is shown the resultant total rise curve in log-log scales. It starts as a straight line with slope 1 corresponding to an exponential rise of \( E \) with time constant \( RC \) against the value \( F_e \).

At a value \( \frac{t}{F_e} \) however the curve bends over in the ideal curve where \( e^2 \) rises exponential with the timeconstant \( \frac{1}{2} \) \( RC \) against the value \( e^2 \) only is the curve delayed \( \frac{1}{2} \) \( RC \).

It is seen that the rise curve is reasonably near to the ideal curve at \( t = \frac{RC}{2} \cdot \frac{4}{5} \). Thus to measure single square wave tone burst of duration \( t \) m.sec.:

\( F \geq \sqrt{\frac{140}{t}} \) For sine wave tone bursts:

\( F \geq \sqrt{\frac{260}{t}} \) and for practical impulse noise signals \( F \) should probably be even a factor
Impulse Noise Measurements

1.5 to 2 higher. For the shortest test tone burst of 5 m sec. \( P = 7.5 \).

Finally are shown some impulse noise measurements on practical noise sources made with a B & K impulse sound level meter, type 22c4.

The oscillograms show that pulse durations can easily be essentially shorter than 5 m sec. down to 0.2 m sec. and the tables show that the peak values can be more than 20 dB above the impulse sound level reading. This means that a parabola crest factor capability of 20 to 40 is needed to give correct evaluation of these signals.

The 2204 has a crest factor capability of 10 at full scale and inversely proportional to the deflection up to about 40 at \( \frac{1}{4} \) scale. The measurements also show that it is recommendable to check impulse sound level meters with tone bursts shorter than 5 m sec. probably down to 0.2 - 0.5 m sec.

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Low Cost Noise Meter with Impulsive Noise Measuring Device.

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Introduction

Presently available conventional sound level meters are quite expensive instrument. They are currently used by scientific institutions and public departments, but noise measurement and control has not become yet as popular as it should be, at least in our country, due to the high price or poor reliability of the instruments available at present in the market. Besides, impulsive noise are not properly evaluated by standard meters. Prof. Reichardt (1) considers that the integration time constant of a sound level meter should be the same as that of the human ear (25 milliseconds according to his conclusions). Our aim has been to develop an instrument accessible to everybody's budget, yet within the standards of precision noise meters and incorporating a device for adequate evaluation of impulsive noises.

Electronic Characteristics

The transducer employed is a locally made standard dynamic microphone. Its frequency response curve to a flat excitation signal, Fig. 1 Curve 1, is shown in Curve 2, of the same figure.

Due to the extremely wide range of signal levels that the instrument must handle, it is necessary to use a suitable attenuator. This attenuator has been divided in two synchronously controlled stage to keep the overall dynamic range of the system,
yet maintaining a low excursion for the operating point of the first amplifying stage, to minimize distortion. The attenuator has seven 10 Db steps, allowing operation with signals ranging from 50 to 120 Db.

The DBA weighting network has been designed without use of inductances to reduce cost and facilitate series production. The standard DBA curve has been approximated using two carefully selected crossover points for low frequencies and one for high frequencies. Feedback has been used in all stages to reduce distortion and improve linearity. The resulting curve has been easily kept within the limits stated for precision instruments.

The next amplifying stage compensates the non-uniform frequency response of the microphone as can be seen from Fig. 1, Curve 3.

R.M.S. readings are obtained by squaring the signal by means of a piecewise linear network which approximates very closely a second degree curve. After averaging the squared signal in a convenient RC network, it is fed in normal use to a conventional moving coil instrument whose scale in Db has been divided by two to perform the square root operation.
A built-in 1000 Hz oscillator is available for calibration purposes.

The SUM can handle without distortion signals the peak levels of which are up to 10 dB above their r.m.s. values.

A complete block diagram is shown in Fig. 2.

**Impulsive Noise Measuring Device**

To evaluate impulsive noises correctly it is necessary to design a special circuit capable of integrating the squared signal with a time constant of about 25 ms according to Reicherdt's experiences, which happens to be much shorter than that of typical moving coil meters. This objective was attained with a transistor in an emitter-follower connection, repeating the voltage developed across the adequate integrating network, and driving a storage capacitor through a low leakage diode. A field-effect transistor with very high input impedance "feels" the voltage across that capacitor charged to the highest value developed during the impulsive signal, and drives a conventional moving-coil instrument which shows the maximum integrating level.

If a very short duration pulse is fed to an ordinary noise level meter, the signal may be vanished even before the instrument starts deflecting and reading.
if any, will be inaccurate. In our device, the capacitor stores a charge, and the meter can take its time to read the correct value, even after the signal has elapsed.

All diodes have a very high resistance near their breakpoints. This fact introduces an uncertainty of some tenths of a volt in the voltage attained by the storage capacitor. Our circuit has been designed in such a way that this value represents a negligible portion of the total signal to be read.

Conclusions

We hope to have developed a compact, light and inexpensive apparatus whose retail price is under one hundred dollars and whose quality makes it suitable for most applications in noise evaluation. If desired, use of the microphone at some distance from the operator is also possible, to avoid influencing the readings his presence in the near-field.

Our experience with the impulsive noise measuring device suggests that big differences may exist between readings with the standard time constant and the 25 msec one, for many impulsive noises common in industry. This would lead to the conclusion that corrections of 5 Db for impulsiveness as now recommended, are not realistic.

References
One of the urgent problems facing the acoustics expert today is the evaluation of noise in industrial environments — where the noise is frequently of an impulsive nature: i.e., hammer blows, press strokes, typewriter clatter, etc. But before he can go about determining, for example, the average sound level which corresponds to the noise dose which a person has been subjected to during a workday, the expert must first be in a position to measure the sound adequately.

The use of sound level meters and common level recorders for this purpose has to be scrutinized carefully. IEC Recommendation 179 does not define the response of precision sound level meters for tone bursts lasting less than 200 ms. In the "fast" mode the response to a 200 ms tone burst at 1 KHz is prescribed to be 1 - 1 dB below the steady-state value. This statement is not sufficient to calculate in advance what will happen if the burst duration is reduced to below 200 ms. Even if this fairly loose specification were construed to indicate a nominal rms time constant of 127 ms, the performance could only be predicted down to a deviation of - 9 dB: because at this point the crest factor of the signal exceeds the tolerable limits and the amplifier may go into an overload condition, depending on whether the steady-state indication is at full scale or not. In practice, however, the performance at 200 ms is not controlled by an RC
time constant associated with the square-law detector but, rather, by the mechanical inertia of the meter movement or, in the case of level recorders, by the dynamic characteristics of the non-linear feedback loop which controls the movement of the stylus.

This effect has been described quantitatively in the literature (1)(2)(3). The lightly shaded area in Fig. 1 shows the multitude of values which, depending on the maximum writing rate and the dynamic range selected, a widely used level
recorder may indicate for each tone burst. From this it is obvious that two things need to be done in order to make possible measurements and recordings of short sound events which are both reproducible and meaningful. Firstly, international agreement should be reached on a specification for an impulse sound level meter which would indicate, among other things, the required response to tone bursts ranging from, say, 5 ms to durations which practically correspond to the steady-state indication. A German Standard to this effect already exists (DIN 45 633, Part 2), and it is hoped that work on this subject will begin soon in IEC/TC 29. The curve, labeled dynamic response "Impulse", in Fig. 1 represents the requirements laid down in the said Standard, and deviations from this curve may not exceed ± 2 dB.

Secondly, new measuring equipment must be designed so as to conform to these requirements. It is clear that electronic means will have to be found to overcome the limitations of the mechanical members of such equipment if a response corresponding to an rms time constant of 35 ms is to be realized. In order to enable the meter pointer or the stylus to reach the prescribed position - even
for sound events which are short compared to the mechanical response time - such equipment will require a stretching circuit which holds the value to be indicated long enough for the mechanical motion to be completed. A block diagram of such an arrangement is shown in Fig. 2. The stretching circuit has to be designed such that its rise time is short compared to the 35 ms chosen as the time constant for the integrating RC-network and its discharging time constant is long enough compared to the mechanical response time of the indicating device; a value of 3 s \( \pm 5 \) s has been fixed for this purpose.

With such an arrangement it is no longer necessary to utilize a recording device with an extremely high writing rate and with the resulting overshoot and stability problems. The combination of a DC logarithmic converter with a moderate speed and a common strip chart recorder with DC-input is all that is required to produce an accurate and meaningful record of sound levels.

Another advantage of the combination of the rms detector and the stretching circuit with its discharging time constant well defined is that the results of sound level classifications, for example, a noise exposure index, are independent of the choice of the sampling frequency as long as the period between two samples is short compared to the discharging time constant of 3 s.

It should be noted that a device which will positively indicate any overload condition, be it of very short duration, is an indispensable requirement for such equipment so that erroneous measurements may be avoided in those cases where the crest factor of the signal exceeds 5 at full scale. The rms detector will measure signals with crest factors of up to 5 accurately.

Such equipment as described above is now on the market, and it is hoped that it may serve a useful purpose in making this noisy world a quieter place.

References:
Optimum Design of Carbon Transmitter Considering Feeding D.C. Current
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Introduction
Carbon microphones have been widely used in most of telephone sets. This is due to their desirable features, economical and highly sensitive. However, the carbon microphone requires the D.C. biasing current. In order to save the electric power, it is desirable to reduce the feeding current to the microphone.

In this paper, the method of designing transmitter under various feeding currents, considering the vibrating system and the deterioration of granular carbon aggregates, is described.

Design Principle
The electro-acoustic sensitivity of the carbon transmitter is estimated by the following expression.

\[ K = 20 \cdot \log_{10} p/R = 20 \cdot \log_{10} \mathcal{U} + 20 \cdot \log_{10} \xi + 10 \cdot \log_{10} R \]  

where, \( p \) : sound pressure, \( e \) : output voltage of the transmitter which results from sound, \( R \) : electric resistance of the transmitter, \( \mathcal{U} \) : transduction coefficient \((=\Delta R/ R \cdot \xi)\); \( \xi \) is displacement amplitude of the movable electrode, \( \Delta R \) is electric resistance variation. \( \xi \) is efficiency of the mechanical vibrating system \((=\xi/p)\). For the carbon transmitter as shown in Fig. 1 in low frequency range is expressed by

\[ \xi = \frac{\xi}{p} = \frac{S_d}{(\Delta_d + \Delta_u + \Delta_c)} \]  

--- D-53 ---
where, $S_d$: effective area of the diaphragm, $A_d$: stiffness of the diaphragm, $A_b$: stiffness of the back chamber, $A_c$: stiffness of the granular carbon aggregates.

The quantities $S_d$, $A_d$ and $A_b$ can be calculated by the dimensions of the diaphragm. While the relations between the electro-acoustic parameters $U$, $A_c$, $R$ of the granular carbon aggregates and the carbon chamber sizes --- especially, electrode area, distance between electrodes $X$, radial ratio $\delta$ --- are obtained by experiments as follows (see Fig. 2)

$$
\begin{align*}
A_c &= S_d A_d (1 + a_a X)^n (1 + \alpha_a \delta)^n (1 + \beta_a \delta) \\
U &= U_d (1 + a_a X)^n (1 + \alpha_a \delta)^n \\
R &= R_d (1 + a_R X)
\end{align*}
$$

where, $A_d$: stiffness per unit electrode area at $(X, \delta) \rightarrow 0$, $U_d$: transducing coefficient $U$ at $(X, \delta) \rightarrow 0$, $R_d$: electric resistance per unit electrode area, $(a_a, a_a, a_R)$: form factors of the carbon chamber, $(\alpha_a, \beta_a, \alpha_d)$: coefficients expressing nonlinear characteristics.

The values of these coefficients were determined by experiments.

Using these relations, the sensitivity of the transmitter can be estimated from the dimensions of its vibrating system. However, the deterioration of the electro-acoustic performance of the granular carbon aggregates is closely connected with the density of feeding current $J$ ($= I_k/S$, $I_k$: feeding current to the transmitter). It is desirable, therefore, to design the electrode area so as to give current density which is smaller than a specific value $J_0$. Considering these facts, the maximum sensitivity and the optimum dimensions of the transmitter's vibrating system were investigated by theoretical calculations. In these studies, the electrode area was given in proportion to the designated...
feeding current $I_k$, namely

$$S = \frac{I_k}{J_0} \tag{4}$$

where, $J_0$ is maximum current density which is permissible for the granular carbon aggregates ($= 76 \text{ mA/cm}$).

As a result of these calculations, it was found that the mechanical efficiency $\eta$ has a maximum for the dimensions of diaphragm ($a_d, b_d, V_b$) when the electrode area is given. For an example, Fig. 3 shows the characteristics of $\eta$ vs. diaphragm radius $a_d$. Therefore, an optimum size of diaphragm which brings a maximum sensitivity $K_{\text{max}}(p)$ can be determined for each designated current. The maximum sensitivities and the resonance frequencies corresponding to the optimum dimensions of the vibrating system are shown in Fig. 4 as function of $I_k$.

Experiment and conclusion

The validity of these estimation is verified by actual performance of transmitters made as models, specified for the feeding...
Optimum Design of Carbon Transmitter Considering Feeding D.C. Current

current of 10mA, 25mA and 50mA.

These transmitters with properly designed diaphragms are constructed by the vibrating systems with three degree of freedom in order to extend the frequency band, and their electrodes have an area determined by the relation Eq.(4). For example, Fig.5 shows the cross-sectional view of the transmitter designed for 10mA.

Fig.6 shows the frequency response of these transmitters. These responses nearly consistent with the presumed values from Fig.4.

By the above-mentioned results, we could establish the method of designing transmitters corresponding to a given feeding current.

Author thanks the following persons for their kind instructions, Dr. Gi-ichi Itō, Dr. Ichizō Nakano, Dr. Ryōzō Araki.

![Diagram of Transmitter](image)

Fig.6 Frequency Response of the Transmitters

Reference.
THE 6TH INTERNATIONAL CONGRESS ON ACOUSTICS
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Foil-Electret Microphones and their Applications
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INTRODUCTION

Some basic features of foil-electret microphones, such as their mechanical design, frequency and impulse response, distortion characteristics, and dynamic range have been discussed in a previous paper. More recently, data on the net charge decay of electrets made of polyester (Mylar) and fluorocarbon (Teflon) foils, taken at elevated temperatures as well as over a 2.5 year period at room temperature, were reported and the effect of this decay on the microphone sensitivity was discussed.

In the present paper, we will report on the following new subjects: (1) A depolarization method which makes possible for the first time the separate measurement of the decay characteristics of homo- and heterocharge on foil electrets; (2) measurements of the charge stability of two new foils, namely polycarbonate and polyimide (H-film); (3) the continuation of sensitivity and surface charge measurements at room temperature over a total time of about 5 years; and (4) the use of foil-electret microphones in directional transducers, particularly microphones with toroidal characteristics.

FOIL ELECTRETS

Foil electrets have been made from a variety of dielectric materials, such as polyester, fluorocarbon, polycarbonate, and polyimide, typically 0.25 to 1 mil thick. A procedure for polarizing thin dielectrics is to heat the foils to temperatures up to 150°C (polyester) or 230°C (fluorocarbon, polycarbonate, and polyimide) and expose them to electrical fields between 10 and 100 kV/cm. The polarization of foil electrets can consist of hetero- and homocharge, where the heterocharge is in many cases due to the alignment of dipoles, while the homocharge is due to space charges close to the electret surfaces. The foils investigated in this study show fields corresponding to a homocharge. A "dissociable capacitor" method was used to determine the polarity and the quantity of the surface charge on foil electrets.

DECAY OF THE SURFACE CHARGE AT ELEVATED TEMPERATURES

The decay processes for space charges and dipole alignments on electrets are controlled by the conductivity of the foil material and by its dielectric relaxation, respectively. The time constants for resistive and dielectric decays can be expressed by $\tau = \tau_0 \exp(-E/kT)$, where $E$ is an activation energy. This suggests Arrhenius plots $\log(t/\tau_0) = -E/kT$ to determine time constants over a range in which the activation energies and $\tau_0$ do not change substantially.

THE CURRENT METHOD OF DEPOLARIZATION

Time constants at elevated temperatures have been obtained by two different methods. In one method ("current method"), the foil electret is exposed to increasing
temperatures between two disk electrodes in close contact to the foil surfaces and the depolarization current is measured as a function of temperature. The current usually changes sign as the temperature rises, indicating the independent depolarization of homocharge and heterocharge. The time constant $\tau(T)$ at a temperature $T$ reached at time $t_1$ can then be obtained from the current $I(t_1)$ at that temperature and the remaining quantity of charge $Q(t_1)$ by the equation

$$\tau(T) = Q(t_1)/I(t_1) = \left(\int_{t=t_1}^{\infty} I(t)dt\right)/I(t_1). \quad (1)$$

Time constants obtained from such measurements for polyester and FEP fluorocarbon electrets are shown in Fig. 1 as a function of $1/T$. As expected, log $\tau$ is found to be proportional to $1/T$. If extrapolated to room temperature, the time constant for the homocharge decay of polyester is 1.5 years while that for the heterocharge decay is about 0.5 years. Activation energies may be determined from the slopes of the lines in Fig. 1. For the homocharge decay, the activation energies below and above the glass transition temperature of polyester (80°C) are apparently not too different. This is evident from the fact that the extrapolated time constant of the homocharge decay corresponds to that found directly at room temperature (see below) for the net charge decay.

For the homocharge decay of FEP fluorocarbon at room temperature, Fig. 1 yields an extrapolated time constant of about 50 years, corresponding again to that found directly at room temperature (see below). The time constant of the heterocharge decay at room temperature is found to be 0.5 years and is thus about equal to the dielectric relaxation time. This indicates that homocharge and heterocharge are, as expected (see above), of different nature.

**THE CHARGE METHOD OF DEPOLARIZATION**

The second method for determining time constants at elevated temperatures consists of measurements of the surface charge before and after the electret has been exposed to temperatures above room temperature ("charge method"). In this case, the foil is heated either suspended freely or making contact to metal surfaces.

In Fig. 2, the time constants of the net charge decay of 1-mil polyester, polyimide, fluorocarbon, and polycarbonate electrets, as determined with the "charge method", are shown as a function of $1/T$. Before these measurements were taken, the foils were subjected to some aging at elevated temperatures after
polarization, resulting in a loss of about 50 percent of the original electret charges. This aging increases the measured time constants considerably. The polycarbonate foils were also baked before polarization in order to remove moisture. Measurements taken at room temperature are also included where available.

The straight line extrapolation (Fig. 2) of the time constant for polyester and FEP fluorocarbon electrets agrees well with the results obtained at room temperature, indicating relatively constant activation energies in the temperature range under consideration. The extrapolated time constants of FEP fluorocarbon and KI polycarbonate are about 100 years. Polyimide is clearly represented by two or more straight lines, indicating a change in activation energy at about 100°C.

Due to the preheating, most of the heterocharge was removed prior to these experiments. The measured time constants are therefore those related to the homocharge decay. They agree well with the corresponding time constant obtained with the "current method". Further discussion of the two methods of depolarization will be published.  

SURFACE CHARGE DECAY OF FOIL ELECTRETS AT ROOM TEMPERATURE

The decay of the surface charge of foil electrets as a function of time at room temperature is shown in Fig. 3 for several polyester, fluorocarbon, and FEP fluorocarbon electrets. Previous measurements have been extended to periods of about five years with the foils stored unshorted and in normal room atmosphere. Aging at elevated temperatures after polarization (to remove less resistant charges) was not applied to these foils. Most of the decays are nonexponential.

For polyester, the time "constant" of the decay has initial values of about 0.2 years and increases to about one year. For fluorocarbon and fluorocarbon FEP, the time constants over the last year of the measurements are about twenty and fifty years, respectively. Other measurements not shown in Fig. 3 indicate that BPA-poly carbonate has time constants of about two to ten years, while KI polycarbonate, although only measured over a period of about 1.5 years, seems to have a lifetime of more than 10 years. Polyimide has a time constant of only 0.1 years. The small fluctuations superimposed on the decays are probably due to environmental changes. Aged foils show initially larger times constants.

THE ELECTRET MICROPHONE

The diaphragm in the electret microphone consists of a foil electret covered with an optically thin metal layer on one side. The foil electret is stretched across a metallic backplate with the metal side facing out. A sound wave impinging on the microphone changes the air layer between foil and backplate and thus disturbs the electric fields. This gives rise to a voltage across a terminating resistance.

The decay of the sensitivity of various polyester and fluorocarbon microphones is plotted in Fig. 4. Measurements over periods up to five years with the microphones stored in normal room atmosphere show polyester systems to have a relatively constant sensitivity for a period of about 1 year. Thereafter, the sensitivity decreases more or less exponentially with time constants from one to two years, corresponding to the
constants found for the decay of the net surface charge. The fluorocarbon systems exhibit smaller fluctuations in sensitivity and show no decay for the time measured (about 4 years). The fact that the sensitivity is constant in spite of the surface charge decay has been explained by a compensation mechanism.  

DIRECTIONAL MICROPHONES

Because of the mechanical and electrical flexibility of the electret transducer, many special types of microphones are more easily implemented. For example, a second order gradient transducer with toroidal directional characteristics, using a single electret microphone diaphragm, has been designed on a principle suggested by M. R. Schroeder. Such microphones have minimum sensitivity in the direction of the rotational axis of the toroid and maximum sensitivity in the plane normal to that axis.

In this plane of maximum sensitivity, the experimental unit has a response which is uniform within ±3 dB for frequencies lower than 2.5 KHz. In a plane through the rotational axis the sensitivity is found to obey a cosine-squared law with maximum attenuation of 20 dB in the mid-frequency range. The equalized frequency response of this microphone is constant within ±2 dB between 0.3 and 3 KHz and corresponds to approximately -80 dBV per pbar. Microphones with toroidal directional characteristics may find use in conference telephony and similar applications where pickup in one plane is desired.

CONCLUSIONS

Various methods have been used to measure time constants of the charge decay of foil electrets. A method utilizing measurements of the depolarization current at elevated temperatures ("current method") made it possible for the first time to determine separately the decay characteristics of homo- and heterocharge of foil electrets. In many cases, results on the time constants of homo- and heterocharge decay at elevated temperatures can be extrapolated to room temperature by Arrhenius-plots. Other measurements of the life-times were made by determining the change in net surface charge of the electret after repeated exposure to elevated temperature ("charge method"). This method measures therefore the net decay of homo- and heterocharge. Time constants of the net surface charge decay have also been determined directly at room temperature. Measurements performed with the different methods yield similar results. The forming of fluorocarbon and polycarbonate foil electrets, which have time constants of the charge decay of about 100 years, made possible electret microphones with correspondingly long lifetimes.

ACKNOWLEDGMENTS

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REFERENCES

Details in the Construction of a Piezo-electric Microphone.

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Two of the basic problems facing the designer of a piezoelectric microphone using a bender type ceramic element are how to mount the bender, and how to attenuate its natural resonance.

This paper describes some of the considerations involved and shows how the problems have been solved in the practical construction of a microphone.

Mounting of the Bending Element.

Basically two different methods of mounting seem appropriate: Simple support of both ends of the bending element, or cantilever mounting. For a given microphone diameter the simply supported bender can be twice as long as that used for cantilever mounting.

Fig. 1 shows the shape of the bending moment curve for the two cases when the bending element is loaded by the same force, P. If the bending moment curves are integrated, the value of the integral will be the same in both cases. This again means that the charge sensitivity of the design is also equal in both cases. However, the compliance of the cantilever construction is twice that of the simply supported bender. To reduce losses from air stiffness and from edge stiffness of the microphone diaphragm the compliance of the bending element construction should be small, a fact which favours the simply supported construction with a factor of 2.

Another important fact to be considered is the effective mass of the construction. This should be small to reduce the sensitivity to vibrations of the microphone. For the simply supported bender case the effective mass is four times larger than for the cantilever type construction. It should, on the other hand, be pointed out that in the microphone design considered the mass of the diaphragm is of the same order of magnitude as the mass of the bender. Thus the sensitivity to vibration is only a factor 2 larger in the simply supported bender case than in the cantilever case.

However, a fact which actually decided the type of support finally chosen for the practical construction was that the angular displacement of the bender at the point where this is loaded by the microphone diaphragm is zero for the simply supported system, while it is maximum for the cantilever type construction. If an angular displacement is transferred to the diaphragm, this will no
Piezo-electric Microphone

longer describe pure translational movements only but it will also be
twisted, which is highly undesirable
for the achievement of sufficient at-
tenuation of the system resonance.

Fig. 2 shows how the simply sup-
ported construction was realized in
practice. On both ends of the bend-
ing element a bronze band has been at-
tached, bent at right angles and
fastened by means of a ring. The int-
egral of the bending moment curve is
then:

\[ \int_0^l M(x)dx = \frac{P}{8} \left[ \frac{1}{1+2ab} \right] \]

where

\[ a = \frac{E_b}{E_b} \cdot I_b \]

and \( b = \frac{l}{h} \)

It can be seen that this expres-
sion tends towards \( P^2/8 \), as in the
case of the simply supported bender,
when \( a \times b \) goes to 0, i.e., when long,
thin bronze bands are used. Also it is
seen that when \( a \times b \) tends towards
infinity the above expression and thus
also the microphone sensitivity, tends
towards zero.

In the actual case the bronze
bands are 0.078 mm thick and 2 mm long.
This results in a reduction in sensiti-
vity of approximately 5% relative to the
maximum achievable sensitivity. One of
the advantages of this realization of a
simple support is that it is very stable
because it consists of fixed joints only.

Attenuation of the Natural Resonance.

One of the requirements to the
microphone was that it should have a
flat free-field response up to around
10 kHz for sound incidence perpendicu-
lar to the diaphragm.

Due to the diffractions around the microphone a frequency dependent pressure
increase is formed at the microphone diaphragm as shown in Fig. 3. To achieve a
flat free-field characteristic it is therefore necessary that the pressure re-
sponse of the microphone shows a fall-off with frequency which corresponds to
the above mentioned increase in pressure, see Fig. 4. On the figure is also
shown the voltage across the capacitor in a series resonance circuit having a
resonance frequency of 5 kHz and a loss-factor of 1.7. As can be seen, this
curve is very close to the desired pressure characteristic.

Originally a microphone construction similar to that shown in Fig. 5 was sug-
gested. Here a number of holes, covered by acoustic damping material, e.g.
tightly woven cloth, connects the cavity behind the microphone diaphragm with
a somewhat larger cavity. An approximately equivalent circuit for this type of
design is shown in Fig. 6. The "dashed" capacity represents the cavity just be-
Piezo-electric Microphone.

![Graph 1](image1.png)

**Fig. 3.** Pressure increase for incidence of sound perpendicular to diaphragm.

![Graph 2](image2.png)

**Fig. 4.** Solid curve: Desired pressure response to give flat free-field response. Dotted curve: Voltage across capacitor in series resonant circuit, $f_0 = 5$ kHz, $d = 1.7$.

Behind the diaphragm. It can be shown, quite easily, that with the ceramic bending elements presently available, and with a desirable loss-factor of 1.7, the cavity behind the diaphragm must be smaller than some 0.03 cm$^3$. For a microphone diameter of 23.77 mm this corresponds to a distance between the diaphragm and the back-"wall" of roughly 0.1 mm. A distance of this order of magnitude requires, however, an extreme conformity of diaphragm and back-"wall". This again requires extreme care in the manufacture and assembly of the microphone, and it was therefore decided to try to obtain the desired loss-factor solely by means of the air viscosity in a narrow cavity between the diaphragm and a back-plate.

![Diagram](image3.png)

**Fig. 5.** First sketch of the microphone.

![Diagram](image4.png)

**Fig. 6.** Simplified equivalent circuit.

The design is shown in Fig. 7. As can be seen, no acoustic damping material has been used, whereby tightening problems have been avoided and the diaphragm has been connected directly to the bending element. A further advantage of this construction is that the loss-factor can be adjusted to the desired value simply by varying the distance between the diaphragm and the back-plate. In this way it is possible to vary the loss-factor within very wide limits.

Fig. 8 shows the pressure characteristic of the microphone for different distances between the diaphragm and the back-plate, measured by means of an electrostatic actuator. The loss-factor varies here between some 0.75 and 5. On the curves some irregularities can be seen at the high frequency end. These
Piezo-electric Microphone.

are caused by resonances in the back-plate.
Finally, Fig. 9 shows an example of the frequency characteristic achieved by means of the above mentioned measures. The lower curve represents the microphone pressure response measured by means of an electrostatic actuator while the upper curve is the free-field response to sound incidence perpendicular to the diaphragm.

The following data have been obtained for the microphone:

Voltage sensitivity: 0.3 mV/μbar
Capacity: 4000 pF
Resonance frequency: 4.8 kHz
(90° phase shift)
Lower Limiting Frequency: 3 Hz
Sensitivity for Vibrations: 1 g represents 100 dB S.P.L.

Fig. 8. Pressure response for different spacing between the diaphragm and back plate.

Fig. 9. Example of frequency response.
Upper curve: Free-field response.
Lower curve: Pressure response.

Components in the Equivalent Circuit:

\[
\begin{align*}
M &= 45 \times 10^{-6} \text{ Kg} \\
C &= 24.4 \times 10^{-6} \frac{\text{m}}{\text{N}} \\
R &= 2.31 \frac{\text{N} \times \text{s}}{\text{m}} \\
N &= 10 \frac{\text{V}}{\text{N}}
\end{align*}
\]
Optimum Design of Electromagnetic Receivers

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Introduction

Sensitivity of electromagnetic receivers is expressed as a function of various dimensions of vibrating and driving systems, and conditions for obtaining high sensitivity are derived. Experimental results for three kinds of model receivers agree with the calculated results.

Specific Response

Let us consider the receiver as shown in Fig. 1. A diaphragm has cone shaped structure with an armature at the center. Magnetic circuit is arranged concentrically, and acoustic vibratory system of 3-degrees of freedom is used.

The sensitivity (specific response) of the receiver at low frequencies is represented as follows:

\[ \eta = \frac{P}{\sqrt{2}} = \frac{A}{\sqrt{2}} \frac{K S_o}{V_l} \frac{1}{\omega_o^2 m_o} \frac{2}{1 + \sqrt{1 + \frac{4 \frac{K S_o^2}{\omega_o^2 V_l m_o} i}} \frac{i}{i}} \]

Fig. 1 The structure of receiver
where: \[ \omega_0^2 = \left( \omega_0^2 - \omega_n^2 + \omega_r^2 \right) / m_0 \] (2)

\( \rho \): sound pressure developed in the acoustic load, \( I \): input current, 
\( Z \): electric free impedance, \( A \): force factor of the driving system, 
\( K \): modulus of volume elasticity of air, \( V \): volume of acoustic load, 
\( (m, x, S) \): effective mass, effective stiffness and effective area of the diaphragm, 
\( \alpha \): ratio of front chamber stiffness (including \( V \)) to sum of front and rear chamber stiffness, 
\( \alpha_n \): negative stiffness, 
\( \alpha_r \): total stiffness of chambers.

The mechanical constants \((m, x, S)\) of the diaphragm are expressed by dimensions of the diaphragm. Magnetic coupling factor \( k \) between voice coil and magnetic air gap, and negative stiffness \( \alpha_n \) are given by Eq. (3) and Eq. (4) respectively:

\[ k^2 = \frac{A^2}{x_n L_d} \] (3)

\[ \alpha_n = \frac{2 k^2 B^2}{f (1 + \alpha)} \] (4)

where \( L_d \): damped inductance of the coil, \( B_{sat} \): saturation flux density in the armature, \( \alpha \): ratio of air gap reluctance to the reluctance in series with it.

By using above relations, we obtain the low-frequency specific response of the receiver as follows:

\[ Q_o = \frac{K}{2 \omega_0^2 V} \cdot \frac{\pi \alpha^2 (2 - \nu)^2}{\left\{ \pi \alpha^2 \cdot \frac{L_d}{h_d} \left( 1 - \frac{22}{11} \nu \right)^2 + \frac{\pi}{\alpha^2} \cdot \frac{L_d}{h_d} \left( 1 - \frac{22}{11} \nu \right) \right\}^2} \]

\[ \times \frac{1}{1 + \left[ \frac{K}{4 \omega_0^2 V} \cdot \left\{ \pi \alpha^2 \cdot \frac{L_d}{h_d} \left( 1 - \frac{22}{11} \nu \right)^2 + \frac{\pi}{\alpha^2} \cdot \frac{L_d}{h_d} \left( 1 - \frac{22}{11} \nu \right) \right\}^2 \right]} \]

\[ \times \frac{\sqrt{2 \kappa \cdot B_{sat} \cdot B_{sat}}}{\sqrt{\omega_0^2 (1 + \alpha)}} \cdot \left[ \frac{R^2}{\omega_0^2} \cdot \left( 1 + \frac{2 \kappa^2 B^2}{\omega_0^2 (1 + \alpha)} \cdot \left\{ \pi \alpha^2 \cdot \frac{L_d}{h_d} \left( 1 - \frac{22}{11} \nu \right)^2 + \frac{\pi}{\alpha^2} \cdot \frac{L_d}{h_d} \left( 1 - \frac{22}{11} \nu \right) \right\}^2 \right] \] (5)

* Eq. (4) is obtained, when

i) air gap length \( g \) is arranged so that \( A \) is made to be maximum,

ii) inner and outer pole face area are equal.

For preventing adhesion of the armature to pole pieces, it is desirable to make \( \alpha = 0.2 \sim 0.8 \).

— D - 66 —
Optimum Design of Electromagnetic Receivers

Restriction on Dimensions

It is necessary that the stability factor ($\lambda_0/\lambda_m$) and the resonance frequency of the diaphragm should not be less than designated values ($\mu_{\text{crit}}$ and $\omega_0$ (Eq.2)). From these conditions, the restrictions are given as follows:

\[ \pi E \frac{h_d^2 (2-\nu^2)}{(1-\nu^2)\alpha^2 \nu^2} \frac{\beta (1+\delta)}{2h_a B_{am}} \geq \mu_{\text{crit}} \quad \cdots \quad (6) \]

\[ \frac{\pi E h_d^2 (2-\nu^2)}{(1-\nu^2)\alpha^2 \nu^2} \frac{\beta (1+\delta)}{2h_a B_{am}} \leq \omega_0^2 \quad \cdots \quad (7) \]

where $E$ and $\nu$ are Young's modulus and Poisson's ratio of the diaphragm material.

Numerical Results and Optimum Design

Some numerical examples of the specific response are shown in Fig.2(a,b), where $\omega_0=2\pi \times 1700\text{Hz}$, $\omega=2\pi \times 80\text{Hz}$, $\beta=0.17\alpha_m$, $h_d=1.192$, $\delta=0.3$, $\alpha=0.7$, $\beta=0.3$, $V_e=6\text{cm}^3$, $\mu_{\text{crit}}=15$. The possible regions which are limited by Eq.(6) and Eq.(7) are represented by solid curves.

From these results, if the radius $a$ of diaphragm remains constant, the specific response can be increased by reducing the radius $a$ of armature and by increasing the thickness $h_d$ of armature, but, on the other hand, the possible regions become narrower. The effect of thickness $h_d$ of diaphragm on the specific response is rather small in the

![Fig.2](image)

Fig.2 The calculated values of the specific response as a function of $h_d$ with parameters $h_a$ and $h_a$. 

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Optimum Design of Electromagnetic Receivers

possible region. Then, designing procedure will be as follows:

First, radius $a$ of diaphragm is decided. Next, radius $r$ of armature is selected as small as possible considering magnetic edge effect. Finally, the other dimensions in the possible region are decided.

Experimental Results

From the above results, three model receivers are made. For all receivers, magnetic circuits are adjusted so that the force factor has maximum at $g = 0.17\text{mm}$. The specific response of these three model receivers are shown in Fig.3. The dimensions of these receivers and the calculated and experimental values of the specific response at $180\text{Hz}$ are shown in Table 1. From these results, it is concluded that the calculated values agree with the experimental values.

![Fig.3 The specific response curves of three model receivers.](image)

![Table 1 Dimension and Response(at 180Hz)]

<table>
<thead>
<tr>
<th></th>
<th>Diaphragm</th>
<th>Armature</th>
<th>Response</th>
<th>$0\text{dB} = \sqrt{\frac{M}{\text{max}}}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Type 1</td>
<td>$20\text{mm}$</td>
<td>$0.18$</td>
<td>$0.06$</td>
<td>$4\text{mm}$</td>
</tr>
<tr>
<td>Type 2</td>
<td>$15\text{mm}$</td>
<td>$0.25$</td>
<td>$0.06$</td>
<td>$4\text{mm}$</td>
</tr>
<tr>
<td>Type 3</td>
<td>$10\text{mm}$</td>
<td>$0.20$</td>
<td>$0.03$</td>
<td>$5\text{mm}$</td>
</tr>
</tbody>
</table>

References

Polypeptides Piezoelectric Transducer

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Isao Yamamuro and Masahiko Tamura
Pioneer Electronic Corporation, Tokorozawa, Saitama

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Introduction

Piezoelectric property has been reported in a variety of polymeric substances such as wood cellulose, silk fibroin and tendon collagen.\(^1\) Piezoelectric effect in poly-\(\gamma\)-methyl-L-glutamate and poly-\(\gamma\)-benzyl-L-glutamate has been recently discovered.\(^2\) Films of these polymer were cast from solution and elongated to about twice of their original length at a temperature higher than 150\(^\circ\)C. A shear impressed in the plane of the film produced a polarization on the film surfaces.

Assigning z axis to the direction of orientation, x axis perpendicular to z axis in the plane of the film, and y axis perpendicular to both z and x axes, the piezoelectric effect is represented by a piezoelectric modulus \(d_{25}\) which correlates the polarization in y axis \(P_y\) to the shear stress in \(xz\) plane \(S_{xz}\): \(P_y = d_{25} S_{xz}\). Since the polymer films are usually viscoelastic, it is expected that a time lag would exist between applied stress and resulted polarization. Then the piezoelectric modulus is a complex quantity given by \(d_{25}^* = d_{25}^r - j d_{25}^\prime\).

Measurement of Piezoelectric Moduli

A rectangular sample was cut out of the film 45 degree obliquely to the elongation axis and thin metal foils were attached to the both surfaces of the sample as electrodes. Sinusoidally varying strain was
Polypeptides Piezoelectric Transducer

given to the film sample and the amplitude of the sinusoidal electric polarization on the electrodes as well as the phase difference between the stress and the polarization were determined.

Figure 1 illustrates the temperature dependence of real and imaginary piezoelectric moduli $d_{25}'$ and $d_{25}''$ for an oriented poly-$\gamma$-methyl-L-glutamate film. Dispersions of the piezoelectric moduli are seen at about 0°C and 100°C.

It is interesting to note that the increase of $d_{25}'$ with temperature is accompanied by the positive value of $d_{25}''$ (polarization delayed behind stress) and that the decrease of $d_{25}'$ with temperature is accompanied by the negative value of $d_{25}''$ (polarization overreached stress).

In order to find out the reason for the temperature variation of $d_{25}'$ and $d_{25}''$ the dynamic viscoelastic measurements were carried out for the same specimen. The elastic modulus $E'$ and loss modulus $E''$ in the direction of orientation are shown in Fig. 2 as functions of temperature. Dispersions of elastic moduli take place at the same temperatures as those for piezoelectric dispersions. The dispersion at about 0°C is usually ascribed to the onset of the thermal motion of side
chains and the dispersion at about 100°C to the start of the large molecular motion in main chains.

**Molecular Interpretation**

Piezoelectric polarization in poly-γ-methyl-L-glutamate is assumed in origin to be produced by the stress induced orientation of dipoles in the molecule. In the present specimen most of the molecules take the helical conformation (α-helix). Peptide group (-CO-NH-) in main chain and polar group in side chain (-CO-O-) should be dipoles responsible for generation of piezoelectric polarization. At low temperatures the stress induced orientation of these dipoles increases with temperature, since the flexibility of molecules increases with the rise of temperature. The decrease of piezoelectric modulus at 0°C and 100°C would be caused by the randomization of dipolar orientation due to the thermal motion of side chains and main chains respectively.

Piezoelectric properties in poly-γ-methyl-D-glutamate were also examined, which has a molecular structure with a mirror image symmetry to poly-γ-methyl-L-glutamate. The temperature dependence of $d_{25}'$ and $d_{25}''$ in the D polymer was quite the same, but their signs were opposite to those in the L polymer.
Polypeptides Piezoelectric Transducer

Application to Transducers

If the specimen was well elongated and annealed at about 180°C, the value of piezoelectric modulus was attained to about $1 \times 10^{-7}$ c.g.s. e.s.u. at room temperature, which was higher than that of quartz crystal. The elastic modulus of the film was $4 \times 10^{10}$ dyne/cm$^2$ in the direction of orientation and $1 \times 10^{10}$ dyne/cm$^2$ in the direction perpendicular to the orientation. The elastic moduli were less than 1/20 of that of quartz crystal. These figures would suggest the possible use of such oriented synthetic polypeptides as a flexible piezoelectric transducer element.

For the purpose of illustration, a headphone as shown schematically in Fig. 3 was set up, in which a strip of poly-$\gamma$-methyl-L-glutamate is used as an electromechanical transducer. The sound intensity and frequency characteristics shown in the figure promises the general use of the present device. Phonograph pick-up and microphone with poly-$\gamma$-methyl-L-glutamate film transducers were also constructed.

1) E. Fukada; Ultrasonics, in press
2) E. Fukada, M. Date, N. Hirai: Nature 211, 1079 (1966)
Artificial Mastoid
Hearing by bone conduction provides an alternative to the normal method of hearing in case of conductive hearing loss, where use is made of bone vibrators. In addition bone conduction audiometry can, in conjunction with air conduction audiometry, provide differential diagnosis of hearing defects. It follows that some standardization of how to describe and measure the performance of the bone vibrator, is necessary to progress, both in the clinical situation and for the exchange of data in research work. This work requires an Artificial Mastoid, which must provide a mechanical load impedance equal to that of the average human forehead or mastoid and a built-in transducer to measure the vibromotive force exerted by the bone vibrator.

In the field of air conduction measurements international standardization of an Artificial Ear has recently been achieved, 1). The real development of the artificial ear first took place when careful measurements of the entrance impedance of the human ear were made, using only accurate physical methods, disregarding both leakage problems and subjective tests, 2), 3), 4).

Compared with the development of the artificial ear, our present knowledge of the problems artificial mastoid is very primitive. The requirements for an artificial mastoid can be summarized as follows:
1) It must present to the bone vibrator under test, the same mechanical impedance as the average human mastoid over the required frequency range, usually 250 Hz to 4000 Hz, but in the future extended to 125 Hz to 6000 Hz.
2) By means of a transducer it must indicate the vibration of the bone vibrator below the skin when the bone vibrator is pressed against the artificial mastoid in nearly the same way it is fixed to the human head. The indication may be in terms of vibromotive amplitude, velocity or acceleration.
3) The mechanical impedance elements and transducer must be stable with both time, temperature, humidity, pressure and most important of all, stable for normal handling and use.

It is generally agreed that the human mastoid can be described with a very rough approximation with the simple electrical analogue shown at fig. 1. Here V o is the analogue of the force transmitted to the head bone m, which causes the sense of hearing. M 2 represents the effective dynamic mass of the skin. This mass is less than 1 g. r and s represent the internal viscous damping and the stiffness of the skin layer. If this simple picture was correct, an artificial mastoid could easily be built, as shown on fig. 2, where \( L_2 = \text{abt. } 1 \text{ g} \) is
acting through a spring parallel with a fluid coupling. Such a device is made, 5), 6), and is also in use at several places. The only practical difficulty is that the mass $m_2$ is very small and to obtain the viscous coupling with air-damping the mechanical tolerances become extremely tight, making the whole instrument very delicate.

![Figure 1: Simplified electrical analogue for the mastoid](image1)

But unfortunately careful investigation, 7), 8), 9), have shown that nature is not so simple that the human mastoid can be described with the simple analogue given in Fig. 1. The problem is that both the effective dynamic mass $m_2$ of the skin, as well as both the viscous damping $r$ and the stiffness $s$, vary with both frequency, static pressure and contact area of the bone vibrator. In other words the simple realization in Fig. 2 can only be correct for some given frequencies and given contact areas and static pressures on the bone vibrator. The absolutely simplest equivalent network of the skin tissues consisting of 6 elements, 7), making a mechanical realization consisting of separate well defined masses, springs and viscous damping elements very difficult, may be even impossible.

The straightforward way of solving the problem is to make an artificial skin layer which has nearly the same thickness, mass density and stiffness characteristics and mechanical hysteresis as the human skin. If such an artificial skin layer could be made on non-ageing rubberlike material, it is almost certain that the distributed masses, the change of stiffness with static pressure and the variation in viscous damping with frequency and contact area, will be very close to that of the human skin layer stretched over the skull.

In complete analogue with the method used for determining the transfer properties for the artificial ear by using a high impedance sound source and measuring the generated sound pressure with a probe microphone at the ear entrance 2), 3), we could here use a high impedance bone vibrator and just measure the vibration as function of frequency, static pressure and contact area of the skull bone. In this way we could determine the human skin layer transfer properties and compare that with our artificial made layer. If sufficient agreement is obtained between the transfer of the real human skin layer and our artificial layer we would have solved the problem. - The difficulty is that it is impossible to fasten an accelerometer on the skull bone inside a living human head and therefore indirect methods have to be applied.

At NFL in England, 10), an Artificial Mastoid is developed using indirect methods. The mechanical realization as seen at figure 3 has been made in accordance with the thought that the mastoid should be so closely related to the human mastoid as possible. The NFL version consists of a 3,5 kg mass with built-in transducer representing the skull bone. Over the curved "bone" is glued a skin layer built up of two different artificial rubber types. This mastoid is, since 1966, a British Standard, 4), and in B.S.4009:1966 is a very detailed description of all the important dimensions and chemical compositions of the artificial rubber. Also the calibration methods are given.

Dalgaard has in a report, 12) compared the wellknown U.S. Mastoid as outlined in Fig. 2 with British Standard type. Here some of the mainpoints are summarized:
Artificial Mastoid

General principles: A certain discrepancy exists between the values of the parameters in the circuit diagram. In the table below are given the values of the British and the American proposal expressed in MKS units. The British values were calculated from the total impedance values in the middle frequency range.

<table>
<thead>
<tr>
<th></th>
<th>m_2</th>
<th>r</th>
<th>c</th>
<th>m_1</th>
</tr>
</thead>
<tbody>
<tr>
<td>UK</td>
<td>0.8 ( \cdot 10^{-3} )</td>
<td>20</td>
<td>3.8 ( \cdot 10^{-6} )</td>
<td>3.5</td>
</tr>
<tr>
<td>US</td>
<td>1.5 ( \cdot 10^{-3} )</td>
<td>38</td>
<td>2.5 ( \cdot 10^{-6} )</td>
<td>3</td>
</tr>
</tbody>
</table>

Table - Values in MKS units

It should however be noted that the British values apply to a force of application of 7.5 newton and a driver tip area of 1.75 cm² whereas the American values apply to a force of application of 5 newton and a driver tip area of 3.14 cm².

American proposal: The mastoid consists of a central magnesium disk with a flat surface suspended to a large mass by three flat springs also of magnesium. The bottom portion of the disk forms a cup which mates with the larger mass forming an acoustic network existing of a cavity and annular air clearance. The necessary resistance is produced as the viscous damping in the air clearance. In order to keep the damping constant up to the upper frequency limits required for the bone calibration, the cavity must be kept as small as practical possible and the ratio of inertial to viscous forces in the clearance should be as low as possible. (Fig. 4). Since the damping is a cubic function of the clearance, care must be taken in the design to minimize change in this dimension as a function of temperature. The force acting on the larger mass is measured by a sensitive piezo-electric accelerometer. The entire structure is supported by compliant members which are fastened to a large metal base. The features of this construction are: The viscous damping is stable in time and independent of the force of application.

The use of a flat metal surface for the application of the bone receiver under test gives rise to certain difficulties when the bone vibrator has a curved surface. It might be necessary to use adjustment pieces in order to secure a good coupling between the bone vibrator and the artificial mastoid.

British proposal: The mechanical impedance presented to the bone vibrator is essentially approximated by pads on non-ageing visco-elastic material attached to an inertia terminal of mass 3.5 kg. mounted on a resilient suspension such that the resonance frequency does not exceed one tenth of the lowest frequency at which measurements are made.

The surface contour of the artificial mastoid is spherical with a radius of curvature of about 9.6 cm.

In order to secure that the properties of the visco-elastic material have the desired values, the proposal contains a detailed description of the manufacturing procedure. Furthermore a method of calibrating the mastoid is included in order to control that the performance remains stable.

The features of the construction are: The mechanical impedance varies with static coupling force substantially in accordance with measurements on human heads. The instrument is specified in terms of the static coupling force used.
Artificial Mastoid

Internationally for audiometric standardization, but should also be suitable for the comparison of hearing aid bone vibrators typically attached with lower coupling forces.

The use of visco-elastic materials might introduce a long time instability which however can be controlled. It is to be expected that the material employed, secures that regardless of the shape of the bone vibrator surface a good coupling between the vibrator and the mastoid is obtained.

At B & K, Denmark, we are working on an artificial mastoid along with the NFL layout. We have been concentrating on the stability of the artificial skin layer and developing a very precise calibration.

We have used the artificial rubber composition described in the British Standard and have found it not too difficult to make different batches sufficiently alike, but the technique of gluing the two layers together and gluing these on the curved skull bone require some care. A little air bubble between the layers imposes large differences with higher frequencies as it is seen in fig. 5.

To keep a constant check of the mastoid a reliable and accurate bone vibrator with built-in accelerometer is indispensable.

Also such a device has been developed as seen on fig. 6 which include means of checking the static pressure.

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**Literature (Mastoid)**

1. IEC recommendation sent out under the six month rule 1968.
2. Ørnel, Frederiksen & Bammusen: Artificial Ears for the calibration of the External Type.
A New Method in Stroboscopy

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Introduction.

A new working principle for stroboscopes, designed mainly around pulse circuits and offering exceptionally high long term stability, has been developed and incorporated into the new B&K Stroboscope. This design provides two new essential features in stroboscopy, namely:

1. The instrument will accept any periodic waveform at its input
2. The phase in which the moving object is observed remains the same, once set, even if the input signal varies.

In this way a very useful tool for the visual inspection of vibrating and rotating phenomena is obtained.

Description of the Instrument.

To allow for the good features mentioned in the introduction the input signal to the Stroboscope is converted into rectangular pulses which again controls a signal sawtooth generator.

The purpose of the sawtooth generator is to obtain a linear phase deviation range between 0° and 360°, and this can be achieved simply by comparing the instantaneous values of the sawtooth waveform with a variable DC. Each time the two voltages are of equal magnitude a pulse releases a flash, see Fig.1. If the DC voltage is replaced by a sawtooth wave, a slow-motion effect takes place. The degree of slow-motion is directly controlled by this sawtooth wave, and varies between 0.5 Hz and 2 Hz.

Fig.1. SIMPLIFIED BLOCK DIAGRAM OF THE STROBOSCOPE
A New Method of Stroboscopy

With vibrating objects it seems as though 2 Hz slow-motion frequency is the most convenient, whereas 0.5 Hz is the better frequency for rotating objects.

When increasing the signal frequency a built-in count-down circuit will automatically count-down the flashing rate in such a way, that the phase in which the moving objects is observed will not be disturbed. Thereby the mean light output appears as practically constant in the entire frequency spectrum.

The formula governing the flash rate is:

\[ f_r = \frac{f_s + \Delta f}{n} \]

\( f_s \) = signal frequency, \( \Delta f \) = slow-motion frequency, \( n \) = positive integral number.

Some Applications of the Stroboscope.

Calibration. During the calibration of accelerometers it is very important to be able determine the vibration amplitude exactly. With the stroboscope the maximum displacement of the object can be measured simply by turning the phase-deviation knob 180° from one peak of excursion to the other. If extremely high accuracy is required the external synchronising signal can be rectified by means of a full wave rectifier thus obtaining an input signal of twice the frequency. The stroboscope lamp thereby flashes with the double frequency.

A point on the test specimen illuminated by the flash, will now appear as two static points the distance between which can be adjusted by turning the phase deviation knob so as to indicate the maximum displacement, see Fig. 2. By means of a microscope it will be possible to measure displacements of the order of micrometers. The use of a flashing light source gives a better determination of the peak amplitude than can be obtained with a constant light source.

Product control. In the checking or adjustment of mechanical relays the stroboscope has been used. When the repetition rate of the relay switching is adequately high (>15 Hz) the movement can be controlled in each phase, e.g. if two contacts are activated from one solenoid, cooperation between contacts can be visually inspected and adjusted. Furthermore, the proper operation of the springs can be checked in order to avoid contact chatter.

Running gearwheels may be inspected by means of the stroboscope detecting insufficient mesh due to wrong shape of the cogs. To do so a magnetic transducer is placed close to the gearwheel and the signal is fed to the stroboscope.
Research.

To study phenomena around the building of drops in fluids the set-up, shown in Fig. 3, can be advantageously used. When the drops pass the photosensitive device with regular intervals the phenomenon can easily be examined.

The electrical signal from this set-up is especially suitable for the stroboscope because of its impulsive character. The input circuit is so designed that the triggering level can be varied around the zero crossing. In the case of uncertain triggering it may therefore help to adjust the triggering level.

Another interesting application of stroboscopy is the examination of musical instruments. The dynamic behaviour of a pianostring has, for instance, been watched when struck by the hammer, and so has the following decay of the string vibration. An elliptical transversal movement of the string was observed, Fig. 4.

To avoid disturbances at the input of the stroboscope caused by harmonics (or background noise) a one octave was connected between the microphone and the stroboscope.

Had a filter not been used the zero crossings originating from the second harmonic would in this case have caused false triggering.

The movement of a violin string and the surface of the violins body can be examined in the way above. When the amplitudes are small, however, a microscope must be used to allow the movements to be studied.

Experiments with organ-pipes in which smoke is mixed with the emerging air can give an indication of edge-tone effects. Here again the sound synchronises the stroboscope triggering via a microphone and a filter.

In the field of medicine some applications of stroboscopy worth mentioning are laryngoscopy and studies of the operation of the internal parts of the human ear. As an actual example a complex equipment currently used on a hospital in Copenhagen is shown in Fig. 5.

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It consists of a microphone, a frequency analyser and a stroboscope. For each patient the professor determine the typical speech spectrum, both before and after the treatment. Furthermore, it is of very great interest to combine these frequency determinations with the visual observation of the functioning of the vocal cords. For men the relevant frequency range is 63 - 200 cps and for women 200 - 630 cps. The most important diagnostic possibilities offered by the above arrangement are:

1. Early determination of Larynx-cancer.
2. Detection of tumours on the vocal cords.
3. Description of the laryngeal function after lesions of the recurrent nerve.

Also in the zoological field stroboscopy may be used to study wing movements and other periodic phenomena like the production of sound by grasshoppers etc.

**Conclusion**

There are numerous other problems in research as well as in education where the principles of stroboscopy can be advantageously applied. It has unfortunately only been possible within the frame of this paper to mention but a few of the possibilities that this technique offers, but it is deemed that the new principles in the design of stroboscopes, will make stroboscopy a very important tool in the science and technology of the future.
Accelerometer Configurations
by Gunnar Rasmussen and Jens August Jensen
given by: Jens August Jensen
eng. A/S Brüel & Kjaer, Nærum, Denmark

Recent years the piezoelectric type of accelerometer has become the preferred type of transducer for measurement of shock and vibration. The piezoelectric accelerometer has several advantages over other types of transducers, such as f.ex. velocity and strain gage type pick-ups. Some of the most obvious advantages are small weight and size, they are self generating and have high reliability. There are, however, various ways to design these accelerometers. Some of them present good frequency response, some have high sensitivity and other are practically uninfluenced by extraneous effects such as temperature, magnetic fields etc.

Most accelerometers are made as a compromise trying to fulfill a specific requirement compromising on some of the other.

The purpose of this paper is to give a survey of the most normally used accelerometer designs showing their main advantages, and to try to give a method of rating an accelerometer in order to evaluate its quality as a general purpose accelerometer.

If we list the desirable criteria for accelerometers it would appear as follows: Table 1.

| 1. High electromechanical conversion efficiency | 4. high stability |
| 2. wide frequency range | 5. low weight |
| 3. large dynamic range | 6. low cross-over sensitivity |

and low sensitivity to extraneous environmental effects such as:

Temperature - humidity - magnetic and acoustic fields - torque - base strain.

If we now look at the most often used designs, these can be classified in the following major classes:

| Table II |
|---|---|
| 1. Twister design (early type) | 5. basic compression type |
| 2. cantilever beam | 6. Isolated compression type |
| 3. Mushroom design | 7. single ended compression type |
| 4. shear design | 8. inverted single ended compression |

All the above types can be used with various types of ceramics.
Accelerometer Configurations

Some of the characteristics of an accelerometer do not only depend on the design, but also to a large degree on the piezoelectric material used. This we shall revert to at a later time.

First we shall look at the characteristics that can be obtained by the various basic designs.

The output voltage \( V \) from a piezoelectric bender is proportional to the deflection it is forced into. Depending on the configuration used we have \( V = K \frac{M a f^2}{E I} \) where \( K \) is a constant depending on the material used, the clamping Fixed-Free; Fixed-Fixed etc. \( M \) is mass of element and eventual applied extra mass, \( f \) acceleration \( l \) free length \( E \) modulus of elasticity and \( I \) moment of inertia of the beam. In fig. 1 is typical curves and sensitivity for a Fixed-Free design shown, and in fig. 2 typical data for a Hinged-Hinged design. As seen the sensitivity of these designs are rather low for the available frequency range.

![Fig. 1](image1)

![Fig. 2](image2)

The mushroom design may perform better than the cantitiver beam types in respect to transverse sensitivity, but it is in general not very effective either. The shear design is very good when low transverse sensitivity is an important goal. Only very few good designs exist for use at higher temperatures. The shear design is utilizing the piezoelectric material in shear deformation. The limitation in these types is in weights or in high temperature performance.

The most effective design is the compression. The output voltage in compression is \( V = K \frac{M a f}{E A} \) where \( t \) is the thickness of the piezoelectric material and \( A \) the area in compression. A more representative figure is the charge sensitivity which in general should be used comparing to the effectiveness of different designs \( Q = K \frac{M a f}{E} \).

In the basic design of compression type the outside housing was used as an effective spring, see fig. 3. This was later improved by using a spring and obtaining the isolated compression. In order to reduce the sensitivity to external acoustic pressure variations and temperature shock it has thus been further developed into compliant rod types, single ended compression types and inverted.
single ended compression types. See fig. 3.

Fig. 3.

1) Bender design
2) Basic Compression
3) Single ended compression
4) Inverted single ended compression
5) Cantilever beam
6) Mushroom design
7) Shear design

It is obvious from the foregoing that it is difficult to judge about the quality at an accelerometer.

For a certain application one may of course look only on the qualification needed for this particular application, but if one wish to select the best type to its price it is difficult. Therefore an attempt is made to propose a formula which takes the most important factor influencing general use into consideration, and try to rate different designs accordingly. This is may be so much more important as also several different piezoelectric materials are available. A table comparing some of the materials available is given in fig. 4.
### Table IV

<table>
<thead>
<tr>
<th>Material</th>
<th>Efficiency</th>
<th>Dielectric Constant</th>
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<th>Max Pressure $\text{N/m}^2$</th>
<th>Max Temp. $\text{°C}$</th>
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<td>Rochelle Salt</td>
<td>80</td>
<td>350</td>
<td>$20 \times 10^4$</td>
<td>70</td>
<td>55</td>
</tr>
<tr>
<td>ADF</td>
<td>40</td>
<td>15</td>
<td>$20 \times 10^4$</td>
<td>210</td>
<td>130</td>
</tr>
<tr>
<td>Barium Titanate</td>
<td>20</td>
<td>1000</td>
<td>$120 \times 10^4$</td>
<td>350</td>
<td>120</td>
</tr>
<tr>
<td>PZT</td>
<td>40</td>
<td>1500</td>
<td>$80 \times 10^4$</td>
<td>700</td>
<td>300</td>
</tr>
<tr>
<td>$\text{PbNb}_2\text{O}_6$</td>
<td>18</td>
<td>250</td>
<td>$30 \times 10^4$</td>
<td>350</td>
<td>450</td>
</tr>
<tr>
<td>Quartz</td>
<td>1</td>
<td>4</td>
<td>$80 \times 10^4$</td>
<td>3500</td>
<td>300</td>
</tr>
<tr>
<td>$\text{LiNb}_5\text{O}_3$, Y-cut</td>
<td>6</td>
<td>80</td>
<td>$80 \times 10^4$</td>
<td>$\sim$</td>
<td>1250</td>
</tr>
</tbody>
</table>

**Fig. 4.**

We should also like to include a stability factor in this scheme. It is, however, not possible to give exact figures for several of the materials. Typical figures for P.Z.T. as used by B&K are less than $0.2\%$ drift per decade of time at room temperature, first decade being one year after artificial aging. For Quartz the figures are much lower as the Quartz in itself is a very old material. The stability for Quartz transducers is determined by the mechanical stability and the ability to use the very low efficiency resulting in very high impedance levels.

Considering the number of factors involved in judging a general purpose accelerometer it is rather difficult to compare different designs. If, however, we look at the different design variables, we may judge about the efficiency by using following reasoning in trying to establish a "quality factor". The sensitivity $S$ in $\text{pC/g}$ is proportional to the mass acting on the sensitive element. The resonance frequency $f_{\text{res}}$ is inversely proportional to the root of the mass. We may thus set up a quality factor: $Q$

$$Q = \frac{S \cdot \sqrt{f_{\text{res}}}}{W} - \left( \frac{Q_1 \cdot t}{100} + \frac{Q_2 \cdot \nu^2}{100} \right)$$

where

- $W$ = Accelerometer weight in grammes
- $D$ = Useful dynamic range in decades going from 1 dB lower limiting frequency determined by accelerometer leak $R$ and capacitance of the unloaded accelerometer, to 1/3 the resonance frequency
- $f_{\text{res}}$ = Resonance frequency mounted on a 180 g steel block
- $t$ = Transverse sensitivity
- $\nu$ = Long term stability

Using this formulae on different designs we will find high quality modem accelerometers with a $Q$ of 500–1000 and spreading out for less good designs or special types down to 10.

Other factors are important. The base strain sensitivity should in general be below $0.002$ $\text{g/\mu in/in}$ in a typical example of very low base strain sensitivity is Brüel & Kjær 4338 with a $Q$ of 600 and base strain sensitivity of less than $0.00002$ $\text{g/s/\mu in/in}$.

Also a response to thermal shock which is equivalent to less than $0.05$ $\text{g}$ for a step function temperature change of $10^\circ\text{C}$ measured across 1000 Mohm resistance is typical for high quality accelerometers.

Cable movements of 5 mm from side to side right outside the cable plug may be used for evaluating cable sensitivity. A good design will respond with an output which is equivalent to less than $0.02$ $\text{g}$ and figures as low as $0.007$ may be obtained.
Vibrometer with Rotional Feedback

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Introduction

It is well known that the vibrometer is one of the most useful acoustic measuring devices. The measuring ranges of the vibrometer are decided by the constants of diaphragm in it. In order to extend the measuring and frequency ranges of it, without replacing the vibration members, we have developed the idea of applying MFB (Rotional Feedback) to it.

By using the method of MFB, the apparent mechanical impedance of vibrometer can be controlled merely by means of electrical operations.

In this paper, we are going to describe briefly the relations between MFB and mechanical impedance, and then show a few results of our experiments from which we found that our new method is considerably useful for the vibrometer.

The relations between MFB and Mechanical Impedance of Vibrometer.

In the present case, for example, we take a condenser type vibrometer. This type may be shown as Fig. 1,

where, $I_{o1}$, $I_{o2}$ : current source, but $I_{o2} = 0$;
$Y_{o1}$, $Y_{o2}$ : current source admittance;
$Y_1$, $Y_2$ : admittance of each transducer.
Vibrometer with hodional Feedback

$B_1$, $B_2$: force-factor of transducer.
$E_1$, $E_2$: voltage of transducer.

Now, the basic formulae of vibrometer are given as follows:

\[
\begin{align*}
\ddot{z} &= B_1 \ddot{z}_1 + B_2 \ddot{z}_2 \\
I_{of} &= (Y_{o1} + Y_1) \dot{z}_1 + \dot{z}_1 \\
0 &= (Y_{o2} + Y_2) \dot{z}_2 + B_2 \ddot{z}_2
\end{align*}
\]

$\ddot{z}$: velocity of diaphragm.
$\dot{z} = z + j\omega m + \frac{\ddot{z}}{j\omega}$: mechanical impedance of diaphragm.

From these formulae, when we drive the diaphragm from transducer 1, the ratio of output current $I_2$ and driving current $I_{of}$ is shown as follows:

\[
G(j\omega) = \frac{I_2}{I_{of}} = \frac{B_1 B_2 Y_{o2}}{(Y_{o1} + Y_1)(Y_{o2} + Y_2)} \left( \frac{B_1^2}{Y_{o1} + Y_1} + \frac{B_2^2}{Y_{o2} + Y_2} \right)
\]

also, in any other types of transducers, the transfer function can be given in the same form as Eq.(2).

Now, we apply feedback, which is proportional to the amount of vibration, to the case of Fig.1. It is shown in Fig.2, and the transfer function of this case is got as the following formulae:

\[
W(j\omega) = \frac{I_2}{I_{of}} = \frac{-B_1 B_2 Y_{o2}}{(Y_{o1} + Y_1)(Y_{o2} + Y_2)} \left( \frac{B_1^2}{Y_{o1} + Y_1} + \frac{B_2^2}{Y_{o2} + Y_2} \right) \{1 + G(j\omega)\beta\}
\]

$\beta$: the transfer function of feedback path.

Here, we can find that the mechanical impedance of diaphragm, including the motional mechanical impedance, is $\{1 + G(j\omega)\beta\}$ times the mechanical impedance without feedback loop.

Hence we find that the mechanical impedance of diaphragm becomes larger when negative feedback is given, and that it becomes...
Vibrometer with Rotational Feedback

smaller when positive feedback is given. If we neglect the admittance \( Y_1, Y_2 \), the denominator of Eq. (3) may be expressed as follows:

\[
\frac{1}{Z_0} + \frac{B_z^2}{c_2} + \frac{1}{j\omega} \left( 1 + \frac{B_z^2}{c_2} - \beta \frac{B_z Y_2}{c_1 Y_1} \right)
\]

\( C_1, C_2 \) : Capacitance of each transducer.

Then, the mechanical impedance of diaphragm varies according to the characteristics of transfer function \( \beta \).

the mutual relations among \( \beta, \) return difference \( \alpha, \) fundamental resonance frequency \( \omega_0 \), and the shape of resonance curve \( Q_0 \) are displayed as Table 1.

<table>
<thead>
<tr>
<th>Transfer Function of Feedback 1st</th>
<th>( \pm \beta' \text{(Const.)} )</th>
<th>( \pm j\omega \beta' )</th>
<th>( \pm (j\omega)^2 \beta' )</th>
</tr>
</thead>
<tbody>
<tr>
<td>Return Difference</td>
<td>( \frac{B_1 B_2 Y_{g2}}{C_1 Y_{01}} )</td>
<td>( 1 \pm \beta' \frac{B_1 B_2 Y_{g2}}{C_1 Y_{01}} )</td>
<td>( 1 \pm \beta' \frac{B_1 B_2 Y_{g2}}{C_1 Y_{01}} )</td>
</tr>
<tr>
<td>Type of Feedback</td>
<td>Displacement</td>
<td>Velocity Type</td>
<td>Acceleration</td>
</tr>
<tr>
<td>Resonance Frequency</td>
<td>( \alpha \omega_0 )</td>
<td>( \omega_0 )</td>
<td>( \frac{\omega_0}{\sqrt{\alpha}} )</td>
</tr>
<tr>
<td>Resonance Curve ( Q_0 )</td>
<td>( \sqrt{\alpha} \omega_0 )</td>
<td>( \alpha Q_0 )</td>
<td>( \sqrt{\alpha} Q_0 )</td>
</tr>
</tbody>
</table>

Table 1.

\( \omega_{01} = \sqrt{\frac{S + B_z^2}{m}} \), \( Q_{00} = \frac{\omega_{01} m}{r + \frac{B_z^2}{2}} \).

the experiments of MFB for Vibrometer.

Blockdiagram illustrating the application of single MFB to the condenser type vibrometer is shown in Fig. 3. And the frequency characteristics are shown as the solid line in Fig. 4. In this case, we could shift the resonance frequency about twice as high as the original.
Vibrometer with Notional Feedback

one.

But the peaks of resonance curves are higher than we estimated, as frequency increases. This phenomenon is caused by the fact that the loop gain involving all the devices was not a perfect second-order system.

From the root locus methods, the loop gain including vibrometer must be a second-order system.

Then, the double feedback, which is proportional to the displacement and velocity of vibration at the same time, is able to control the peaks of resonance curves. In consequence, the frequency response curves are given as the dotted lines in Fig. 4.

Conclusion.

1. We could find that the apparent constants of the vibrometer (r, m, s) can be controlled independently by only external electric devices.

2. The results of single AFB proportioning displacement were that the apparent stiffness of vibrometer could be decreased about to $\frac{1}{5}$, when negative AFB was given, and that it could be increased about to 5 times when positive AFB was given.

3. The result got from double AFB, proportioning displacement and velocity, was that the measuring frequency range could be expanded about twice as large as the original case.

References.


Normal Function Driven Electrostatic Loudspeaker with Motional Feedback

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1. Introduction

An ideal sound reproducing system has been required not only as high-fidelity reproduction but also as a standard sound reproducing system in the improvement of conventional loudspeakers using compared hearing tests or another psychological techniques. This report is a try to approach to an ideal system which has a uniform response on the front axis over audible frequency range.

In this paper, a new method of the distribution of driving force that is proportional to normal function of the radiator is discussed. This method is derived theoretically that the vibrating mode can be controlled to desired one, if the applied driving force across the vibrating member is distributed under the condition governed by the characteristics of the vibrating member, and some concrete methods to embody are described with some preliminary experiments.

2. Fundamental Consideration

The irregularity of the response of a loudspeaker vs. frequency is composed mainly by change of vibrating mode of radiator corresponding to driving frequency. Therefore, it is necessary for realizing an ideal system to make the vibrating mode of radiator single over whole frequency range. One conventional method for this purpose is to make a rigid vibrating member which is suspended by a pure stiffness element.

Furthermore, when the motional feedback that is composed by detecting the vibration of radiator and feeding back to the driving amplifier is adopted, the vibration of any point must correspond to sound output. This is another reason why
Normal Function Driven Electrostatic Loudspeaker with Motional Feedback

vibrating mode must be unity.

The relation between the distribution of driving force and the forced vibrating mode is expressed as is well known.

Displacement of the vibrating radiator $\xi(r)$, that is the forced vibrating mode, is expanded by the series of the natural vibrating mode, that is the normal function, $\zeta_m(r)$ and the corresponding resonant frequencies $\omega_m$ with the coefficient series $A_m$ because of the orthogonality of the normal function.

$$\xi(r) = \sum A_m \zeta_m r \frac{1}{\omega^2 - \omega_m^2}$$  \hspace{1cm} (1)

The driving force $F(r)$ is also expanded by the normal function with the coefficient series $B_m$.

$$F(r) = \sum B_m \zeta_m$$  \hspace{1cm} (2)

The coefficient series $B_m$ is determined by a simple integration and the forced vibrating mode may be expressed as equation (3)

$$\xi(r) = \sum C_m \frac{1}{\omega^2 - \omega_m^2} \int F(r) \zeta_m(r) r dr$$  \hspace{1cm} (3)

When the driving force is distributed proportionally to the normal function $\zeta_m(r)$, the forced vibration may be regarded as a single resonant system.

3. Concrete Method

In the dynamic loudspeaker, the driving force is concentrated to a circle around the center of the diaphragm, paying no attention to the characteristics of the diaphragm, and this fact is unavoidable because of the electro-mechanical transducing mechanism.

Contrary to the dynamic loudspeaker, the electrostatic loudspeaker has a convenience to embody the driving method described above. Namely, the driving force of the electrostatic loudspeaker having a conductive diaphragm is simply determined by the distance and the voltage difference between the diaphragm and the facing driving electrode as equation (4) per unit area,

$$F = \frac{\varepsilon_0 E \cdot e}{d^2}$$  \hspace{1cm} (4)

Where $\varepsilon_0$, $E$, $e$, and $d$ stand for the dielectric constant, polarizing voltage, signal voltage and the distance between the diaphragm and the driving electrode respectively.

It is obvious that these values are able to distribute along the surface of the diaphragm by some constructional modification.

— D - 90 —
From equation (2) and (4), the driving force must be distributed as equation (5),

\[ F(r) = D_m \sum_m(r) \]  

(5)

where \( D_m \) is a constant and therefore

\[ \frac{\varepsilon_o(r) \cdot E(r) \cdot E(r)}{d^2(r)} = D_m \sum_m(r) \]  

(6)

must be satisfied. In equation (6), \( r \) means the function of the position.

To make the explanation simple, a uniform circular stretched membrane is adopted as the vibrating radiator here. In this case, normal function \( \sum_m(r) \) is expressed by the Bessel function \( J_0(\alpha_m \frac{r}{a}) \) where \( a \) stands for the radius of the membrane and \( \alpha_m \) represents the series of the roots that \( J_0(\alpha) \) is equal to zero.

One method to embody the theory in this case is to adopt the driving electrode divided coaxially, insulated each other and fed the driving voltage individually as shown in Fig. (1).

In other words, the equation (6) was approximated in the manner like a stairway. Some another methods are available, for example, the effective area of the driving electrode can be varied.

4. Some Preliminary Experiments

In order to confirm the method described above, a simplified model was tested by a strip as shown in Fig. (2).

The displacement of the strip was observed by detecting electrostatically when the signal current flowed the strip in the distributed magnetic fluxes. The magnetic flux was controlled sectionally by varying electro-magnets along the strip.

Fig. (3) shows the displacement of the center of the strip where the solid curve and the dotted curve are the response to uniform driving and distributed sinusoidally.

It was recognized that the higher mode vibrations decreased at a large amount when the magnetic field which is proportional to the driving force is distributed as a sinusoidal curve, that is the normal function of a string, compared with uniform distribution.

In the next, a circular electrostatic loudspeaker which has the driving electrode divided to 2 coaxial conductors as shown in Fig. (4). The sound output on the front axis was observed when the outer electrode was feeded zero voltage. This is the lowest approximation of distribution driving.

The frequency characteristics of sound output is shown in Fig. (5). It may
be consented that the response in high frequency shows quite smooth characteristics. The peak of fundamental resonance can be cancelled by motional feedback that is not illustrated in this paper.

5. Conclusion

From these results, it may be conjectured that one of the way to approach to the ideal sound reproducing system is practical. That is (1) the radiator is driven by distributed force that is proportional to the normal function of the radiator electrostatically and (2) The membrane driven by means of item (1) vibrates at the same phase over all frequencies as a single resonant system and the other higher modes vanish. (3) This fact is confirmed experimentally by a simplified model and by the first order approximation (4) the vibration must be detected and feeded back to the driving amplifier especially in low frequency.

However, there is a great deal of problems to be solved, for examples, the most effective approximation of the desired distribution of the driving force, or the effect of the radiation impedance across the radiator.

Experiments for divided driving electrode of 3 or 4 parts are going to be carried on.
Electrostatic loudspeaker with acoustical transformation and horn

Josef Merhaut
Professor of electroacoustics of the Prague Technical University

Introduction

The known electrostatic loudspeakers have usually fairly flat frequency response and a good transient response. However the low frequency limit of such a speaker is limited. There will be shown, that an essential improvement of this parameter is possible, when an acoustical transformation and a horn is used.

The effect of acoustical transformation

The radiation mechanical impedance at the input of an exponential horn is above the critical frequency approximately resistive and approaches the real value

\[ r_r = c_0 \rho S_1 \]  \hspace{1cm} (1)

if \( c_0 \) is the propagation velocity of the sound, \( \rho \) the density of medium and \( S_1 \) the throat area of the horn. If \( S_1 \) is made \( k \) times smaller then the area of diaphragm \( S_d \) the radiation mechanical resistance \( r_{rd} \) at the diaphragm is \( k^2 \) larger then \( r_r \), due to the acoustical transformation and is

\[ r_{rd} = c_0 \rho S_1 \left( \frac{S_d}{S_1} \right)^2 = c_0 \rho S_d k \]  \hspace{1cm} (2)
The low-frequency limit

The low-frequency limit is given by the equation

$$\frac{1}{\omega_t c_d} = r_{rd}$$  (3)

where $\omega_t = 2\pi f_t$, if $f_t$ is the frequency, where 3 dB decay in the
frequency response occurs, and $c_d$ denotes the mechanical compliance of
the stretched diaphragm. This compliance depends upon the mechanical
tension of the diaphragm in Newtons per unit length $\nu$. If the diaphragm
is rectangular, and stretched in one direction only the compliance is
given by

$$c_d = \frac{\ell}{12 \nu b}$$  (4)

if $\ell$ is length and $b$ the width.

From eqs. (2), (3) and (4) we get for $\omega_t$

$$\omega_t = \frac{12 \nu}{c_0 \rho \ell^2 k}$$  (5)

From eq. (5) is apparent, that for the transformation ratio $k > 1$ the
low-frequency limit gets lower. If for instance the $f_t$ without transfor-
mation would be 2000 Hz and $k = 10$, we get the low-frequency cut-off of
200 Hz, when using the horn.

The tension $\nu$ must be due to the mechanical stability of the sys-
tem larger, then the tension $\nu_e$ caused by the electrostatical attrac-
tive forces. If we introduce the safety coefficient $\beta > 1$, there must
be $\nu = \beta \nu_e$. As $\nu_e$ is given for a push-pull system by the for-

mula

$$\nu_e = \frac{E_0^2 \ell^2}{8 d}$$  (6)

if $E_0$ is the permittivity of the air, $E_0$ the electrical field intensi-
ty and $d$ the distance between the electrodes, we may using (6) and (5) write
Electrostatic loudspeaker with acoustical transformation and horn

\[
\omega_t = 1.5 \frac{\beta \varepsilon_0 E_0^2}{c_0 \rho} d k
\]  

(7)

The high-frequency limit

The upper 3 dB-decay cut-off is given by the equation

\[
\omega_h m_e = r_{rd}
\]

where \( m_e = \sigma b \ell \) is the mass of the diaphragm and \( \sigma \) its specific mass per unit area.

From (8) and (2) we get

\[
\omega_h = \frac{c_0 \rho k}{\sigma}
\]

(9)

We may see, that the coefficient \( k \) increases the high-frequency limit and improves also the upper part of the frequency response.

The design of the new loudspeaker

The author has designed an electrostatic loudspeaker with a horn, using \( k = 10 \). The body of one of the electrodes is segmented, forming the slits running to the throat of the exponential horn. The length \( \ell = 1.5 \text{ cm} \), separation between the diaphragm and back electrode \( d = 0.25 \text{ mm} \), and \( E_0 = 4 \text{ kV/mm} \). For the given \( d \), the polarising d.c. voltage results in \( U_0 = 1.0 \text{ kV} \). Tension \( \nu \) was chosen \( \nu = 85 \text{ N/m} \) and for safety coefficient \( \beta = 5.3 \) the low-frequency limit was 196 Hz. On fig. 1. we see the cross-section of this speaker.

The unit has very flat frequency response, which corresponds fairly with the theory. The frequency response measured as the pressure at the throat of the horn by means of a 1/2 " measuring microphone is shown on fig. 2. System has been patented as the invention of the author.
The directivity index of studio monitoring loudspeaker equipments and experiments for defining the subjective evaluation parameter "presence"

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Introduction

For the time being it is not possible to establish definite relations between physical quantities (frequency response, directional characteristic, transient response etc.) and the corresponding subjective parameters (sound balance, presence, nonlinear distortions etc.) of loudspeaker equipments. Therefore, the subjective evaluation of the quality of studio monitoring loudspeakers by listening groups is of more importance for decision on quality questions than the results of objective measurements.

It has been found that the subjective assessment of high quality studio equipments can be restricted in the main to the determination of the balance between lower, medium and higher frequency ranges, as well as at the co-ordination of transmission curve (frequency response) and directional characteristic. This co-ordination influences the reflected-to-direct sound ratio (R/D ratio) in the different frequency ranges at the listeners place in the used reproduction room. It can be assumed, that the R/D ratio is in connection with the so-called "presence" which is to be intensively investigated.

The subjective term "presence" characterizes the impression of "stepping forward of a part of the sound source out of the sound picture" during the listening to a sound reproduction. It was further ascertained that sound pictures of different types of monitoring loudspeakers can be made similar by equalizing their sound level curves, however, a different presence impression remains if their directivity indices do not agree. Therefore, the general demand for a linear transmission curve cannot be valid any more. It was assumed that the subjective term "presence" could be interpreted by a certain co-ordination of the transmission curve and the directional characteristic of the loudspeaker equipment. For this reason the directivity indices of some loudspeakers with differently evaluated presence effects are firstly determined.

Definition of the directivity index

a) The directivity index can be calculated \[ d \] by means of the directional characteristics recorded in an anechoic room as follows:

\[
d = 10 \log_{10} \log \frac{\pi \Delta \varphi}{\sum_{n=1}^{2} \left( \frac{P(\varphi)}{P_{\text{ax}}} \right)^2 \cdot \sin \varphi_n \cdot \Delta \varphi}
\]
Directivity index of studio loudspeakers

with $\Delta \varphi = 10^\circ$; $p(\varphi)$ = sound pressure level at the point $F_{ax}$ = axial sound pressure level

b) On the basis of measurements both in an anechoic and reverberant room the directivity index of monitoring loudspeakers can also be defined as [2]:

$$d = 10 \log \frac{L_{RT} - L_{KL}}{I_{RT}} \text{dB}$$

with $L_{RT} = \left[ I_{FR} + 10 \log \frac{S}{m^2} - 6 \right] \text{dB}$ (sound power level measured in reverberation room)

and with $L_{KL} = \left[ I_{FK} + 11 \right] \text{dB}$ (equivalent freefield axial sound pressure level)

The result can be determined in simple manner by measuring the sound level difference with a ruler.

Fig. 1 shows the comparison of both the methods applied to the monitoring loudspeaker equipment Z 131-1.

The agreement is sufficient for the intended purpose. The criterion for the accuracy of the measurement with method b is the limit frequency of the reverberant chamber. In the other case with method a it is the measuring distance. Besides both the methods possess systematic errors depending on the accuracy of reverberation time measurements and on reading uncertainties in evaluating the curves. However, both the methods provide a similarly correct tendency of the directivity index. Corresponding to the given circumstances it must be occasionally decided which method should be used for the determination of the directivity index.

Fig. 1 Monitoring Equipment Z 131
Determination of Directivity Index
a) - computed according to method a)
b) - determined according to method b)
Directivity index of studio loudspeakers

\[ \alpha = 10 \log \frac{\gamma}{(\alpha B)} \]

Fig. 2 Monitoring Equipments -2 and -3 compared with Z 131-1
Comparison of Directivity Indices

Fig. 2 shows directivity indices of three Z 131 equipments as computed from the data of method a. Fig. 3 shows the directivity index of an usual loudspeaker related to Z 131 and measured with method b.

Fig. 3 Comparison of the Directivity Index of a 50-l-Box to the Reference-Loudspeaker Z 131-1 Directivity Index

---D-99---
Directivity index of studio loudspeakers

Subjective investigations

By means of well-known subjective methods [3] nine studio monitoring equipments (type Z 131) were examined. Though the transmission curves of them have only small differences the test results showed, that the equipments were different in transparency, sound balance, presence-effect, timbre, and distribution depth of sound sources. Among samples of the type Z 131 the device -1 was found to be the mostly balanced one, the mentioned subjective parameters were estimated to be better with that model than with other equipments. So Z 131-1 was nominated as reference standard.

It is very interesting to compare the directivity index response of Z 131-1 with that of the loudspeakers Z 131-2 and Z 131-3, which were judged more unfavourably with regard to their presence-effect than the reference device (Fig. 2), but had the same transmission curve. The deviations in the 1000 - 2000 Hz band are typical evident.

Subjective tests on ordinary type loudspeakers which don't get usually as high quality as studio equipments, results a very annoying "bundling - effect" (directivity effect) i. e. a too high presence factor (Fig. 3).

Conclusions

1. Transmission curves of studio monitoring loudspeaker equipments show such small differences that it is not possible to find definite explanations for the subjectively perceived deviations.

2. It seems to be possible to use the directivity index for interpreting the subjective parameter "presence". Fig. 2 shows the distinction of the directivity index with the equipments -2 and -3 compared with reference equipments -1 in their frequency ranges 100 Hz ... 500 Hz as well as 1000 Hz ... 3000 Hz, which was perceived in subjective tests as a higher presence effect, in good agreement with the measurements.

It is obviously desirable for that studio equipment to have a decrease of the directivity index at 2000 Hz related to 1000 Hz and 5000 Hz respectively. But it must be emphasized that the audible differences are very small.

The subjectively perceived discrepancy between studio monitoring equipments and ordinary loudspeakers becomes obvious with the directivity index response. Fig. 3 shows the directivity index of a small 60-liter-box, the very high presence effect of which was objected to, in comparison with the standard type Z 131-1.

Some other questions arise necessarily as well, e. g. the searching for numerical relationships between directivity index and presence effect to confirm objectively such judgements as "good" or "unsufficient presence". This paper can therefore only be regarded as the first part of further extensive studies.

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Technische Mitteilungen RFE 2 (1959), Heft 2, S. 27-30, 36
Introduction

As the loudspeaker has complicated distortion characteristics, data of conventional distortion measurements are not appropriate to predict distortion produced from a loudspeaker in its actual performance. In this paper, a method of distortion measurement using programme sound as test signal is proposed and the results of this method applied to several loudspeakers and their relation with aural judgement are described.

1. Method of Measurements

With band-elimination and band-pass filters, 1/2 octave-band distortion components were separated out from the test signal and measured by square-integrating circuits as shown in Fig.1. The test signal is filtered, and the outputs are presented in the spectrum shown in the diagram. The band-pass filters allow the relevant frequency components to be isolated, and the square-law integrator accumulates the energy at these frequencies.

Fig.1. Block Diagram of the Apparatus.
Fig. 7. 1/2 Octave-Band Spectrum of Distortion Component Produced by Loudspeakers. (0 dB = Overall Level of Original Signal) 

--- original signal --- distortion component (--- subjectively detected) 

parameters are average nominal power applied to the loudspeaker.
On Distortion Measurement of Loudspeakers

Signals of about 10 sec. were chosen from the part of programmes where abrupt intensity change was not found. 1/2 octave-band level of distortion component averaged for duration of the signal was successively measured in the frequency range from 70 to 10,000 Hz. As the reduction of level caused by band-elimination is different for each frequency band, the input level of loudspeaker was adjusted to maintain constant input level.

2. Results of Measurements

Distortion characteristics of loudspeakers measured by this method are shown in Fig.2. For comparison, distortion of an amplifier is shown in Fig.3. From these measurements, following features of distortion of loudspeakers can be observed:

(1) Distortion around 1kHz is most dominant almost in every case.
(2) Distortion around 500 Hz is relatively more prominent for larger input power and for smaller loudspeakers. These components are presumably due to the large amplitude vibration of a cone against low frequency component of signal.
(3) Compared with amplifiers, loudspeakers produce much more distortion at low input level, but growth rate of their distortion is small.

For practical convenience, wide-band noise shaped to have spectrum similar to that of programme sound can be also used as test signal. In Fig.4, obtained results with noise and programme sound are shown for comparison. Similar results are obtained in both cases. Almost the same
tendency was observed in other loudspeakers. But for distortion of amplifiers, different results were obtained for different test signals.

3. Relation with Aural Judgements

To consider the observed results in connection with subjective tone quality, just perceptible level of each distortion component in the presence of the signal was measured: Male voice was used as test signal, and its distortion components in 1/2 octave-band were recorded. Then they are mixed with the original signals and presented to the subjects. Results are shown in Fig. 5, together with samples of measured distortion characteristics. The results suggest that it is possible to predict whether distortion of a loudspeaker in its actual use will be perceptible or not, by comparing the measured distortion level with the just perceptible level. Such a procedure will be useful in establishing criterion to decide acceptable input power of loudspeakers.

To apply the procedure to wider use, further experiments are being considered.
On Propagation of Sound in the Axially Symmetric Tube
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Introduction

The precise analysis of acoustical transmission systems based on the wave theory, in spite of their complexity and difficulty, gives fundamental informations on the designing of acoustic devices and their elements. There are many important acoustical transmission systems with axial symmetry, and one-dimensional wave equation has been widely used to analyze them. This equation is, however, only applicable in a low-frequency range where the wave fronts depart least from planeness. Since the appropriate methods have not been developed yet, it has been used beyond the restriction and there appear various discrepancies between theoretical estimations and experimental results particularly at a high frequency side. Therefore it is desirable to establish a method which is widely applicable and accurate as well.

In this paper, as an attempt to develop a desired method, the axially symmetric tubes with rigid wall are divided into many sections whose radii vary stepwise and an accurate analysis of their acoustic properties is performed including higher order modes in each section. And finally the obtained results are compared with those based on the other approximations.
Theoretical Method and Procedure

The axially symmetric tube adopted in this analysis is shown in Fig. 1. If the cylindrical coordinate \((r, \theta, z)\) is introduced and the axially symmetric sound field with harmonic time dependence is developed by the linear combination of intrinsic modes of each tube, the expressions for the pressure, \(p_n(r, z)\), and the normal component of velocity, \(u_n(r, z)\), in \(n\)-th section are given by

\[
p_n(r, z) = \sum_{j=0}^{\infty} b_j \left( \frac{a_j}{r} \right)^{1/2} \sin \left( \frac{a_j}{r} \right) e^{-j(2-2m)/\gamma_n} (n=1, 2, \ldots, M)
\]

\[
u_n(r, z) = \sum_{j=0}^{\infty} \left( \frac{a_j}{r} \right)^{1/2} \sin \left( \frac{a_j}{r} \right) e^{-j(2-2m)/\gamma_n} (0 \leq r \leq r_n)
\]

where

\[
k = \omega/c, \quad i = -1, \quad \bar{a}_j = \frac{\omega}{c} b_j, \quad \bar{a}_j = \frac{\omega}{c} d_j, \quad \bar{a}_j = \frac{\omega}{c} e_j, \quad \bar{a}_j = \frac{\omega}{c} f_j
\]

\[
\bar{a}_j, \bar{a}_j \quad : \text{forward and backward component of} \ (0, j) \text{ mode in} \ n\text{-th section}
\]

\[
f(z) = \text{radius of tube along the axis,} \quad c = \text{velocity of sound}
\]

\[
\omega = \text{angular frequency}
\]

\[
p = \text{mass density}
\]

and the time factor \(e^{\omega t}\) is omitted for simplicity. We have, making use of the boundary conditions at the discontinuities, the linear difference equations whose solutions for input and output conditions give the relevant sound field, etc.

Example and the Numerical Results

As an example the difference equations corresponding to the axially symmetric tube with non-decreasing radius are considered.

\[
\left( \frac{e_n^{i} \delta^i}{\Delta z^i} + R^i \right) J_0^{r^i} = \sum_{j=0}^{\infty} \frac{a_j}{r} \left( \frac{a_j}{r} \right)^{1/2} \sin \left( \frac{a_j}{r} \right) e^{-j(2-2m)/\gamma_n} (n=2, 3, \ldots, M)
\]

\[
\left( \frac{e_n^{i} \delta^i}{\Delta z^i} - R^i \right) J_0^{r^i} = \sum_{j=0}^{\infty} \frac{a_j}{r} \left( \frac{a_j}{r} \right)^{1/2} \sin \left( \frac{a_j}{r} \right) e^{-j(2-2m)/\gamma_n} (n=1, 2, \ldots)
\]

where \(e_n^{i} \delta^i / \Delta z^i (\leq 1)\), \(e_n^{i} = e_n^{i} \delta^i / \Delta z^i\), \(\Delta z_n = z_{n+1} - z_n\)

\[
\left( \frac{1}{\pi} \right)^{1/2} \int_0^{r_n} \frac{1}{r} \frac{a_j}{r} \left( \frac{a_j}{r} \right)^{1/2} \sin \left( \frac{a_j}{r} \right) e^{-j(2-2m)/\gamma_n} (r) r dr = \begin{cases} 1.0 & (j=0) \\ \frac{2\pi \int_0^{r_n} \frac{a_j}{r} \left( \frac{a_j}{r} \right)^{1/2} \sin \left( \frac{a_j}{r} \right) e^{-j(2-2m)/\gamma_n} (r) r dr}{(\delta_j \gamma_n) \int_0^{r_n} \frac{a_j}{r} \left( \frac{a_j}{r} \right)^{1/2} \sin \left( \frac{a_j}{r} \right) e^{-j(2-2m)/\gamma_n} (r) r dr} & (\text{elsewhere}) \end{cases}
\]
On Propagation of Sound in the Axially Symmetric Tube

The solution of those equations can be easily obtained as a recurrence equation

\[ P_S = \frac{e}{S_0} \frac{5n}{m} \frac{5s}{m} \left( \frac{T_{n-1}}{S_0} \right) \left( \frac{T_{s+1}}{S_0} \right) \]

\[ (n=2,3,\ldots,M) \]

\[ (s=0,1,2,\ldots) \]

provided that the radius of tube varies gradually enough and the backward wave components are negligible. In usual case, however, the backward wave is intimate associated with the forward one and cannot be neglected. Thus, we used the computer (HITAC 5020) in order to solve the difference equations accurately. A few examples of the pressure distribution along the axis based on the other approximations as well as the present one are illustrated in Fig. 2 where we have assumed that the incident wave is \((0,0)\) mode with amplitude of \(P_0 = 1\) and the tube radius varies as

\[ f(z) = \begin{cases} R_1 & (z < z_1 = 0) \\ R_0 e^{mz} & (0 < z < z_M = L) \\ R_M & (z > L) \end{cases} \]

where \(R_1, R_M, m, L\) are constants satisfying the relation \(R_M = R_0 e^{mL}\).

![Fig. 2 Pressure amplitude distribution along the axis](image)

- accurate solutions of difference equations
- neglecting the interference between discontinuities
- neglecting the backward components except \((0,0)\) mode
- one-dimensional equation

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On Propagation of Sound in the Axially Symmetric Tube

The conclusions which can be derived out of the results are summarized as follows.
1. When the tube radius varies gradually, the reflection of sound is small and so the variation of the field appears rather monotonous.
2. At a high frequency side above the cut-off frequency of (0,1) mode \( kR_1 > \lambda^2 / 3.832 \), the sound field is complicated and greatly different from that at a low frequency side.
3. The sound field based on one-dimensional equation may be used in a low frequency range \( kR_1 < 1 \).
4. The calculated field is useful up to \( kR_1 = 2 \), when the backward components except (0,0) mode are neglected and the tube is terminated with characteristic impedance \( R_c \) of (0,0) mode.
5. The field which neglects the interference between discontinuities gives rather good approximation on the whole.
6. The local quantity like the field depends on the methods of approximation more sensitively than the global quantities such as the transmission and reflection coefficient and input impedance do.

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Introduction

Many electro-acoustic transducers, such as telephone transmitters and artificial mouths, are used in near sound field. In order to design these transducers, the knowledge of a property of near sound field is important. Then, we consider an increase in sound pressure on the rigid sphere placed in near field of a spherical sound source.

General Outline

As it is difficult to solve the wave equation with respect to the case existing two spheres as shown in Fig. 1, we adopt an approximate method using superposition of scattering waves from each sphere. At the first step, assuming that the sphere II does not exist, we derive a radiation from the spherical sound source I. Next, this radiated wave is considered as an incident wave to the sphere II, and a scattering wave from the sphere II is expressed by the incident wave.

Fig. 1 The geometry of the system. The sphere I of a radius \(a_1\) vibrates with \(\nabla \cos \theta, e^{i\omega t}\) on \(0 \leq \theta, \leq \theta_s\), and is rigid on another part. The sphere II of a radius \(a_2\) is rigid. The centers are a distance \(a_{12}\) apart. A field point \(P\) has the coordinates \((r, \theta, \phi)\) with respect to sphere I and \((r_s, \theta, \phi)\) with respect to sphere II.
On the Diffraction Effect of Two Adjacent Spheres

In a similar way, we can obtain the scattering waves from each sphere to infinite repetition. Hence, we may express the velocity potential at an arbitrary point \( P \) as follows:

\[
\Phi_2 = \sum_{l=1}^{\infty} \left[ \Phi_{21}(r, \theta_1) + \Phi_{23}(r, \theta_2) \right],
\]

where \( \Phi_{21}, \Phi_{23} \) are velocity potentials of scattering waves from the sphere I and II, respectively, \( \gamma \) denotes the order of the scattering.

Consider the sphere with \( \partial \Phi / \partial r = -\rho \cos \theta e^{i \omega t} \), on 0 \( \leq \theta_1 \), \( \gamma \), and with \( \partial \Phi / \partial r = 0 \) on another part. The velocity potential \( \Phi_{23} \) is given by\(^1\)

\[
\Phi_{23} = \sum_{m} b_{23m} \frac{J_m(kr_1)}{kr_1} \cos \theta_1, \quad (2)
\]

\[
b_{23m} = \frac{2\pi i}{k^3} \int_{0}^{\infty} \frac{\varepsilon(\varepsilon^2 - 1)}{\varepsilon} \frac{P_m(\varepsilon \cos \theta_1)}{P_m(\varepsilon)} d\varepsilon, \quad (3)
\]

where \( k = \omega/c \), \( c \) denotes the velocity of sound, \( \alpha_1 = k a_1 \), \( k J_m(x) \) denotes the spherical Hankel function of the second kind, \( k J_m(x) \) denotes the Legendre function. Eq.\((2)\), which denotes the scattering wave from the sphere I, is supposed as an incident wave with respect to the sphere II. Hence, each individual term in Eq.\((2)\) requires an infinite series of zonal harmonics for its expression in terms of coordinates based on the center of sphere I. The \( m \)th term of Eq.\((2)\) is represented as follows:\(^2\), \(^3\)

\[
h_m^{(2)}(kr_1) P_m(\cos \theta_1) = \sum_{m=0}^{\infty} K_m \frac{J_m(kr_1)}{kr_1} P_m(\cos \theta_2), \quad (4)
\]

where \( j_m(\theta) \) denotes the spherical Bessel function. From the relation between sphere I and II as shown in Fig.1, we obtain the relation

\[
\frac{1}{kr_1} \frac{d}{d(r_1)} = \frac{1}{kr_2} \frac{d}{d(r_2)}, \quad (5)
\]

where \( \mu_1(\cos \theta_1) \), and \( \alpha_1 = k a_1 \). Performing this operation on Eq.\((4)\) \( m \) times, and letting \( r_2 = 0 \) and \( \cos \theta_2 = 1 \), after differentiation \( kr_1 \) is to be replaced by \( ka_1 = \alpha_1 \), then coefficient \( K_m \) becomes

\[
K_m = A(n,m) \left[ \sum_{n=1}^{\infty} a_m^2 (a_m)^2 c_m \right]^2 \left[ \sum_{n=1}^{\infty} c_n (a_m)^2 \right], \quad (6)
\]

where \( c_m \) is the binomial coefficient. Hence, the velocity potential of sphere II with respect to the first scattering is

\[
\Phi_{23} = \Phi_{21} + \Phi_{23} = \sum_{m=0}^{\infty} \left[ d_{21m} j_m(kr_1) + b_{23m} h_m^{(2)}(kr_1) \right] P_m(\cos \theta_2), \quad (7)
\]

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On the Diffraction Effect of Two Adjacent Spheres

where

\[ a_{21m} = \sum_{n=0}^{\infty} b_{nN} A(n, m). \]

Let \( \partial \phi_2 / \partial r = 0 \) on the surface of sphere II, coefficient \( b_{21m} \) becomes

\[ b_{21m} = -a_{21m} i_m(d_i) / k^{\alpha_2}(d_i), \]

where \( d_i = k \theta_i \). Next, the wave given by the second terms of Eq.(7) is scattered by the sphere I, and the wave again goes back to the sphere II. In this case, the velocity potential on the sphere II with respect to the second scattering is given as follows;

\[ \phi_{22} = \phi_{21} + \phi_{22s} = \frac{e^{i\theta}}{k_2} \sum_{m=0}^{\infty} \frac{\omega_2}{\alpha_2^m} \left\{ \left[ -i_m(d_i) / k^{\alpha_2}(d_i) \right] \cdot \sum_{m=0}^{\infty} b_{21m} A(m, \ell) \right\} . \] (8)

In a similar way, we can obtain the velocity potential in case that the order of the scattering is more than two, where the boundary condition of the sphere I is assumed rigid since the second scattering.

Calculations and Measurements of Diffraction Coefficient

A diffraction coefficient \( D \) is defined as follows;

\[ D = 20 \log_{10} \left[ \sum_{m=0}^{\infty} \frac{\phi_{2m}}{\phi_2} \right] \] (dB),

\[ \phi_2 = \sum_{n=0}^{\infty} b_{nN} k^{\alpha_2}(k \theta_i) P_n(\cos \theta_i), \]

where \( r_1, \theta_1 \) denote the distance and the angle between center of the sphere I and a point on the sphere II, respectively. The diffraction coefficients deal with two cases that the distance between centers of two spheres is large and small.

The calculated values of \( D \) for the large separation are shown in Fig.2. Dotted line denotes the diffraction coefficient of the plane wave striking the sphere II. On comparison of solid line and dotted line, solid line turns out to be similar to dotted line in the high frequency region. Calculated

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Fig.2 Diffraction coefficient for large separation, where \( a_i = a_2 = 10 \text{cm}, \ a_{10} = 120 \text{cm}, \ \theta_0 = \sin^{-1}(0.25) = 14.30^\circ, \ \theta_2 = 0^\circ \). Dotted line is the diffraction coefficient of the plane wave.
values are verified with measured values as shown in Fig.2. Absolute values of \( \phi_{21} \) and \( \phi_{22} \) are shown in Fig.3, we conclude that the term of the second scattering of the sphere II can be neglected.

In Fig.4, the values of \( D \) for the small separation considering the second scattering are shown. Absolute values of \( \phi_{21} \) and \( \phi_{22} \) are shown in Fig.5, the term of the second scattering of the sphere II can not be neglected. Calculated values are approximately verified with measured values as shown in Fig.4.

Conclusion
Results of the study are applicable to design of telephone transmitters considering real-mouth field.

Reference

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Fig.3 Comparison of absolute values of \( \phi_{21} \) and \( \phi_{22} \), where the dimensions are same in Fig.2.

Fig.4 Diffraction coefficient for the small separation, where \( a_1=10 \text{cm}, a_2=3 \text{cm}, a_{12}=15 \text{cm}, \theta_1 = 5 \sin(0.25) = 14^\circ3^0', \theta_2=0^\circ \).

Fig.5 Comparison of absolute values of \( \phi_{21} \) and \( \phi_{22} \), where the dimensions are same in Fig.4.
On the Spherical Standard Sound Source
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Introduction

As the standard sound source for free field measurement, the spherical sound source seems to be most suitable. This sound source is constructed by the rigid spherical baffle and the piston diaphragm shaped as a part of the spherical baffle. One of the features of such a sound source is that the radiated sound field can be obtained analytically. In this paper, the detailed characteristics of the sound field produced by the spherical sound source are described. As an example of application of such a sound source, the measurement of the diffraction coefficient of the standard microphone is also mentioned.

Sound Field Produced by the Spherical Sound Source

The mathematical model of the spherical sound source is shown in Fig. 1, and the sound field produced by it is expressed using polar coordinate system (r, θ, φ). Now, it is assumed that the part of 0 ≤ φ ≤ φ₀ of the sphere of radius a vibrates as a piston with velocity of \( Ve^{i\omega t} \). The sound pressure produced by such a sound source is shown as follows:

\[
P = -i \rho c \frac{V}{2} \sum_{m=2}^{\infty} \frac{2m+1}{\mathcal{h}_m^{(2)}(ka)} \mathcal{h}_m^{(2)}(kr) P_m(\cos \theta) \int_{\cos \theta_1}^{\cos \theta_2} \frac{d}{d\mu} P_n(\mu) \, d\mu
\]

where
- \( \rho \): density of the medium
- \( k = \frac{\omega}{c} \): wave number
- \( \omega \): angular frequency
- \( c \): velocity of sound
- \( \mathcal{h}_m^{(2)}(x) \): spherical Hankel function of the second kind
- \( \mathcal{h}_m^{(2)'}(x) = \frac{d}{dx} \mathcal{h}_m^{(2)}(x) \)
- \( P_n(x) \): Legendre function
If we put the expression of the sound pressure as

$$P = i ho c V \sin^2 \phi_0 S(ka, kr, \theta, \phi_0)$$  \hspace{1cm} (2)$$
the function $$\frac{S(ka, kr, \theta, \phi_0)}{ka}$$ represents the sound pressure under the condition that the volume acceleration of the sound source is kept constant (mass-controlled). The function $$\frac{S(ka, kr, \theta, \phi_0)}{ka}$$ is the complex value, and it is expressed as follows:

$$\frac{S(ka, kr, \theta, \phi_0)}{ka} = \frac{1}{2ka \sin^2 \phi_0} \sum_{m=0}^{\infty} \frac{2m+1}{h_m^{(2)}(kr)} P_m^{(m)}(\cos \phi_0) P_m^{(m)}(\mu) d\mu$$

$$= R + i X$$

Fig. 1 Mathematical model of the spherical sound source.

So, the magnitude and the phase angle of the sound pressure is obtained by following calculation.

$$|\frac{S}{ka}| = \sqrt{R^2 + X^2} \hspace{1cm} \angle \frac{S}{ka} = \tan^{-1} \frac{X}{R}$$  \hspace{1cm} (4)$$

In equation (3), by putting $$\theta = 0$$ and take the values of $$\phi_0$$ and $$\frac{kr}{ka}$$ as parameters, the frequency response on the main axis under arbitrary condition can be obtained. An example of the calculated value is shown in Fig. 2.

In this case, $$\frac{kr}{ka}$$ is fixed to 5.
As the value of $$\phi_0$$ becomes larger, the variation on the response becomes larger in the higher frequency region. Such characteristics are maintained in higher value of $$\frac{kr}{ka}$$.

Fig. 3 shows the effect of the axial distance on the frequency response. In this case, the magnitude of the sound pressure $$|\frac{S}{ka}|$$ is multiplied by

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\[ \frac{r-a}{a} \] so as to cancel the decrement of the magnitude of the sound pressure due to the increment of the distance. By this operation, minute changes on the frequency characteristics can be determined quite easily.

If we take the variation of \( \theta \) into account, the details of the radiated sound field can be obtained. As an example of such cases, the sound pressure distribution is shown in Fig. 4. In this figure, the value of \( \mathcal{N}_0 \) is selected as \( \sin^{-1} \frac{1}{6} \approx 9.36^\circ \), which is the same value as the sound source used in the experiment. The solid lines show the calculated values and the points show the experimental values. Identification of the both values can easily be verified.

Spherical Sound Source\(^2\)

Actual construction of the spherical sound source is shown in Fig. 5. This is of electrodynamic type. Convex diaphragm is made of metal titanium and spherical baffle is made of brass. Vibrating system acts as a one-degree-of-freedom system in the frequency range of 100-10000Hz. In Fig. 6, frequency response on the main axis is shown. Difference between measured value and calculated value of the response is kept within 2dB.

\[ \mathcal{N}_2 = \sin^{-1} \frac{1}{6}, \quad ka = 5 \]

Fig. 4 Sound pressure distribution in the radiated sound field.
Application of the Sound Source to the Measurement of Microphone Sensitivity

Because of many features as above mentioned, the spherical sound source can be used as reliable standard sound source. As an application of the standard sound source, measurement of the sensitivity of the microphone is important. In Fig. 7, free field diffraction coefficient of the standard condenser microphone measured using the spherical sound source as the standard source is shown. The microphone is fixed to the semi-infinite length housing. Deviation between measured values (points) and the theoretical values (solid line) is kept within 0.4 dB. For the measurement of the microphones having other arbitrary shapes, the same order of precision shall be available.

Fig. 6 Frequency response and electrical impedance of the spherical sound source.

Fig. 7 Diffraction coefficient of the standard condenser microphone MR-103.

References
Introduction

If we had a sound receiving system which could focus on any point in the sound field, we could pick up only the desired sound signal, so that a high signal-to-interference ratio (S/I) could be obtained. In this paper, the focusing effect (F.E.) of the linear array sound receiving system (L.A.S.R.S.) has been studied, which consists of microphones arrayed at equal intervals in a line and delay networks. Moreover, when a desired signal and many undesired signals are distributed in the field, the S/I of the output signal of the L.A.S.R.S. has been analyzed and compared with that of other type directional microphones.

Method of Obtaining F.E. by L.A.S.R.S.

First, consider the case in which 2n omnidirectional microphones are arrayed at intervals of a on the Y axis, as shown in Fig.1. In order to focus the system on a point F(D,0), it is necessary that the output signals of the whole microphones have the same phase when a sound source is situated at F. To achieve this, the output signal of the i-th microphone should be combined with the output signals...
Focusing Effect of Linear Array System

of other microphones via a delay network with the delay time $t_i$. The value of $t_i$ is given as

$$t_i = \frac{1}{c} \sqrt{v^2 + (\Omega - 0.5) \alpha^2} - \frac{1}{c} \sqrt{v^2 + (i - 0.5) \alpha^2}, \quad t_{i+1} = t_i, \quad i = 1, 2, \ldots, n,$$

where $c$ is the velocity of sound.

**Focusing Effect of L.A.S.R.S.**

Fig. 2 shows the variation of the total output voltage, that is F.E., of the L.A.S.R.S. with $2n=8$, $D=10a$ while a point sound source travels along the $X$ axis. Fig. 3 shows the variation of the total output voltage while a point sound source travels along the line $x=D$. These calculated results indicate the following characteristics. The F.E. becomes sharper as the frequency increases. The F.E. along the $Y$ axis is about 10 times sharper than that along the $X$ axis. A grating lobe appears at a direction of $\frac{y}{D} \approx \sqrt[4]{(ka/2\pi)^2 - 1}$.

**Experimental Verification of F.E.**

Fig. 4 shows the block diagram of the L.A.S.R.S. consisting of 8 omni-directional moving coil microphones and a delay network. The F.E. of the system have been measured by moving a sound source in an anechoic room. Fig. 5 and Fig. 6 show the measured F.E. at the frequencies 0.9, 1.3, 3.6 and 7.2 kHz ($ka=5, 10, 20, 40$) along the $X$ axis and the line $x=D$ respectively. The curves in Fig. 5 and Fig. 6 agree fairly well with the calculated results.
Focusing Effect of Linear Array System

shown in Fig. 2 and Fig. 3. Fig. 7 is the F.E. along the X axis measured by the 0.3~9.6 kHz band noise. Small circles in this figure indicate the calculated results, and agree with the measured curve. This figure shows that some F.E. can be obtained even for a wide band noise.

Effects of Various Parameters on F.E.

The variations of the F.E. caused by the change of the focal length D, the total length of the array L and the total numbers of the microphones N have been studied. From the analysis, it becomes evident that a considerable F.E. can be obtained in the region of D\leq L, and the sharper F.E. can be obtained by the use of the larger number of microphones. The variations of the F.E. caused by deviations of the sensitivity or phase angle of the microphones from the ideal values have also been analyzed. Then, it becomes

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Fig. 4 Block diagram of linear array sound receiving system

Fig. 6 Focusing effect along the line x=0.

Fig. 5 Focusing effect along the X axis.

Fig. 7 Focusing effect along the X axis measured by 0.3~9.6 kHz band noise.
clear that the deviation of the sensitivity has little effect on the F.E., however, the deviation of the phase angle degrades the F.E. a little.

Comparison of S/I between Various Directional Microphone Systems

Consider the case in which 24 undesired signals are distributed around the desired signal at intervals of 1m, as shown in Fig. 8, and they radiate random signals of $1/4$ intensity of the desired signal. Table 1 shows the calculated S/I of the output signals of the L.A.S.R.S. with $N = 40$, $a = 25\text{cm}$, a parabolic reflector with $1\text{m}$ diameter, a line microphone with $1\text{m}$ length, an unidirectional microphone or an omnidirectional microphone, when the systems are gathering the both signals from a point $10\text{m}$ apart from the desired signal. According to our experiment, the allowable S/I that one can catch one speech out of two mixed speeches by a little effort is about $0 \text{ dB}$. Therefore, Table 1 indicates that the L.A.S.R.S. can pick up an understandable signal out of the undesired signals shown in Fig. 8.

Conclusion

As the results of the analysis and measurement of the L.A.S.R.S., the following conclusions have been attained. The calculated results of F.E. agree fairly well with the measured results. A considerable F.E. can be obtained in the region of $D \leq L$. Some F.E. can be obtained even for a wide band noise. The L.A.S.R.S. can pick up the desired signal with a better S/I than the case of ordinary directional microphones out of the noises.

![Fig. 8 Distribution of desired and undesired signal.](image_url)

### Table 1

<table>
<thead>
<tr>
<th>Frequency (kHz)</th>
<th>0.27</th>
<th>0.54</th>
<th>1.08</th>
<th>2.16</th>
</tr>
</thead>
<tbody>
<tr>
<td>L.A.S.R.S. (N=4)</td>
<td>-1.5</td>
<td>-0.1</td>
<td>3.0</td>
<td>8.0</td>
</tr>
<tr>
<td>Parabolic Reflector</td>
<td>-7.8</td>
<td>-7.4</td>
<td>-5.9</td>
<td>-2.8</td>
</tr>
<tr>
<td>Line Microphone</td>
<td>-8.0</td>
<td>-8.0</td>
<td>-7.9</td>
<td>-7.8</td>
</tr>
<tr>
<td>Unidirectional Microphone</td>
<td>-7.9</td>
<td>-7.9</td>
<td>-7.9</td>
<td>-7.9</td>
</tr>
<tr>
<td>Omnidirectional Microphone</td>
<td>-8.0</td>
<td>-8.0</td>
<td>-8.0</td>
<td>-8.0</td>
</tr>
</tbody>
</table>
Speed-Tuned Porous Pipe Receivers for Sound Reception in Motion

Ewald Eichler

Raytheon Company
Bedford, Massachusetts 01730, U.S.A.

Line receivers comprising porous pipes are advantageous for sound reception in rapid motion under the conditions that the sound signals arrive as plane waves from nearly the upstream direction (direction of receiver motion) and that the inherent noise is largely self-generated flow-noise. Their advantages derive from an aerodynamic shape that minimizes the generation of flow-noise and an acoustic circuit that reduces, with respect to the signal, the remaining noise.  

The schematic diagram of a porous-pipe receiver is shown in Figure 1. The slender streamlined body exhibits as central portion a porous pipe of length \( L \) and diameter \( D \). Its pores are narrow enough to give a relatively smooth surface and a relatively high specific wall impedance \( Z_w \). For \( Z_w \gg \rho cL/D \) (\( \rho \) mass density and \( c \) sound speed of the medium), or weak coupling, the porous pipe receiver approaches a simple line receiver: the fluctuating pressures that exist across the outer surface of the pipe and that comprise both signal and noise cause proportional inflow and outflow of medium through the elements of the porous wall. The flows, in turn, cause elements of sound waves traveling inside the porous pipe. The elements combine to form two waves, one traveling to the front and one traveling to the rear of the pipe. The travel occurs at a characteristic phase-slowness

\[
\alpha_p = \left[ \frac{1}{c_i^2} - 2\bar{\nu}/\pi \bar{D} \left( Z_w + Z_r \right) \right]^{1/2}
\]

\( c_i \) sound speed of medium inside pipe, \( \bar{\nu} \) frequency, \( Z_r \) radiation impedance). The optimum wall impedance, the result of a compromise between directivity and sensi-
Speed-Tuned Porous Pipe Receivers for Sound Reception in Motion

tivity, is of order of magnitude

\[ Z_w = 10 \rho_c (1 + M)^2 L/D \]  \hspace{1cm} (2)

\( M = U/c \) Mach number of receiver velocity \( U \).

The wave traveling to the rear is presented via duct terminal 3 (and connecting ducts), to an acousto-electric transducer. By virtue of their phase relations and the appropriate phase-speed \( 1/\text{Re}_{p} \), the elements composing the wave interfere constructively, as far as they are caused by external sound signals of appropriate directionality, and more or less destructively, as far as they are caused by noise. This scheme reduces, with respect to the desired signal, both the flow-noise that remains being generated in spite of the streamlining and also the general background noise.

The wave traveling to the front inside the pipe must be prevented from being reflected back. The reentrant pipe 2, continuing the channel inside the porous pipe, allows to extract this wave at duct terminal 4 or to absorb it by means of a porous absorber placed inside the reentrant pipe.

The phase-speed of the line receiver defines a "matched velocity" \( U_m \) and its Mach-number \( M_m \) according to

\[ \frac{1}{\text{Re}_p} = c + U_m = (1 + M_m)c \]  \hspace{1cm} (3)

At this velocity, endfire reception of the receiver in the upstream direction is optimal. At other velocities, the Doppler effect leads to a deterioration of the directivity pattern that becomes appreciable when

\[ |M - M_m| > (1 + M_m)\lambda/4L \]  \hspace{1cm} (4)

(\( \lambda \) sound wavelength). Speed-tuning counters this trend by a change of phase-slowness \( (1) \) via a change of \( c_1 \). Figure 2 shows the effect of speed-tuning in some simple line receivers.

The change of \( c_1 \) can be realized by heating the pipe or by filling it with an appropriate gas mixture. Duct terminals 3 and 4 (Figure 1) allow also the circulation of gas through the structure. To prevent excessive loss of heat or filling gas, the roughened inside of the porous pipe 1 may be lined with a very thin, loosely attached plastic film.
The three types of flow-noise that require reduction with respect to the signal are (1) local noise from fluctuating drag forces due to medium inhomogeneity (inclusive of turbulence), (2) radiated flow-noise, and (3) local noise due to an attached turbulent boundary layer.

The reduction of inhomogeneity noise (the ratio of the speed-matched receiver spectrum over the spectrum of a point receiver located at the porous-pipe surface) is given by

\[ NR_1 = j_0^{-2} \left\{ \frac{1}{2\pi f L/(1 + M)} \sqrt{U} \right\} \]  

\[ (j_0 \text{ spherical Bessel function}). \text{ The reduction becomes appreciable when } \bar{f} > U/2L[1 - M/(1 + M)]; \text{ it could be increased at higher frequencies by end-tapering of the spatial sensitivity of the line. Under the assumption that radiated flow-noise comes entirely from downstream of the receiver, its reduction is given by} \]

\[ NR_T = j_0^{-2} \left\{ \frac{2\pi L f/c(1 - M^2)}{1 + M^2} \right\} \]  

\[ \text{It becomes appreciable when } \bar{f} > c(1 - M^2)/4L, \text{ and could likewise be increased at higher frequencies by end-tapering.} \]

The boundary-layer noise has been calculated on the basis of an assumed correlation model of the pressure fluctuations that has formerly been used for homogeneous, flat, constant mean-pressure wall-layers. In terms of an equivalent plane wave arriving from upstream, the noise spectrum is given approximately by

\[ S_{eq}^P(f) = \frac{8(1 + M)^2(\bar{p}^2) \theta}{\kappa^2 DL^2[1 + (2\pi f)^2(1 + M)^2]} \]  

\[ (\bar{p}^2) = (2.66\tau_w^2), \tau_w \text{ wall shear stress, } \theta = 300\delta^2/U, \delta^* \text{ displacement thickness,} \]

\[ \kappa = 1.25/\delta^* \]. Figure 3 shows computations for a speed-matched porous-pipe receiver. The boundary-layer parameters are taken from the midpoint of the pipe.\(^1\)

Preliminary wind-tunnel measurements between \( M = 0.097 \) and \( M = 0.35 \) exceed the results of Figure 3 substantially (by 7 to 23 db) but are still much lower than the flow-noise achieved with wind-screens and streamlined caps of conventional microphones. They are also lower than those reported for a discretely perforated pipe receiver.\(^1\)\(^2\)

**Acknowledgement**

This work was supported by the U.S. Navy, Office of Naval Research.
Speed-Tuned Porous Pipe Receivers for Sound Reception in Motion

References

![Diagram](image)

Fig. 1. Schematic diagram of porous-pipe receiver.

![Graph](image)

Fig. 2. Dependence of directivity of simple line receivers upon velocity and speed-tuning. (The received frequency \( f \) equals the transmitted frequency only at \( \alpha = 90^\circ \).)

![Graph](image)

Fig. 3. Flow-noise from turbulent boundary layer attached to speed-matched porous-pipe receiver (\( L = 305 \text{ mm}, D = 31.6 \text{ mm}, Z_w = 75 \text{pc}, \) without reentrant pipe).
Parametric Directional Microphone

Hikaru Date and Yoshinori Tsuchida

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(Japan Broadcasting Corporation), Tokyo

There has been two methods for directional sound reception of relatively wide frequency range of audio signal, such as the well-known phase difference method and the Doppler method\(^1\). The third method is proposed here, which utilized the sound velocity modulation by strong acoustic waves.

**Two Different Types of Parametric Directivity**

If the particle displacement is not infinitesimal, the sound velocity is given by

\[
C = C_0 \left( 1 + \frac{\gamma + 1}{2 \gamma} \frac{P}{P_0} \right)
\]

in which \(P_0\) is the static pressure, \(\gamma\) is the ratio of specific heats at constant pressure and constant volume, \(C_0\) is the sound velocity for small amplitude and \(P\) is the sound pressure.

If strong ultrasonic field is locally produced, the wavelength constant of the incident audio wave, passing through the strong intensity region, is modulated by the ultrasonic wave because of modulated sound velocity. Therefore, as travelling together with the ultrasonic wave through some distance in that region, the incident audio wave is gradually phase-modulated. Owing to the locally limited region, the effective length of the common path changes as the incident direction varies. The degree of phase modulation observed by a micro-
Parametric Directional Microphone

phone located at the far end of that region becomes a function of incident angle of the audio wave and one type of "parametric directivity" can be obtained. We call this type the case "a".

Conversely, for the case of ultrasonic wave with very high frequency, the ultrasonic wave can be effectively phase modulated by the audio wave even if intensity of ultrasonic wave is not very large. Let us call this type the case "b".

Derivation of Directivity

Two plane waves propagating in different directions are considered (Fig. 1). If one wave, whose angular frequency is \( \Omega \), is of finite amplitude, the wavelength constant \( k \) of the other weak wave whose angular frequency is \( \omega \) becomes

\[
k(x,t) = \frac{\omega}{c} = \frac{\omega}{c_o} \left(1 - \frac{\Omega + 1}{2 \gamma} \cdot \frac{p_a(x,t)}{p_o} \cdot \cos \theta \right),
\]

where \( p_a(x,t) \) is the instantaneous pressure of the \( \Omega \)-wave. The second term of the equation (2) contributes the phase modulation of the \( \omega \)-wave during the propagation. The factor "\( \cos \theta \)" shows the modulation mechanism is not isotropic.

Suppose the phase of the \( \Omega \)-wave is \( \chi \) at the initial condition \( (t=0, x=0) \) and the equiphase line of the \( \omega \)-wave at the initial condition arrives at the position \( x = x_o \), the position \( x_o \) is just on the equiphase line of the \( \Omega \)-wave whose phase is \( \frac{\Omega}{c_o} x_o (1 - \cos \theta) + \chi \). Therefore, the degree of phase modulation \( \Delta \Phi \) of the \( \omega \)-wave, caused by travelling with the \( \Omega \)-wave along a path between two points \( x = 0 \) and \( x = d \), is calculated from the following integration

\[
\Delta \Phi = \frac{\omega}{c_o} \frac{\Omega + 1}{2 \gamma} \frac{p_a}{p_o} \int_0^d \sin \left\{ \frac{\Omega}{c_o} (1 - \cos \theta) x + \chi \right\} \cdot \cos \theta \cdot dx,
\]

where \( p_a \) is the pressure amplitude of the \( \Omega \)-wave. Because the initial phase \( \chi \) can take an arbitrary value between 0 and \( 2\pi \), the modulation index \( \Delta \Phi \) observed at the position \( x = d \) is the maximum value of \(|\Delta \Phi|\).

\[
\Delta \Phi = \max_{0 \leq \alpha \leq 2\pi} |\Delta \Phi| = \max_{0 \leq \alpha \leq 2\pi} \left| \frac{\omega}{c_o} \frac{\Omega + 1}{2 \gamma} \frac{p_a}{p_o} \cdot \frac{1}{\delta} \left\{ \cos \alpha - \cos (\alpha + \delta) \right\} \cdot \cos \theta \right|
\]

\[
= \frac{\omega}{c_o} \frac{\Omega + 1}{2 \gamma} \frac{p_a}{p_o} \cdot \sin \left( \frac{\delta}{2\pi} \right) \cdot \cos \theta
\]
Parametric Directional Microphone

where \( \delta = \frac{d}{c} \), \( \sin c(x) = \sin \pi x / \pi x \). Therefore the directivity function \( D(\theta) \) is given by

\[
D(\theta) = \sin c \left( \frac{\delta}{2 \pi} \right) \cos \theta
\]

(6)

The first term is equal to the directivity of the linear continuous array of simple sources and the second term is the directivity due to the non-isotropic property of sound velocity modulation.

This model is suitable for the case "b". For the case "a", its directivity is expected to become sharper than \( D(\theta) \) of the equation (6) and independent from frequency of audio signals, chiefly because of the localization effect.

**Experimental Results**

A PZT transducer, 1cm in diameter and 0.7cm in thickness, was employed to radiate 90kHz ultrasonic wave. The Brüel & Kjær Model 4135 condenser microphone was used as a receiver.

Fig. 2 shows the observed phase modulation when 18 and 90kHz waves travel together in the same direction. The amplitudes of 18 and 90kHz waves decrease with distance in accordance with \( r^2 \)-law or more in the region of microphone position, from 20 to 100cm; nevertheless the side band wave of 72 and 108kHz decay more slightly in the same region. The amount of this discrepancy is approximately proportional to the distance. This confirms the existence of phase modulation in the field.

The directivity patterns were recorded by a level recorder, Brüel & Kjaer Model 2305, with 10-100mV potentiometer under the following conditions. The sound pressure level of 90kHz wave was about 101db at the microphone position 30cm apart from the radiator. The sound pressure level of audio wave was changed from 85 to 95db. The common path length was 30cm.

Fig. 3(b) and (c) are the measured patterns of the parametric directivity for 10 and 8kHz audio waves respectively. The first null angle for the directivity patterns agree with the calculated values from the equation (6), and the tendency of frequency-dependent direc-
Parametric Directional Microphone
tivity patterns is observed. These facts suggest the experimental results belong to the case "b".

The parametric directivity, introduced here as a new principle, has possibility of sharp directional sound reception without significant disturbance of original sound field, because it directly utilizes modulation of medium property. As far as high frequency ultrasonic wave is used, transducers can be made so small that this merit will be acquired in practice.

Reference
1) H. Date: Variable Directional Microphone Using Doppler Effect, J46, 5th ICA, Liège, (1965).

Fig. 1 Wavefronts of two plane waves.

Fig. 2 Growth of side band signal along the travelling path.

Fig. 3 Measured directivity patterns, (a) radiation of 90kHz transducer, (b) and (c) are parametric directivity of 10 and 8kHz sound respectively.

-D-128-
Introduction

In magnetic tape recording, reproducing voltages decrease over high frequency ranges. One of the reasons for this seems to be that magnetization distribution in the magnetic material of a tape has a phase difference from the direction of the tape depth. It is difficult, however, to consider the reproducing voltages in tapes having a phase difference between the magnetization distribution and the tape depth, because there isn't a clear way to determine magnetization distribution in the recording process. In this paper, a method will be described in which tapes are made to have a known phase difference, permitting the reproducing voltages to be measured, and the effects of the phase difference on those voltages to be clarified.

Basic Considerations

Many papers have been written on the determination of reproducing voltages. Three methods of calculation have been used. One method uses the reciprocal principle:① If all parts of the medium, such as the tape material, behave linearly in
Effects of the Magnetic Distribution with Phase Difference

to their magnetic and electric properties, voltage $e$ in the case of a harmonically recorded signal, is given by

$$e = -N \frac{d}{dt} \left[ I_0 \int_0^1 J_0(y,z) \cdot H_0(x,y,z) \cdot \cos(\beta x - 2\pi ft) \, dt \right],$$

(1)

where $N$ is the number of windings of the reproducing head; $J_0(y,z) \cos(\beta x - 2\pi ft)$ is the magnetization distribution of the tape, and $H_0(x,y,z)$ is the intensity of the magnetic field which occurs in the case of a flowing recording current $I_0 \cos 2\pi ft$ into the head. In this method, $H_0$ is considered to be a kind of weighting function for $J_0$.

The second method uses the Gauss’s theorem.(2) In this case, the minus second power of distance is considered to be the weighting function.

The third method, unlike the other two, doesn’t use a weighting function. Instead, it considers the possibility of some kind of reaction between all the poles in the recorded tape material.(3) It is important to know which has a larger effect on the reproducing voltage, such a reaction between poles, or a weighting function.

Experimental Procedure

Fig.1 shows an experimental method of determining which has the larger effect. Two tapes, their magnetic coating facing each other, move over the air gap of a reproducing head at speeds which vary slightly, thus causing the phase difference to change from zero to $\pi$ radians(Fig.2).
Fig. 3 is a typical waveform of voltage at the playback head. The maximum and minimum points correspond to the phase difference of Fig. 2, respectively. The abscissa represents changing phase difference with time; the difference between the maximum and minimum points along the abscissa is \( \pi \) radians. The envelope has irregular pulses due to the variation of tape speed as the tape joint passes the tape guide. By analyzing photographs of reproducing voltage (see Fig. 3), Fig. 4 is obtained. Nonlinearity is found between reproducing voltage and the quantity of magnetization. Fig. 5 is obtained from the analysis of these same photos with regard to maximum and minimum values of reproducing voltage envelopes. The points on the chart are measured data. These lie along two straight lines.

**Discussion**

The weighting function (the second on reproducing voltage, for method) is applied to the experiment. Different amount of magnetization of the rear tape shown in Fig. 1(c) Since the remnant magnetization of the tape has only the longitudinal component because the wavelength of the recorded signal, given by \( \lambda_B \), is 475 \( \mu \), the magnetization \( J_x \) is given by

\[
J_x = \begin{cases} 
J_m \sin 2\pi x/\lambda_B & (\delta \geq z > 0) \\
kJ_m \sin (2\pi x/\lambda_B - \theta) & (0 > z \geq -\delta),
\end{cases}
\]

where \( J_m \) is the amplitude of magnetization and \( \delta \) is the tape depth.

From Eq. (2), reproducing voltage is calculated and maximum and minimum values with respect to phase difference are given by

\[
\frac{E_{max}}{E_{min}} = (1 + \frac{4\delta}{\lambda_B})^k e^{-2\pi \delta/\lambda_B}
\]
When the reaction between poles is similarly applied to the experiment, these values are given by

$$E_{\text{max}} = \left(1 + \frac{2f}{f_l}\right)k,$$

(4)

$E_{\text{min}}$.

Fig. 6 is the graph of Eqs (3) and (4). The solid line A represents the case of weighting function. The dotted line B is the case where only reaction between poles occurs.

Referring to Fig. 5, we can see that the weighting function is the more important factor.

Conclusion

It has been shown that, if the tape depth is very small compared to recorded wavelength, the weighting function is the more important factor. With such a condition that the reaction between poles can be ignored. However, if the tape depth isn’t small, the component of tape depth of magnetization seems to have a reaction between poles too great to be ignored.

Therefore, in such cases, it is necessary to consider this reaction.

Acknowledgement

Helpful suggestions were given by Prof. Nobuyoshi Tanaka and my fellows at the Electrical Engineering Department, Tokyo University of Agriculture and Technology. Mr. Takuo Sakagami also kindly helped by making part of the equipment for the experiment.

REFERENCES

Amplitude Distribution of Orchestral Music

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Introduction

Since orchestral music shows the limits of physical values among usual program sounds as regards the magnitude of sound pressure and the range of frequency band and dynamic level, measurements were made of the instantaneous-sound-pressure distributions for orchestral music, which were composed between the eighteenth and the twentieth century.

Definition

Sample amplitude distribution $F(x,T)$ for a signal within duration time $T$ depicted in Fig. 1 is defined by

$$F(x,T) = \frac{T x}{T}$$  \hspace{1cm} (1)

where $T x = \sum t_i (X(t) \geq x)$. When $T$ is equal to the total duration of a whole-work, $F(x,T)$ will be denoted by $F_x$.

Preliminary Investigation of Some Statistical Features of Amplitude Distribution for Orchestral Music

Two remarkable properties were ascertained.

(1) The invariability of the distribution with delay-time frequency distortion.

(2) The symmetry of the distribution for the polarity of a signal.

The first property made it possible to use the usual microphones and tape-recorders for broadcasting application which have considerable amount of delay-time frequency distortion. The second made it sure that measurements for half-wave signal were sufficient to obtain the full information of the distribution. Regarding nonlinearity effects on the distribution, the upper limit of the linearity of recording level on the VU-meter reading was determined. This reading was 3-1/2 VU.

* Speech distribution does not have these properties.
Approximate Expression of the Distributions for Whole-Work or Movements

Some examples of measured distributions are shown in Fig. 2. They are almost exponentially shaped over much of the range except in the portion at lower level. Therefore, it seems natural to approximate the obtained distributions by the normalized composite expression, for \( x \geq 0 \)

\[
F_n(x) = \frac{a}{2} \exp\left(-\frac{\sqrt{2} x}{x_{n1}}\right) + \frac{b}{2} \exp\left(-\frac{\sqrt{2} x}{x_{n2}}\right) + \frac{c}{2} \int_a \delta(x) dx
\]

where \( x \) denotes instantaneous sound pressure, and \( x_{n1} \) and \( x_{n2} \) are the normalized root-mean-square values of sound pressures, while \( a \), \( b \), and \( c \) are coefficients of the partitions of these sounds, each of which gives respective distribution shown in the above expression.

The first two terms give exponential distributions with a large \( x_{n1} \) and a small \( x_{n2} \) for the root-mean-square values. The third term is a distribution function with a density function \( \delta(x) \) so called Dirac's, for the extremely low level signal such as attenuated reverberant sounds.

We have here two constraints for the coefficients in (2),

\[
\begin{align*}
\frac{a+b+c}{a} & = 1, \\
\frac{x_{n1}^2 + \theta x_{n2}^2}{a} & = 1
\end{align*}
\]

The former is probability condition and the latter is power condition.

The Results and Analysis of the Approximated Distribution

The estimated statistics of the coefficients of the approximated distributions are tabulated in Table 1.

<table>
<thead>
<tr>
<th>Symphony</th>
<th>Concerto</th>
</tr>
</thead>
<tbody>
<tr>
<td>( m )</td>
<td>( s )</td>
</tr>
<tr>
<td>( \sigma ) (%)</td>
<td>50.0</td>
</tr>
<tr>
<td>( b ) (%)</td>
<td>44.8</td>
</tr>
<tr>
<td>( \theta )</td>
<td>5.2</td>
</tr>
<tr>
<td>( x_{n1} )</td>
<td>1.41</td>
</tr>
<tr>
<td>( x_{n2} )</td>
<td>0.36</td>
</tr>
<tr>
<td>( \nu )</td>
<td>1.02</td>
</tr>
<tr>
<td>( \beta )</td>
<td>4.26</td>
</tr>
<tr>
<td>( PF ) (dB)</td>
<td>22.6</td>
</tr>
<tr>
<td>( F_n ) (%)</td>
<td>9.3</td>
</tr>
<tr>
<td>( p )</td>
<td>0.92</td>
</tr>
<tr>
<td>( n )</td>
<td>0.08</td>
</tr>
</tbody>
</table>

\( a, b, \sigma, x_{n1} \) and \( x_{n2} \) are constants in the expression (2).

\( PF \): ratio of peak to r.m.s. pressure, i.e. peak factor.

\( F_n \): percentage of time for which signal exceeds r.m.s. level.

\( \rho_a = a x_{n1}, \rho_b = b x_{n2} \) : average powers of signal, for the first two distributions in the expression (2).

\( \Omega_{n1} = \sqrt{2}/x_{n1}, \Omega_{n2} = \sqrt{2}/x_{n2} \)

\( m \): mean, \( \sigma \): standard deviation, \( \rho \): coefficient of correlation, \( \rho_{nc} \): coefficient of regression.

Fig. 2. Examples of amplitude distributions of orchestral music.

1. Khachaturian, Gayne, ballet suite no. 1. 2. Beethoven, symphony no. 3. 3. Beethoven, piano concerto no. 3 op. 37. 4. Mozart, symphony no. 29 K 201.
Some characteristics of the distributions for the orchestral music can then be derived from these obtained estimates:

(a) The principal power of the orchestral-music sounds is included in the signal that gives the first exponential term of \(2\).

(b) There are large correlations between \(\sigma\) and \(\lambda_{\text{rms}}\) because of the constraint \(3\).

(c) The coefficients \(a\) and \(b\) have comparatively large standard deviations.

Thus, we can show in Fig. 3 the typical behaviour of the orchestral-music distributions approximated by the first term only in \(2\), according as the various values of \(\sigma\) or \(\lambda_{\text{rms}}\).

**Variation of Sample Distribution**

The distributions for the shorter-time orchestral sounds than whole-work or movement duration show great deal of variability.

If it is assumed that a series of \(F(x, T)\) is an independent discrete stationary time series with a definite positive variance, the relation between the population coefficient of variation \(CV\) of \(F\) and measurement time \(T\) is theoretically given by

\[
\sigma(F, T) = \frac{\sigma_F}{\sqrt{T}}
\]

where \(\sigma\) is a coefficient. The sample coefficient of variation \(CV\) is expressed by

\[
CV(x, T) = \frac{S'(x, T)}{F_x}
\]

where \(S'\) is the unbiased estimate of the standard deviation of \(F\).

It was found that for almost all of the music the plots of \(CV\) and \(T\) give satisfactory fits to the inverse root law \(4\). By the application of the least-squares method to the sample coefficient \(CV\), estimates \(\hat{\sigma}\) for \(\sigma\) were obtained.

Since these coefficients \(\hat{\sigma}\) plotted against \(F_x\) on a semilogarithmic paper show considerable linearity, an experimental formula

\[
\sigma(F_x, T) = -a \log_2(F_x / 0.5) + b
\]

was obtained for the measured sample distributions.

**Physical Meaning of \(a\) and \(b\)**

If a time-series of segmental sinusoidal waves with sample measurement time \(T\) is considered, the values of \(a\) and \(b\) for the process would become zero. If we give some variation in rms values to the segmental sine waves, then \(a \geq 0\) and \(b = 0\). Moreover, when various lengths of recess times are provided in respective samples without changing the duration time \(T\), \(a\) would become not less than zero and \(b = 0\) in almost all cases.

From this consideration, it can be said that \(a\) represents the intensity of slow level-variation of orchestral music and \(b\) gives that for very low level signal. In this sense the author names the pair of constants \((a, b)\) the variation constants from the viewpoint of level distribution.

---

D-135
A list of variation constants for the representative 27 orchestral music is given in Table 2.

<table>
<thead>
<tr>
<th>No.</th>
<th>Composer</th>
<th>Title of work played</th>
<th>Conductor</th>
<th>Orchestra</th>
<th>Level-variation constant ( \alpha )</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Mozart, W. A.</td>
<td>No. 40, K. 550, G minor</td>
<td>A. Rumpf</td>
<td>NHK</td>
<td>2.0</td>
</tr>
<tr>
<td>2</td>
<td>Haydn, F. J.</td>
<td>No. 103, E flat major “Mit dem Paulenwirbel”</td>
<td>T. Mori</td>
<td>Tokyo*</td>
<td>5.4</td>
</tr>
<tr>
<td>3</td>
<td>Beethoven, L. V.</td>
<td>No. 2, D major</td>
<td>H. Iwaki</td>
<td>NHK</td>
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<td>4</td>
<td>Beethoven, L. V.</td>
<td>No. 4, B flat major</td>
<td>A. Rumpf</td>
<td>NHK</td>
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<td>5</td>
<td>Beethoven, L. V.</td>
<td>No. 6, F major “Pastorale”</td>
<td>A. Rumpf</td>
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<td>No. 1, C minor</td>
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<td>Brahms, J.</td>
<td>No. 1, C minor</td>
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<td>“Pacific: 2 3 17&quot;, mouvement symphonique</td>
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<td>27</td>
<td>Honegger, A.</td>
<td>“Prelude, fugue et postlude”</td>
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*1 NHK Symphony Orchestra.  
*2 Tokyo Philharmonic Symphony Orchestra.  
The conditions for the above results in no. 8, 9, differ in hall, no. 10, 11 in the day of performance, no. 12, 13, 14, in the position of a microphone, respectively.

**Consideration on Dependent and Independent Factors of the Whole-Work Distribution and Variation Constants**

It was proved from other experiments that the whole-work distribution and variation constants are independent of such variable conditions as a microphone, its position, and the music hall used.

Also, when the general function of a conductor is taken into consideration from the physical viewpoint, it is believed that the measured distributions and the values of the variation constants are dependent on the interpretation of the music by the individual conductor. Accordingly, the measured distribution and the pair of variation constants are not intrinsic to each orchestral music.

**Conclusion**

So far as the author knows, this investigation provides the first results in the pressure distribution for orchestral music which meet the requirements for the application of information Theory.

Further study on the spectral amplitude distribution would be of interest.
Transducer Techniques for Measuring the Effect of Small-Arms' Noise on Hearing
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U.S. Army Human Engineering Laboratories
Aberdeen Proving Ground, Maryland, 21005, USA

Improvements in small arms, within recent years, have raised the Sound Pressure Level (SPL) at the operator's ear until many firers show large hearing losses. The purpose of this report is to help establish a uniform procedure for measuring small arms' pressure waves accurately -- the primary requirement for evaluating how such weapons affect hearing.

One of a transducer's most important characteristics when measuring impulse noise is its rise time capability. Although no present-day transducer can follow the pressure rise exactly, the device chosen must be able to reach a peak before significant pressure decay occurs. In order to evaluate this characteristic several pressure transducers were exposed to the pressure produced by a shock tube. The shock tube nominally produces a shock wave which rises "instantaneously" to a preselected pressure, remains at that pressure for a short time, then gradually returns to ambient pressure. When a transducer measures the shock-tube pressure wave, the pressure-vs-time history oscillogram is an accurate index of the transducer's rise-time capability for a given pressure and a given angle of incidence. Moreover, the shock tube produces an accurate, preselected pressure, which is a useful reference for verifying other calibration methods. The transducer's ringing and overshoot characteristics may also be evaluated. The results indicate that at a grazing transducer incidence angle, rise time for different transducers varied from 10 to 170 \( \mu \)sec; it can be seen that at this incidence angle the shortest possible rise time will be determined by transit time -- the time it takes the pressure wave to cross the face of the transducer. At normal incidence all transducers exhibited wave distortion due to overshoot and several showed severe ringing.

It was determined that the overshoot at normal incidence was produced by reflections off the face of the transducer. Therefore, it was decided to investigate how intermediate incidence angles affect measured pressure-time histories. The wave shape chosen for the investigation was the shock wave of a supersonic 7.62 mm projectile in flight which produces the classic "N" wave.
This wave shape was measured with two transducers: a) a BRL\textsuperscript{1} 250-kc lead zirconate pressure gauge and b) a Bruel & Kjaer (B&K) type 4136 capacitor microphone. Both transducers were oriented at incidence angles varying in 30\textdegree increments from 0\textdegree to 90\textdegree with reference to the bow wave of the supersonic projectile. The resulting oscillographic waveshapes are shown in Fig. 1 for both transducers at 0\textdegree and 90\textdegree incidence. Ideally, the "bow wave" of the projectile should produce an instantaneous increase to some positive amplitude $P_1$.

![Waveforms](image)

Fig. 1. Pressure vs Time History Produced by the Shock Wave of a 7.62mm Projectile in Flight When Measured with BRL 250-kc and B&K 4136 Transducers at 90\textdegree and 0\textdegree Incidence (sweep time $\approx$ 50 $\mu$sec/major division).

The pressure then decreases linearly until it reaches a negative value, $P_2$, where $|P_1| \approx |P_2|$, and then the "stern wave" instantaneously returns the pressure to ambient. The important point here is that the pressure decrease from positive to negative pressure ($P_1$ to $P_2$) should approach a straight line, and have no overshoot when returning to ambient. The two transducers, when positioned at 90\textdegree incidence, agree within 0.3 dB when measuring the peak pressure produced by the "N" wave; and its duration is the same (150 $\mu$sec).

Also the wave shapes produced by the two transducers are very similar, and both have the required straight line between $P_1$ and $P_2$.

As the transducers are rotated from an incidence of 90\textdegree to 60\textdegree several changes occur. The peak-pressure measurements are higher than at 90\textdegree incidence. The higher peak is caused by the pressure reflected off the transducer's face. Also the decay from $P_1$ to $P_2$ is no longer linear and a peak is created as the pressure returns from $P_2$ to ambient. This smaller peak is due to reflection from the transducer's face as the stern wave passes, as well as slight transducer overshoot. The reflected pressure phenomenon may be seen clearly in Fig. 2 which shows a projectile's shock wave striking

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\textsuperscript{1}Ballistic Research Laboratories, Aberdeen Proving Ground, Maryland.
Transducer Techniques

a transducer. A small spherical shock wave is generated, expanding until it reaches the corner of the transducer, and then dissipating, since there is no surface to support this reflected pressure.

Table 1 shows how peak SPL increases over the 90° incidence measurement as the transducers are rotated from an incidence angle of 90° to 0°.

<table>
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<th>Incidence Angle (degrees)</th>
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<td>90°</td>
<td>B&amp;K 3.6</td>
</tr>
<tr>
<td>60°</td>
<td>B&amp;K 1.1</td>
</tr>
<tr>
<td>30°</td>
<td>B&amp;K 3.6</td>
</tr>
<tr>
<td>0°</td>
<td>B&amp;K 3.6</td>
</tr>
</tbody>
</table>

The B&K 4136 produces a smaller variation from the 90° measurement at 0° than the BRL 250-kc since it has a lower frequency response (70 kHz vs 250 kHz) and also is more heavily damped. These characteristics can be readily seen in Fig. 1 where the BRL 250-kc transducer produces minor higher frequency deviations during the pressure decay than the B&K which averages these variations into a smoother decay curve.

Fig. 1 also indicates that the BRL 250-kc transducer at 0° incidence creates additional difficulties -- the measured peak SPL includes overshoot inherent in the design of the transducer, as well as reflected pressure. The net result is a misleading incident pressure-time history. The B&K transducer’s record at 0° incidence indicates that its best rise time is about 10 μsec, which agreed with the shock tube measurements. Because of this poorer rise time capability, reflected pressure and overshoot will not increase peak SPL at 0° as much as with the BRL 250-kc transducer.

Thus far in evaluating a transducer’s ability to accurately measure small-arms’ pressure waves, angle of transducer incidence has been found to be very important. Also, since the pressure wave we are measuring is of such short duration, rise time must be kept as short as possible. Therefore we also investigated the rise time capabilities of various types of transducers at different pressures. This was accomplished by measuring the pressure time history of the expanding muzzle gasses of a 7.62mm rifle at points 0.25, 0.5, 1, 2, 4 and 8 meters to the side of the muzzle. The transducers were placed at 90° incidence and tested individually starting at 8 meters.

The results indicate that at the higher pressures the rise time of the capacitor microphones became longer. Some exhibited rise times as long as 200 μsec and measured pressures of 167 dB when in actuality the rise time was less than a microsecond and the pressure was 180 dB.
Transducer Techniques

The design and operation of a capacitor microphone are such that the output signal is proportional to diaphragm displacement when displacements are small. Measuring high pressures forces the diaphragm into relatively large displacements. Then the diaphragm does not move linearly and the rise-time capability deteriorates. The B&K 4136 began to exhibit rise-time deterioration at 170 dB. The BRL 250-kc piezoelectric transducer did not show this non-linearity and consequent rise-time deterioration; it rose to a peak in less than 10 μsec at all pressures tested.

For acoustical transients such as those produced by small arms which have positive pressure durations in the order of 200-300 μsec and rise times of less than 1 μsec, assuming there is a linear decay, the percent error can be written as:

\[
\text{Percent error} = \frac{\text{Tr}}{\text{Td}} \times 100
\]

where: Tr = rise time measured by the transducer
Td = duration of the transient.

Therefore if a transducer’s rise time is 10 μsec the error when measuring small arms will be 3.3 to 5.0 percent (about 0.3 to 0.5 dB).

In summary, our recommendations for measuring small-arms’ pressure waves are:

a. Use a transducer which has a rise-time capability of ten microseconds or less at the pressure being measured.

b. Transducer ringing and overshoot should be less than 1.5 dB at the pressure being measured.

c. The transducers used should have (a) enough sensitivity to allow a signal-to-noise ratio of 25 dB or greater, and (b) minimum drift caused by temperature instability.

d. In relation to the weapon, the transducer should be where the left ear of a right-handed firer would be (firer not present). It should be oriented (a) at 90° incidence, and (b) with its sensitive surface approximately parallel to the ground (Fig. 3).

Fig. 3. Recommended Transducer Orientation for Measurements Made at the Operator’s Left Ear Position.
ARCHITECTURAL ACOUSTICS

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Construction and Test of a "Deutlichkeit" Measuring Equipment.

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Institut für Fernmeldetechnik

Besides the parameter syllable articulation which is to be found out subjectively and the objectively measureable parameter reverberation time since a few years also the definition Deutlichkeit is used to characterize the acoustic conditions of a room.

Fig. 1 shows the connection between syllable articulation and reverberation time - found out by experiments - of rooms in which the reverberation follows an exponential function, i.e. that it is given by the equation

\[ I = I_o \exp(-k \cdot t) \]  

(I = sound intensity, \( k \) = constant, \( t \) = time)

Curve 1, Fig. 2, shows the sound intensity differences \( \Delta p \) between direct sound and echos, depending upon the delay time \( \Delta t \) of the echo which leads to the same sound impression. Curve 2 shows the values of \( \Delta p(\Delta t) \) at which there is no more noticeable interference caused by the echos.

In the range \( \Delta t = 8 \div 32 \text{ ms} \) the echos
Construction and Test of a "Deutlichkeit" Measuring Equipment.

may be more intensive than the direct sound without being noticeable.

In the case of equal levels of direct sound and echoes ($\Delta p = 0$) the connection between syllabic articulation $S$ and the echo's delay time $\Delta t$ is given in Fig. 3 valid for a speech velocity of 5.3 syllables per second.

The parts of the sound reaching the listener with only a small delay time increase the loudness but do not disturb. They are ranked as a useful part of the total sound. According to the experimental results shown in Fig. 1 to 3 its duration generally is supposed to be 30 to 60 msec.

The ratio of the useful sound energy to the total sound energy gives a roomacoustical parameter which better takes into consideration the details of the reverberation process than the parameter reverberation time.

In the definition Deutlichkeit $D$ by Thiele the duration of the useful sound is assumed to 50 msec:

$$D = \frac{\int_{0}^{50\text{ms}} I \, dt}{\int_{0}^{\infty} I \, dt}$$  \hspace{1cm} (2)

With an exponential decay process one gets:

$$D = 1 - \exp\left(0.05 \frac{\ln 10^{-6}}{T}\right)$$  \hspace{1cm} (3)

Fig. 4 shows the course of the Deutlichkeit, calculated from eq. 3, and the measured values of the syllable articulation versus reverberation time $T$. This gives a connection between the syllable articulation $S$ and the objectively measurable Deutlichkeit $D$.

For the judgement of the conditions in acoustically disadvantageous rooms and the judgement of the variable quality of different places in a room the Deutlichkeit is better suitable than the reverberation time.

With $I \sim p^2$ ($p = $ sound pressure) and $u \sim p$ ($u = $ output voltage of a measuring microphone) one gets

---E-2---
Construction and Test of a "Deutlichkeit" Measuring Equipment.

\[ D = \frac{\int_0^{50 \text{ms}} u^2 \, dt}{\sqrt{\int_0^{\infty} u^2 \, dt}} \]  \hspace{1cm} (4)

For this a direct reading measuring equipment can be constructed as an analog computer as shown in Fig. 5.

**Fig. 5**

The direct sound and the echos generated by a puls generator are converted by means of a measuring microfon into a voltage proportional to the sound pressure. This voltage is squared within the Deutlichkeit measuring equipment (both half-waves).

With the arrival of the first puls a main control unit opens two analog gates for the duration of the both integration times. Therefore at the outputs of the integrators appear voltages which correspond to the numerator and denominator of eq. 2. By dividing these both voltages one gets an output voltage proportional to the Deutlichkeit. For reading can be used pointer-type instruments, digital voltmeters or printers.

Before the transmission of the acoustic pulse the main control unit automatically resets all computing units.

The different operations are done by close looped integrated operational amplifiers (Fig. 6).

For squaring \( Z_1 \) is represented by two chains of 10 biased silicon diodes each which yield a quadratic characteristic curve and \( Z_0 \)
Construction and Test of a "Deutlichkeit" Measuring Equipment.

by an ohmic resistor.

With the integrators $Z_1$ is an ohmic resistor and $Z_o$ a capacitor. Of course, the integrator $Z$ is not allowed to integrate for an infinite time. Depending upon the magnitude of the reverberation time integration times of 1 to 5 sec can be chosen. With correct adjustment the error caused by the shortened integration stays below 0.5%. With longer integration the error will be greater caused by integration of noise.

For division one forms the logarithms of the both integrator output voltages. Then the difference of these both logarithms will be delogarithmed and this represents the result.

To logarithm and to delogarithm diods with logarithmic characteristic curves are used for $Z_1$ and $Z_o$ respectively.

The pulse generator should generate an acoustic pulse short compared with 50 msec with a spectrum similar to that of the human voice.

In Fig. 7 are shown the spectra of a 6 mm-pistol's crack and of a spark gap discharging 250 Wsec within 0.1 msec. The broadband measurements amounted to a sound level of 142 dB$_{SL}$ for the pistol and to 133 dB$_{SL}$ for the spark. These pulses are very well reproducible in respect to the loudness as well as to the spectrum (1 ± 2 dB difference).

A spark gap has the advantage that it can be triggered by the measuring equipment but the spectrum is less favorable.

With an exact exponential decaying voltage the Deutlichkeit measuring equipment can be checked and calibrated according to eq. 3.

The values of $D$ measured with the equipment described above can be given with an error of less than 10%.

As the measurements are fast to carry out it is easy to investigate many points of a room. With simultaneous observation of the direction of the incidence of strong echos with a directional microphone and an oscilloscope it is possible to determine the optimal places for absorbing or reflecting materials.
A Versatile Artificial Reverberator

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It has been desired to realize an artificial reverberator which can be controlled easily by the acoustical parameters in room acoustics. Several of these parameters are reverberation time, reverberation time-frequency characteristics and amplitude-frequency characteristics. Electronic reverberator with multi-channel delays may be suitable for this purpose. One promising method in this direction is to utilize the system of all-pass type, invented by M.R. Schroeder, because the reverberation time and amplitude-frequency characteristics are able to be controlled independently.\(^1\) Recently, an extension of the above system has been proposed by H. Date et al. whose frequency characteristics of reverberation time can be controlled independently.\(^2\) Unfortunately, however, numerous phase equalizers are necessary. This paper proposes an improved method without the shortcoming mentioned above and describes a reverberator which is constructed by the method with a delay device consisting of FM-recording magnetic drum.

Principle

Fig.1 shows the block diagram of the system. Its reverberation time is given by

\[ T = T_0 T(\omega) = (60/20 \log \varepsilon_0 \cdot |g(j \omega)|) \tau, \]  \hspace{1cm} (1)

where \( T_0 = (60/20 \log \varepsilon_0) \tau. \)
The transfer function is given by
\[ H(j\omega) = -e^{j(\omega t + 2\beta)} \frac{1 - g_e j\omega}{1 - g_e j\omega e^{-j(\omega t + \beta)}} \]  \hspace{1cm} (2)

where \( \beta \) is the phase of \( g(j\omega) \). Since \( |H(j\omega)| = 1 \), the system is one of all-pass networks.

As seen from Fig.1, maximum reverberation time \( T_o \) and normalized frequency characteristics \( T(w) \) can be independently controlled respectively, because the circuit elements, such as \( g(j\omega) \), \( g^2(j\omega) \), and \( e^{2j\beta} \) which determine \( T(w) \) are separated from the circuit elements \( g_o \) and \( g_o^2 \) which determine \( T_o \). This is a very convenient property to design an analog reverberator. Furthermore, the circuit elements in Fig.1 can be easily synthesized by \( L, C, R \) and operational amplifiers as shown in Table 1. The reverberation time-frequency

<table>
<thead>
<tr>
<th>Frequency characteristics of ( g(j\omega) )</th>
<th>Transfer function of ( g(\lambda) )</th>
<th>Networks</th>
<th>Symbols</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Fall at higher frequencies</strong></td>
<td>( \frac{1 + \lambda f_1}{\lambda f_1} )</td>
<td>( R_1, C_1 )</td>
<td><strong>Buffer Amp.</strong></td>
</tr>
<tr>
<td><strong>Approach to constant values at higher and lower frequencies</strong></td>
<td>( \frac{(\lambda f_1 - \lambda f_1)}{1 + (\lambda f_1)^2} ) ( S &lt; 1 )</td>
<td>( R_1, C_1 )</td>
<td><strong>Two outputs Amp (+1, -1)</strong></td>
</tr>
<tr>
<td><strong>Single resonance at some frequency</strong></td>
<td>( \frac{\lambda f_0}{\lambda f_1} ) ( \lambda = \omega / \omega_0 ) ( \omega_0 = (1/2)(R/L) )</td>
<td>( R_1, C_1 )</td>
<td><strong>Differential Amp Gain = 1</strong></td>
</tr>
</tbody>
</table>

Table 1 Realization of circuit elements for various frequency chs.
A Versatile Artificial Reverberator

characteristics in Table 1 cover most of familiar characteristics.

Developments

An analog artificial reverberator was constructed for broadcasting purpose. It consists of tandem connection of five all-pass network units. Their delay time are chosen so as to avoid periodic echos. The loop gains of the first three units with longer delay times can be controlled in terms of $T_0$ and $T(w)$. The gains of the last two units with delay times less than 30 ms are fixed to 0.7.

The most difficult problem was to design the delay device. Required performance of the device is as follows. (a) Maximum delay time is about 150 ms for simulating large halls. (b) Both gain stability and deviation of frequency characteristics are less than $\pm 0.1$ dB in order to make the error of decay rates less than $\pm 10\%$ and to make deviation of frequency characteristics less than $1$ dB. (c) The dynamic range of each channel is more than 74 dB in order to have 60 dB of overall reverberator. In order to fulfill these requirements, a magnetic drum with five channels is developed. Owing to both fine mechanical structure and FM recording, these requirements are completely satisfied.

The specifications are shown in Table 2. The loop gain $g_o$ is adjustable from 0.5 to 0.8, and delay time are controllable by changing drum speed within the region ($l$) of 0.75 to 1.5 times of standard values: eg, 100 ms etc. Therefore, maximum reverberation time is variable from 0.5 to 4.5 sec. Each frequency characteristics of reverberation time

![Fig.2 Equipped frequency characteristics of reverberation time](image-url)
A Versatile Artificial Reverberator

Reverberation time shown in Fig.2 can be controlled by parameters in terms of characteristic frequency and ratio of maximum to minimum reverberation time.

An example of frequency dependent reverberation curves and amplitude frequency characteristics is shown in Fig.3 and 4 respectively.

Conclusion

An analog artificial reverberator was constructed by the principle newly proposed, which assures independent control of various room parameters. This apparatus can serve as not only a sound control equipment in broadcasting but also a mean to study psychological effects of physical parameters.

References


3) H. Date, et al. NHK Laboratories Note, No.110, (1967).

| Reverberation Time | 0.5 ~ 4.5 sec |
| Delay Time Type (A) | 100, 60, 42, 244, 64 ms |
| Type (B) | 75, 52, 36 |
| Variable Scale of Delay Time | 0.85, 0.75 |
| Loop Gain \( g_0 \) | -2, -3, -3.3, -3.7 |
| Frequency Range | 4.3 ~ 5 ~ 6.0 dB |
| Amplitude-Freq. chs. | 50 ~ 8000 Hz |
| S/H | ± 1 dB |
| FM Carrier Frequency | 600 MHz |
| Table 2 Specifications |
| \( H = 2 kHz \) |
| \( C : A \) |
| \( g_0 = 0.7 \) |
| \( \alpha = 6.3 Hz \) |
| \( \alpha f = 25 Hz \) |

Fig.3 One example of frequency dependent reverberation curves. (\( \alpha : \) Modulation Frequency \( \alpha f : \) Modulation Swing)

Fig.4 Amplitude frequency characteristics observed under the same condition as fig.3
Vorwort.

und bis zum Ende der Messung ist eine Beaufsichtigung nicht erforderlich. Die Vollendung dieses Meßgerätes vermeidet viel Zeitaufwand bei der Nachhallzeitmessung im Hallraum.

Skizze und das Prinzip der Meßmethode des oben erwähnten Nachhallzeitmessers.


1. Der anwendbare Frequenzbereich ...von 100 Hz bis 5000 Hz.
2. Die Meßsignale ...Terz-Oktave Rauschen.
3. Die meßbare Nachhallzeitlänge ...von 1,00 s. bis 99,99 s.
4. Die Pegelunterschiede der gemessenen Nachhalldauern ...20 dB oder 30 dB.
5. Die Häufigkeit der Wiederholung ...10, 20, oder 40 Male/je an jedem Frequenzband.
6. Die Zeitdauer, während der die Schallquelle zum Tönen gebracht wird. ...3 s. oder 6 s.
7. Der Eingangswiderstand ...600 Ohm.
8. Die Angabeform der Meßergebnisse ...Die Zahl bis zu 2 Stellen hinter dem Punkt. (z.B. 12,34 sec)

Vor der Messung braucht man die Kontrollknöpfe der obigen Abschnitte 4), 5) und 6) nur einzustellen, und das Meßergebnis kommt aus der Druckmaschine zusammen mit der Mitteffrequenz des Meßsignals. (Abb.2)


Hingegen dient der 2. Verstärker zum Auffangen der Schallpegelunterschiede (20 dB oder 30 dB), um die den Schallpegel nach der Ausschaltung der Schallquelle in Hallraum abfällt. Das Tor 1 öffnet sich, wenn die Pegelhöhe unter einen bestimmten Pegel (0,5, 10, 15 oder 20 dB niedriger als der Pegel, an dem die Schallquelle ausgeschaltet
wurde.) sinkt, und schließt sich, wenn der Pegel auf 20 dB oder 30 dB abgefallen ist, indessen werden die Impulse für die Zeitausrechnung vom Quarzoszillator aus an den Impulszähler weitergeleitet.

Verstärker kontrolliert mit Hilfe des Regulators, der sich am Ausgang des Rauschoszillators befindet, den Schallpegel im Hallraum. Sieb 1 und Sieb 2 sind Terzoctave-Siebe, die von 100 Hz bis 5000 Hz veränderbar sind, und parallel getrieben werden; wenn die Messung auf einem Frequenzband vollendet ist, so schalten sich die Kontakte der beiden Siebe auf das nächste Band um. Das Tor 2 schaltet die Schallquelle aus und ein. Verstärker ist der Verstärker für den Lautsprecher.

Der Impulszähler summier die Zeitimpulse, die von Tor 1 abgeteilt werden. Nach Beendung der Messung werden die summierter Impulse an die Druckmaschine geschickt. Von der Druckmaschine werden die eingesandten Impulse als Durchschnittswert gedruckt. Die Funktion dieses Geräts und das Messverfahren wird in Abbildung 4 gezeigt.

Die Behandlung des Apparates.

Um die Stabilität und Zuverlässigkeit des "Automatischen Digital-Nachhallzeitmessers" zugewährleisten, wurde ihm ein Stromkreis mit vielen Transistoren zu Grunde gelegt. Wie oben erwähnt ist die Behandlung sehr einfach. Am Anfang der Messung muß man die folgenden Punkte einstellen: den Pegel, an dem Tor 1 sich öffnet (0,5,15 oder 20 dB), den Pegelunterschied zu messenden Nachhalldauern (20 oder 30 dB), die Häufigkeit der Wiederholung jeder Messung (10,20 oder 40

— E-11 —
Male), die Schallquellenzeitdauer einstellen und das Messfrequenzband wählen (aus 18 Bändern von 100 Hz bis 5000 Hz). Darauf legt man das Registerrolle in die Druckmaschine ein. Wenn man den Knopf von dem Messfangsignal ("START") drückt, so beginnt die Nachhallszeitmessung, die vollautomatisch bis zum Ende der Messung fortläuft. Nach Beendigung der letzten Messung ertönt ein Summer. Dann kann man eine Reihe von Nachhallzeiten auf dem Registerpapier, wie links gezeigt wird, finden.

**Abb. 2.** Gedrucktes Messergebnis aus einem Hallraum.

**Abb. 3** Blockschaltbild des Apparates.

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**Nachwort.**

On a Simplified Hybrid Analog-Digital Correlator for Sound Measurements

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Introduction

In recent years, correlation techniques have been useful in acoustic signal processing. To utilize these techniques in practice, electronic correlators are employed. Generally, correlators have three basic parts: time delaying circuit, multiplying circuit and averaging circuit. A number of attempts have been made to realize correlators with simple circuits.

If inputs are Gaussian-, random-phase periodic- or binary signals with zero mean, a simplified hybrid analog-digital correlator (relay correlator) is available. This correlator may be widely used in acoustic signal processing, since signals occurring in such processing have properties almost same as Gaussian-, random-phase periodic- or binary signals. Although this correlator is not available to signals with arbitrary probability distribution, it can be used for these signals, by means of adding a uniform distribution noise to only one of the two signals.

Simplified hybrid analog-digital correlator

For the correlation function of two signals $x(t)$ and $y(t)$, the following estimate $R_{xy}^{\tau}(t)$ is usually considered:
On a Simplified Hybrid Analog-Digital Correlator

\[ R_{xy}(\tau) = \frac{1}{T} \int_{t-\tau}^{t} x(t) y(t-\tau) \, dt. \]

However, in many cases, a simplified estimate \( R_{xy}^s(\tau) \) may be used:

\[ R_{xy}^s(\tau) = \frac{1}{T} \int_{t-\tau}^{t} x(t) \text{sgn} \, y(t-\tau) \, dt. \]

A correlator circuit based on the simplified estimation is shown in Fig. 1. As shown in Fig. 1, relatively expensive analog delay and multiplier circuit can be replaced by simple digital circuit and analog switch.

![Circuit Diagram]

**Fig. 1** Simplified hybrid analog-digital correlator

For Gaussian signals \( x(t) \) and \( y(t) \) with zero mean, the simplified estimate \( R_{xy}^s(\tau) \) is proportional to true correlation estimate:

\[ R_{xy}^s(\tau) = \frac{\sqrt{2}}{\pi} \frac{1}{\sigma_y} \, R_{xy}(\tau). \]

Similar formula

\[ R_{xy}^s(\tau) = \frac{K}{\sigma_y} \, R_{xy}(\tau). \]

can be derived for random-phase periodic- or binary signals.

If one signal to be discriminated its polarity is the signal \( y(t) \) with uniform distribution noise \( u(t) \), the simplified estimate \( R_{xy}^s(\tau) \) for arbitrary signals \( x(t) \) and \( y(t) \):

\[ R_{xy}^s(\tau) = \frac{1}{\alpha} \, R_{xy}(\tau) \]

\[ = \frac{1}{\alpha T} \int_{t-\tau}^{t} x(t) \text{sgn} \{ y(t-\tau) + u(t-\tau) \} \, dt \]

where \( \alpha \) is a constant in probability density \( 1/2\alpha \) (\( |u| < \alpha \)) of the noise \( u(t) \).
Statistical error of estimate

Correlation function is originally defined by infinite-time average. In actual, however, the correlation function must be estimated by finite-time average. Therefore, the correlation estimate has some statistical error.

The variance of the simplified estimate \( R_{xy}^s(\tau) \) for Gaussian signals \( x(t) \) and \( y(t) \) is represented by

\[
\text{Var} \ R_{xy}^s(\tau) = \frac{2 \sigma_x^2}{T^3} \int_T^T (1 - \frac{\theta}{T}) \left\{ \sigma_x^2(\theta) \sin^2 \theta R_{xy}(\theta) + (1 - R_{xy}(\theta))^2 \left\{ \frac{1}{\tau} \right\} R_{xy}(\tau) \right\} d\theta - \frac{2 \sigma_x^2}{T} R_{xy}(\tau)
\]

where \( \rho \) represents the correlation coefficient.

When the two signals identical and Gaussian, and \( R_{xx}(\theta) = \exp(-|\theta|/\tau_x) \), the following inequality is satisfied:

\[
\frac{T_x}{T} (1 - e^{-\frac{T_x}{T}}) \leq \frac{T}{2 \sigma_x^2} \text{Var} \ R_{xy}^s(\tau) \leq \frac{T_x}{2 T} (1 - e^{-\frac{T_x}{2 T}})
\]

Fig. 2 shows the variance of the normalized simplified estimate \((T/2 \sigma_x^2) \text{Var} \ R_{xy}^s(\tau)\) and the variance of the normalized conventional estimate \((1/\sigma_x^2) \text{Var} \ R_{xx}(\tau)\).

The variance of the simplified estimate \( R_{xy}^s(\tau) \) is given by

\[
\text{Var} \ R_{xy}^s(\tau) \leq \frac{1}{\tau_x^2} \text{Var} \ R_{xy}(\tau) + \frac{2 \sigma_x^2}{T^3} \left( 1 - \frac{T_x}{T} \right) \left\{ \sigma_x^2(\theta) \rho_{xx}(\theta) \right\} d\theta,
\]

where \( \rho_{xx}(\theta) \) is the auto-correlation coefficient of the Gaussian noise \( n(t) \) which is used to realize a uniform probability distribution noise \( U(t) \).

When the two signals are identical and Gaussian, and \( \rho_{xx}(\theta) = \exp(-|\theta|/\tau_x) \),
On a Simplified Hybrid Analog-Digital Correlator

\[ P_n(\theta) = \exp\left(-|\theta| / \tau_n\right), \]
the following relation is obtained:

\[ \frac{Q^*}{6 \tau} \sum_{n=1}^{\infty} r_n^* \leq \frac{T_s}{T} \left(1 + e^{-\frac{T_s}{T}}\right) + \frac{Q^*}{6 \tau} \frac{T_s T_n}{T(T_s + T_n)} . \]

Fig. 3 shows the variance of the simplified estimate \( r_n^* \). (\( \tau \)).

Applications

1) Measurement of sound attenuation in panels

As shown in Fig. 4-A, two microphones \( M_1 \) and \( M_2 \) are placed on each side of the panel under test a loudspeaker is used. Let \( \alpha(t) \) be attenuation coefficient, there is the following relation:

\[ \alpha(\tau) \otimes r_n^* (\tau - \tau) = r_n^* (\tau) . \]

From this equation, the attenuation coefficient can be obtained.

If the panel can be removed, the system of measurement as shown in Fig. 4-B may be applied. When the input voltage to the loudspeaker is binary noise or M-sequence,

\[ \alpha(\tau) \otimes r_M^* (\tau) = r_M^* (\tau) . \]

Similarly, measurement of sound absorption on panels can be performed. For these measurements, an ordinary room may be used.

2) Analysis of noise

3) Detection of periodic signals in random noise

Conclusion

The simplified estimate will be fairly good estimate of the correlation function. The correlator based on this simplified estimation is realized by using simple hybrid analog-digital circuits and it has high accuracy. This correlator may be widely used in acoustic signal processing.
Measurement on the edge phenomena of finite sound absorbing panels, using a new method of wave form measurement

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Introduction

The absorbing coefficient of an acoustical panel normally varies according to the size of the panel area. This phenomenon is known as Area Effect, and depends upon the diffraction at the edge of the panel.

To study this phenomenon, it is necessary to measure the reflected sound wave forms at an arbitrary point on the panel in the case of an arbitrary incident wave. For this purpose, the authors propose a new method using the cross-correlation function, which is equivalent to the pulse method, but more accurate. As this method gives very stable data, the reflected sound waves close to the surface of a panel can easily be obtained by subtracting the incident waves from the compound waves consisting of incident and reflected waves.

Principle and system of measurement.

We used a noise sound source. Evidence that the cross-correlation function shows the same behaviour as a pulse follows: $f_2(t)$ and $n_2(t)$ denote respectively the output waves of an acoustic system corresponding to impulse input $f_1(t)$ and noise input $n_1(t)$. Then,
Measurement on the edge phenomenon

\begin{align*}
  f_2(t) &= \int_{0}^{\infty} k(\omega) f_1(t - \omega) \, d\omega \\
  n_2(t) &= \int_{0}^{\infty} k(\omega) n_1(t - \omega) \, d\omega
\end{align*}

where \( k(t) \) is the impulse response of the acoustic system.

Introducing some reference noise wave \( n_0(t) \), which is actually the driving voltage of the sound source, we considered two cross-correlation functions, \( \varphi_{10}(\tau) \) between \( n_1(t) \) and \( n_0(t) \), and \( \varphi_{20}(\tau) \) between \( n_2(t) \) and \( n_0(t) \). They are as follows.

\begin{align*}
  \varphi_{10}(\tau) &= \lim_{T \to \infty} \frac{1}{2T} \int_{-T}^{T} n_1(t + \tau) n_0(t) \, dt \\
  \varphi_{20}(\tau) &= \lim_{T \to \infty} \frac{1}{2T} \int_{-T}^{T} n_2(t + \tau) n_0(t) \, dt
\end{align*}

Substituting Eq. (2) with Eq. (4) and comparing the resulting equation with Eq. (3), Eq. (4) can be written as

\[ \varphi_{20}(\tau) = \int_{0}^{\infty} K(\omega) \varphi_{10}(\tau - \omega) \, d\omega \]

Eq. (5) has the same form as Eq. (1). Namely, it shows that the outputs \( f_2(t) \) and \( \varphi_{20}(\tau) \) have the same wave forms, when inputs \( f_1(t) \) and \( \varphi_{10}(\tau) \) have the same wave forms. In this study, input corresponds to a reflected wave or to a compound wave consisting of incident and reflected waves.

Fig. 1 shows the block diagram of the system of measurement based upon the principle mentioned above. As a noise source, a maximal-length linear sequence having a longer repeating period than the reverberation time of the measurement room was used.

![Fig. 1 Block diagram of the system of measurement](image)
Measurement on the edge phenomenon

This method has the following advantage:

1. Even if the power of the speaker is comparatively small, the measurement is not disturbed by outside noise, because it has no correlation with the reference noise.

2. The use of a maximal-length linear sequence, one of binary signal, simplifies electronic circuits, such as a multiplier or a delay circuit. Therefore, a shift register can be used as delay circuit, and the delay time can be set accurately, using a frequency counter.

3. The use of a maximal-length linear sequence ensures high stability of sound wave amplitude, because the amplitude of the driving voltage for the speaker can be kept constant, using a limiter and other similar devices.

Measurement Results

The purpose of this measurement is to obtain the reflected wave forms from the acoustical panel for normal incident wave. The panel is composed of a glass wool subpanel and a hard board subpanel as Fig. 2 and is placed at the center of an anechoic room. We have obtained the reflected wave forms by substracting an incident wave from compound waves. The incident wave can be measured by taking the panel out of the room. Reflected sound pressure wave forms from A - C parts of the panel in Fig. 2 are shown in Fig. 3-A, 3-B, 3-C.

Early parts of the waves shown in Fig. 3-C, which is at the central part of hard board sub-panel, have good agreement, because no influence from the edges reaches the early parts.

By the use of this method, it is possible to obtain the oblique incident characteristics of infinite panels. These are almost impossible to obtain with the former methods.

Sound energy absorbed by edge phenomena.

Using the data obtained by the above method the authors calculated the reflected sound energy, as functions of time and the position on the panel. Fig. 4 shows the
reflected sound energy during the early period, about 0.5 milli-seconds, as a function of position on the panel. The curve in Fig. 4 show the existence of remarkable edge effect. The authors intend to verify this phenomenon through continued experiments.

Fig. 2, The acoustic panel

Fig. 3-A, Reflected sound waves at A part in Fig. 2

Fig. 3-B, Reflected sound waves at B part in Fig. 2

Fig. 3-C, Reflected sound waves at C part in Fig. 2

Fig. 4, Reflected sound energy from the acoustic panel
Sound Dissipation of Different Profile Surfaces.

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In order to increase a sound field diffusity in halls walls and ceilings are usually employed. Their cross-sections have mostly invariably profile in one direction and may be considered as rectangular planes with long bars on them. Up to the present there has been no convenient coefficient characterizing the efficiency of the scattering action of such surfaces.

It may be expedient to judge the scattering property of the surface by deviation of the energy distribution in indicatrix of diffusion (echoing characteristic by power) of a given profile from the energy distribution of the uniformly scattering surface equivalent to a given one. In orthogonal coordinates where degrees are plotted on abscissa and energy on ordinate the echoing characteristic of the uniformly scattering surface is presented by horizontal line. An area cut off by said line on the echoing characteristic of the structure analysed is proportional to irregularly scattered energy (Fig. 1). The ratio of the irregularly diffused energy to the whole scattering irregularity $K_{si}$ may be expressed in general as

$$K_{si} = \frac{\int_{\theta} E_c(\theta) \, d\theta}{E_c \theta_o}$$
Sound Dissipation of Different Profile Surfaces

where: $E_e(\theta) = E(\theta) - E_u$, $E(\theta)$ - value of energy measured in the echoing characteristic, $E_u = \frac{E(\theta)}{\pi}$ - value of energy from uniformly scattering surface, $E_0$ - measured value of energy reflected from a lengthy plane in the region of regular reflection, $\theta_0$ - angle corresponding to regular (geometric) reflection in diffusion indicatrix from the plane equivalent in width to the surface investigated.

Therefore, $E_e(\theta)$ is a local excess of the measured value of energy over the energy value $E_u$ of the uniformly scattering surface. $K_{Sl} = 1$ - for limited width plane and $K_{Sl} = 0$ - for uniformly scattering surface.

In order to calculate $K_{Sl}$ one must determine the echoing characteristics from the model investigated; then with the help of these characteristics the area above $E_u$ level is to be found. The value $E_0$ is measured then by substituting the investigated surface with a wide plane. In order not to take into account the limited dimensions of the surface the measuring of characteristics is made from the rectangular surface of sufficient length. The width of surface is accepted equal to 2-5 Fresnel regions which is optimum from the standpoint of measurements and simplicity of calculation. When the width of the measured surface is over optimum the accuracy of calculation of numerator $K_{Sl}$ decreases. When the width of a plane is less than a half of the Fresnel region the plane by itself possesses the scattering effect and its $K_{Sl}$ is less than 0.9 (in this case it is necessary to take into account the limited width of the plate).
Sound Dissipation of Different Profile Surfaces

Fig. 2 Factor $K_{Si}$ for the surfaces with rectangular bars.

Fig. 3 Factor $K_{Si}$ for structures with semicylindric bars.

In order to simplify the problem the echoing characteristic of the plane may be considered as a rectangle received on the base of ray principle. According to this concept the energy reflected by plane is proportional to product $6h$.

To calculate $K_{Si}$ the measuring were made by means of equipment intended for measuring the echoing characteristics by pulse-wise. The angle of arrival of sound waves was constant and equal to 30°; the width of plane was 60cm and the distance from it to point-source and microphone were 1,85 m and 1,55 m accordingly. The measurements were made in bandwidth some less than 5-10kHz and 10-20kHz octaves. In order to get the stationary picture of reflection the duration of the radiated pulse (5-7ms) was taken on the one hand sufficiently long, but on the other hand the said duration was sufficiently short to avoid troublesome echoes. From the stable part of the reflected pulse a more short signal (0,5-2ms) was cut out which was recorded by level recorder. Samples per second were constant and equal to 50 c/s frequency of the net. The dependence of $K_{Si}$ on ratio of structure step to the wave length ($d/\lambda = 1+24$) and width (height) of bar to the structure step ($b/d = 1+7/8$) was investigated.
In fig. 2 the dependence of $K_{SI}$ upon $d/\lambda$ and $h/\lambda$ for rectangular profile with ratio of sides 1:2 and 1:1 is shown. The diagrams show that structures with vertical rectangular bars give the best results. The best sound scattering is observed for ratios $d/\lambda = 2 + 4$ depending on $b/d$. For $b/d = 1/2$ coefficient $K_{SI}$ has the least value when $d/\lambda = 2$; for $b/d = 1/4$ corresponding least value of $K_{SI}$ occurs at $d/\lambda = 4$.

For profiles with semicylindrical bars and ratios $b/d = 1:1/2; 1/4$; and $1/6$ the dependence of $K_{SI}$ is shown in fig. 3. It may be considered that at $b/d = 1$ the structure gives nearly ideal sound scattering (when $K_{IS} < 0.1$ the accuracy of measuring is low). The profile with segment bars and $b/r = 1.5$ provides the sound scattering a little different from semicylindrical profile. With decreasing of $b/r$ rapid change of the uniformity of scattering (especially sharp from $b/r = 1$ to $b/r = 0.75$) for the worse takes place.

In fig. 4 the dependence of $K_{SI}$ for different saw-tooth profiles is shown. The structure of this kind as might be expected possesses the small scattering properties and echoing characteristics only slightly displaced the regular reflection. When sound is directed against the crest of saw-tooth surface the scattering of sound is increased. For segment-saw-tooth surface the scattering is defined by the segment profile.

Reference:
Reflection Characteristics
of Diffusing Walls
Teruji Yamamoto
NHK Technical Research Laboratories,
Tokyo, Japan.

Introduction
In most large studios and halls, many sound diffusing walls have been used in order to obtain a certain diffuse sound field in the room and to prevent the geometrical reflections of sound which often lead to greatly delayed echoes. But, quantative data for the acoustical designing of diffusing walls are few. This paper reports the reflection characteristics of single projecting surfaces such as hexagonal and cylindrical walls, and of acoustically transparent grid- and slit-walls.

Test surfaces
Table I gives dimensions and details of the test surfaces.

Methods of measurement
The measurement was made by scale-models of the walls in an un-echoic room. The schematic arrangement for the measurement is shown in Fig. 1. As to hexagonal and cylindrical walls, the test surface was set at the center of the room. On the other hand, as to grid and slit walls, a partition wall with a rectangular aperture of 20x20 cm\(^2\) was set in the room and the test surface was mounted in the aperture. A sound source was placed at a...
distance from the test object $s$ at $\theta_1$ to an axis normal to the test surface. A microphone was fixed to a long movable rod of $r$ meters in length so that it could be moved around the test object. The reflected sound contribution was measured as a function of the angle of incidence $\theta_1$, the angle of reflection $\theta_r$, and of the frequency. In both measurements, pure tone impulses having a duration of 5 ms were used.

Results and discussion

(1) Single projecting walls

(A) Reflection directivities: First, as a fundamental experiment, directional reflection characteristics of a rectangular reflecting panel were measured. These characteristics closely agreed with the theoretical values below 20 kHz.

Fig. 2 shows measured reflection directivities of the hexagonal diffusing surface. The reflection characteristics depend on the effective reflecting area, the setting angle of the object and on frequency. These results can be summarized in the following way.
a) When a side length of the wall is of the order of a wavelength, the hexagonal diffusing wall can be regarded as a circumscribed spherical surface about the polyhedron. Therefore, the reflection level can be calculated from the dimension and the curvature of the wall.
b) If the side length exceeds about 4 wavelengths, the sound tends to reflect at specular angles, where the specular angles are determined by the normal to each surface of the polyhedron. The reflection sound pressure level is approximately proportional to the area of each surface.

As to the cylindrical surface, the reflection level diminished according as the extension of the reflection domain produced by the cylindrical surface became large. The measured values agree with the values calculated from the area and the curvature of the wall.

On the basis of these results, it can be concluded that the reflection directivities of single projecting walls can be calculated from diffraction theory.

(B) Sound pressure reflection coefficient $R_f$. In the paper, we define the $R_f$ of single projecting walls as the ratio of the pressure of sound reflected from the wall to that reflected from an infinite rigid wall. From the above mentioned results, this $R_f$ is given as a function of

![Fig. 3 Relation between the sound pressure reflection coefficient $R_f$ and the ratio of the area of the wall to the wavelength $(r; 1.0 \text{ m}, s; 2.0 \text{ m})$.](image)

![Fig. 2 Reflection directivities of the hexagonal wall.](image)

![Fig. 4 Reflection characteristics of the grid walls with $d$ of 5 mm.](image)
the wall area $S$, wavelength $\lambda$, the angle of incidence $\theta_i$, the distance between the sound source and the object $s$ and of the distance between the measuring point and the object $r$. In Fig. 3, the calculated $R_f$ for $\theta_i$ of $0^\circ$, $s$ of 2.0 m and $r$ of 1.0 m is shown. When this value is denoted as $f(S/\lambda)$, the reflection coefficient for optional values of $\theta_i$, $s$ and $r$ is given as $(1/r+1/s)(\cos \theta_i/1.5) f(S/\lambda)$. That is, when the value of the abscissa in Fig. 3 substitutes in $1.5S/\lambda(1/r+1/s)\cos \theta_i$, $R_f$ is given by the solid curve in Fig. 3. As to cylindrical walls, $R_f$ can be given as $k f(S/\lambda)$, where $k$ is a constant to be obtained from the curvature of the walls. As an example, $R_f$ for $\theta_i$ of $60^\circ$ is shown by the dot-dash curve in Fig. 3.

(2) Grid wall

(A) Reflection characteristics: Measured results are shown in Fig. 4. The reflected sound pressure depends on the grid depth. When the grid depth is integer times as large as 1/2 wavelengths, the sound pressure shows the minimum. The frequency, at which the pressure has the minimum, agrees with the value calculated from impedance of the grids. The shapes of reflection directivities above 10 kHz agree with those of a rigid panel with same area. From these results, it is found that reflection coefficients of grid walls are smaller than 0.1 below 10 kHz. As to at the frequencies above 10 kHz, these are at most 0.1.

(B) Transmission characteristics: Measured transmission directivities are shown in Fig. 5. The transmission sound below 10 kHz is independent of the existence of the grids. Its level and directional shape accord with those of the aperture. Furthermore, even if the frequency is 20 kHz, if $w$ is 10 mm wide, which is not so large as a wavelength, its shape approximately agrees with that of the aperture. On the contrary, when $w$ is 20 mm wide, sides of each grid act as reflective walls.

If the ratio of the depth of the grid to the wavelength is smaller than 0.3, the grid wall can be regarded as an acoustically transparent wall.

(3) Slit wall

(A) Transmission characteristics: In Fig. 6, measured transmission characteristics

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Reflection Characteristics of Diffusing Walls

are shown as a function of the ratio of the slit period L to the wavelength \( \lambda \). Furthermore, the transmission directivities of the walls at 10 kHz are shown in Fig. 7.

When the ratio \( L/\lambda \) is smaller than 0.8, almost all sounds transmit through the wall. On the contrary, when the ratio is larger than about 1.6, transmitted relative sound pressure level becomes about -6 dB, which agrees with the value calculated from diffractive theory on condition that the slits only concern with the transmission. The shape of transmission directivities neatly agrees with that of the aperture even if the ratio \( L/\lambda \) is larger than 1 or not, and its relative level only varies. Therefore, it can be concluded that the transmission characteristics of infinite slit walls are equivalent to those of the slit wall shown in Fig. 6.

Reflection characteristics: Measured characteristics are shown in Fig. 8.

As like as the transmission, when the ratio \( L/\lambda \) is larger than about 1.6, the relative reflection level becomes about -6 dB, which agrees with the calculated theoretical value. Therefore, reflection coefficients of infinite slit walls can be estimated from the results of the slits shown in Fig. 8. But, when a slit wall has a rear air space of and order of cm, the reflection coefficient substantially becomes about 1. As an example, Fig. 9 shows reflection characteristics of slit walls with rear air space of 20 mm. Their resonances occur at about 3 kHz. The mean level of reflected sound pressure above 5 kHz agree with the level of reflected sound from a rigid panel with the same area.

![Fig. 7](image1)

*Fig. 7* Comparison of the measured transmission directivities of the slit walls and the theoretical values of the aperture of 20x20 cm.

![Fig. 8](image2)

*Fig. 8* Variation of the reflection level with the ratio of period of the slits to the wavelength.

Conclusion

a) The sound pressure reflection coefficient of projecting surfaces depends on the area, shape, frequency and angle of incidence. The coefficient for optional conditions can be obtained from the results shown in Fig. 3.
b) The coefficient of the grid walls are at most about 0.1.
c) The coefficients of slit walls are proportional to the ratio of the area of slat surface to that of slits if \( L/\lambda \) exceeds about 1.6. At other frequencies, the coefficients are given by the results shown in Fig. 6 and Fig. 8.
This fundamental study on sound reflection from plane panels and a panel array was undertaken to derive a method of predicting the effects of suspended panels or "clouds" on auditorium acoustics.

§ 1  SOUND REFLECTION FROM A RIGID RECTANGULAR PLANE PANEL

1-a. Calculation by the Fresnel integrals

In Fig.1, the origin of the coordinates put on the center of a rigid rectangular plane panel $2a \times 2b$.

The velocity potential of reflected sound from the panel at the receiving point, $\tilde{\Phi}(p)$, was derived from Fresnel-Kirchhoff diffraction formula, with Kirchhoff's boundary conditions as below,

$$\tilde{\Phi}(p) = B \int_S f(\alpha_i, \alpha_r, m, \ell) \, ds,$$

where $B$: constant. When the source and the receiver be in far field,

$$\tilde{\Phi}(p) = B \left[ \cos \alpha_{io} + \cos \alpha_{ro} \right] \int_a^b d\xi \int_b^d d\eta \cdot e^{-ik f(\xi, \eta)},$$

and

$$f(\xi, \eta) = -\frac{Z_0^2}{m_0} - \frac{Z_0^2}{\ell_0} + \frac{\xi^2 + \eta^2}{2 m_0} + \frac{\xi_0^2 + \eta_0^2}{2 \ell_0} - \frac{Z_0^2 \eta_0^2}{2 m_0^2} - \frac{Z_0^2 \eta_0^2}{2 \ell_0^2} + \ldots,$$

where $k$: wave number of sound, and other symbols are shown in Fig.1. The Fresnel integrals required for calculating this Formula were computed by a digital computer.


1-b Measurements. The reflected sound pressure levels from a square panel of glass or marble plate were measured in an anechoic room with pulsed-tone method.

Table 1. Scope of experiments

<table>
<thead>
<tr>
<th>Frequency</th>
<th>3.15 ~ 20 (Kc)</th>
<th>(cm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Side length of a panel</td>
<td>6, 10, 43.5</td>
<td></td>
</tr>
<tr>
<td>Incident angle</td>
<td>0°, 22.5°, 45°</td>
<td></td>
</tr>
<tr>
<td>Distance of source m₀</td>
<td>3.5 (m)</td>
<td></td>
</tr>
<tr>
<td>Distance of receiver l₀</td>
<td>1.0, 1.5 (m)</td>
<td></td>
</tr>
</tbody>
</table>

The receiving microphone was rotated on the semi-circle (Fig.1). The measured values agree closely with the calculated values. An example is shown in Fig.2, refer to the sound pressure at a point distant (l₀ + m₀) from the source in free field.

§ 2 SOUND REFLECTION FROM AN ABSORBING PLANE PANEL

An acoustical reflection-diffusion function \( R_e (\alpha_i, \alpha_r) \) is defined for an absorbing panel. The velocity potential, Eq. (1), can be written as

\[
\tilde{\Phi}(p') = B \int \int f (\alpha_i, \alpha_r, m, \ell) \cdot R_e (\alpha_i, \alpha_r) \, ds. \quad (4)
\]

Considering in far field, \( R_e (\alpha_i, \alpha_r) \) can be regard as a constant and Eq. (4) is written as

\[
\tilde{\Phi}(p') = R_e \cdot \tilde{\Phi}(p). \quad (5)
\]

And when the panel reflects regularly, the sound-pressure reflection coefficients, measured by the interference pattern method, are reasonably used for value of \( R_e \). (See Fig.3.)
§ 3 SOUND REFLECTION FROM A PLANE PANEL OF IRREGULAR SHAPE

Sound Reflection from a panel of irregular shape are estimated by dividing the panel into rectangles, and then summing the effect of each rectangle calculated by the Formula (2) or (5). The smaller dividing the panel, the better approximation is obtained. On the other hand, the Fraunhofer diffraction formula becomes more useful for each divided smaller panels, because it is more simple and more easy to calculate than Fresnel diffraction Formula (2), and it has sufficient accuracy for smaller panel. For the case of it, Eq. (3) becomes very simple, neglecting the second and higher order terms of $\xi\eta$ and $\gamma$ in Eq. (3).

The velocity potential of reflected sound $\Phi_{rj}$ from one of the fractional rectangle, as shown in Fig. 4, becomes

$$
\Phi_{rj} = C_j \left[ \cos \alpha_{io} + \cos \alpha_{ro} \right] \sin \frac{U}{U} e^{ik(m+\ell)}
$$

$$
= X_j + iY_j \quad (j = 1, 2, \ldots, n)
$$

where $C$ : constant, and

$$
U = \frac{k}{b} (\cos \beta_i - \cos \beta_f).
$$

Then, the total velocity potential $\Phi_R$ is

$$
\Phi_R = \sum_{j=1}^{n} X_j + i \sum_{j=1}^{n} Y_j.
$$

The values calculated by this method for a panel, dividing into a number of squares as shown in Table 2, are compared with the measured values in Fig. 5.

<table>
<thead>
<tr>
<th>Side length $2a$ (cm)</th>
<th>number of squares</th>
<th>curve in Fig. 5</th>
</tr>
</thead>
<tbody>
<tr>
<td>8</td>
<td>4</td>
<td></td>
</tr>
<tr>
<td>4</td>
<td>17</td>
<td></td>
</tr>
<tr>
<td>2</td>
<td>64</td>
<td></td>
</tr>
<tr>
<td>1</td>
<td>267</td>
<td></td>
</tr>
</tbody>
</table>

measured values

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Fig. 4. A calculating method for a plate of irregular shape.

Fig. 5. Reflected sound pressure levels from a rigid panel of irregular shape.
The values calculated by the larger number of smaller squares agree more closely with the measured values, though the values calculated by smaller number of squares are likely to be sufficient to agree with measured values in the first lobe of the pattern shown in Fig. 5.

§ 4 SOUND REFLECTION FROM AN ARRAY OF PLANE PANELS

At the experiments, an array of twenty-five square panels shown in Fig. 6 was used. The sound reflection from each panel is calculated by Formula (6). Then, the total sound reflection from the array is calculated by summing each reflection by Formula (7), and compared with the values measured by the pulsed tone method as shown in Fig. 7.

From the practical point of view, good agreement between theory and experiment has been found over a wide range of reflection angles and frequencies, except in lower frequencies.

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**REFERENCE**

1) Born and Wolf; "Principles of Optics" p. 374 ~ 383 (1959)


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Fig. 6 Array of twenty-five square panels

Fig. 7 Examples of the reflected sound pressure levels from an array of rigid square panels. Solid lines: measured, Dotted lines: calculated, and Dashed line: calculated value of a single rectangular panel 435 mm × 435 mm.
The "Interference Pattern Method" of Measuring the Complex Reflection Coefficient of Acoustic Materials at Oblique Incidence

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Kobe University

Introduction

Various methods (1...6) have been presented, of measuring the acoustic properties of materials at oblique incidence.

Those methods, however, require special apparatus or techniques, for instance, it is not easy to make "pure plane wave" in the free field.

Therefore, in the present report, making progress the suggestion of Strutt(7) and Cremer(8), it is more accurately discussed as the "Interference Pattern Method" in a point source field for higher frequencies (1.0-20.0kc/s). In addition, it is shown by means of an example that the method is applicable even for a scattered reflection from rough surfaces.

The method

As shown in Fig.1, a point source is located at s, and the total sound pressure at a receiving point m is the sum of the primary wave and the reflected wave,

* This original paper by the author has presented to the Institute of Electronics and Communication Engineers of Japan for the Transactions on the 4th September, 1967.
\[ \hat{P} = \hat{K} \left[ \frac{1}{d_i} \exp(-jkd) + \frac{\hat{R}}{r} \exp(-jkr) \right] \]
\[ \hat{R} = \hat{R} \exp(-j\varphi) \]
\[ \varphi_n = (2n+1)\pi - k(r_n - d_n), \quad n = 0, 1, 2, \ldots \]
\[ |\hat{R}|_N = \frac{\left[ \frac{\varphi}{j} \right]_{M,N}}{\left[ \frac{\varphi}{j} \right]_{M,N} + 1}, \quad (M=N \text{ or } N+1) \]
\[ |\hat{R}|_N \text{ and } [\varphi]_{M,N} \text{ are usually calculated from Nth pressure minimum and Nth or (N+1)th maximum.} \]

\[ \Theta_n = \tan^{-1} \left( \frac{D}{(H + h_n)} \right) \]

**Measurements**

A schematic diagram of the method is shown in Fig. 2.

It is cleared experimentally that \( |\hat{R}| \) should be taken mean of \( |\hat{R}|_M \) (3 \( N \leq 5 \)) so that the error caused by diffraction tend to negligible small. A demention of material to be tested is shown in Fig. 3. It is better for \( \varphi \) to take \( \varphi_1 \) or \( \varphi_2 \) computed from 1st or 2nd minimum point, for example, as shown in Fig. 4 (see Table 1).

The measured reflection coefficients of a felt (2mm-thickness), a
Table 1
Explanation of measured values in Fig. 4.

<table>
<thead>
<tr>
<th>N</th>
<th>$\hat{R}<em>{W}$ $\hat{V}</em>{SW}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>0</td>
</tr>
<tr>
<td>2</td>
<td>△</td>
</tr>
<tr>
<td>3</td>
<td>▽</td>
</tr>
<tr>
<td>4</td>
<td>▽</td>
</tr>
<tr>
<td>5</td>
<td>x</td>
</tr>
</tbody>
</table>

Fig. 3 The dimension of the materials, to be the range of measured reflection coefficients $R_{W}(N=10\theta)$ are within ±0.1. (The dimension $=A/\cos\theta$)

Fig. 4 Measured and calculated reflection coefficient of the felt as a function of incidence.

Fig. 5 Measured specific normal impedance ratio $(5 - \mu + j\chi)$ of the felt as a function of angle.

(By method I)
The "Interference Pattern Method"

typical porous material, is shown in Fig.4 as a function of the angle of incidence for 10.0 kc/s, and the specific normal impedance ratios of the material are given in Fig.5. The impedance ratios are obviously depending on the angle of incidence, however, the impedance ratios of a glass-fiber (50mm-thickness) were not depend on the angle.

Scattered reflection from a semi-cylindrical boss

In this section, it is shown that the method is useful even in scattered field. The reflection coefficients were measured by method (II) for a 5.8mm-radius semi-cylindrical boss on the rigid plane.

Consequently, as shown in Fig.6, measured reflection coefficients as a function of $k R_b$ agree with calculated coefficients(9).

Acknowledgements

The author gratefully acknowledges helpful discussions and encouragements with Prof. Z. Maekawa and Prof. K. Matsuzawa during this work. He also wish to thank some students of Kobe University for assisting with the experiments.

References

A Simple Method of Evaluation the Diffusion of Sound Field in the Rooms

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To attain high-quality acoustics in the rooms, the sufficient diffusion of the reflected sound is very important. When omnidirectional sound source, whose power is \( P \), radiates in stationary or nearly stationary state in the room the density of direct sound energy at distance \( \gamma \) from the source is equal to:

\[
E_d = \frac{P}{4\pi \gamma^2 c}
\]

where \( c \) is sound velocity. In case of full diffusion reflected energy density will be:

\[
E_r = \frac{4P(1-\bar{\alpha})}{\bar{\alpha} Sc}
\]

where \( S \) is the area of the internal surfaces of the room and \( \bar{\alpha} \) mean sound absorption coefficient. When \( E_r/E_d = 6 \) the full level of sound in the room to within 0.6 dB is equal to the level of the reflected sound if the last is fully diffuse (not taken into account the absorption of direct sound at its grazing propagation along the audience makes this value still lesser). This condition takes place at distance

\[
\gamma = 0.35 \sqrt{\frac{\bar{\alpha}}{1-\bar{\alpha}}} S
\]

at greater distances the sound pressure level at full diffusion of the reflected sound should be constant, by incomplete diffusion the level should be variable.

These considerations lead to a simple method of the experimental evaluation of the sound diffusion in the rooms. The measurements
were carried out in three multi-purpose (moving pictures, conferences, theatrical performances and music) halls. The shape of the halls is shown on Fig. I. Hall A has good proportions and a greatly dismembered interior surface. Hall B is notable for its complicated form comprising several connected volumes. Hall C has an elongated form, small height and sound absorbing ceiling. The principal data of the halls are given in Table I. The value \( \zeta \) in the table was calculated for \( \alpha \) determined using Eyring formula for the least reverberation time of the hall with full audience (\( \zeta \) value for empty hall proved to be lesser).

On the stage of each hall at 1.5m level above the floor spherical sound source consisting of 12 small cone loudspeakers fixed in the sides of dodecahedron is installed; the source radiates the octave bands of white noise. The directional res-

Fig. I.
A Simple Method of Evaluation the Diffusion ...

Table I.

<table>
<thead>
<tr>
<th>Hall</th>
<th>Number of seats</th>
<th>Volume m$^3$</th>
<th>Time of empty hall, sec.</th>
<th>Time of hall with full audience, sec.</th>
<th>Reverberation time of hall with full audience, sec.</th>
<th>Reverberation of hall with full audience, sec.</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td></td>
<td>c/s</td>
<td>c/s</td>
<td>r m.</td>
<td></td>
</tr>
<tr>
<td>A</td>
<td>800</td>
<td>3450</td>
<td>1.6</td>
<td>1.2</td>
<td>1.4</td>
<td>1.0</td>
</tr>
<tr>
<td>B</td>
<td>1200</td>
<td>5650</td>
<td>2.2</td>
<td>1.8</td>
<td>1.6</td>
<td>1.2</td>
</tr>
<tr>
<td>C</td>
<td>600</td>
<td>1400</td>
<td>0.6</td>
<td>0.9</td>
<td>0.5</td>
<td>0.6</td>
</tr>
</tbody>
</table>

The measurements of the sound pressure level created by source in different octave bands (an octave bandpass filter was installed in the receiving set) were made at many points in empty halls; the number of points in halls with the full audience was restricted. These points are shown in fig. 1 by black circles while the position of the sound source is shown by white circle. The relative sound pressure levels measured at the points along the empty (full lines) and full halls (dotted lines) are shown in fig. 2. The lengths plotted on the abscissa axis were taken from the nearest point the distance of which from source exceeds $\sim$. The

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frequencies shown are geometric mean frequencies of the octave bands. The sound pressure level in hall A, empty or full, as is evident in fig. 2, is nearly constant along the whole length of the abscissa (there is only a small drop under the balcony) this indicates a good diffusion of sound field. More or less satisfactory diffusion is observed in empty hall B while at 40 per cent of seats occupied the diffusion in this hall becomes considerably worse. In hall C, empty or full, the diffusion is poor. This corresponds with the subjective judgements on acoustic quality of the halls.

While the reverberation time in halls A and B is satisfactory and all of three halls have not echo or greatly delayed first sound reflections hall A has a good acoustics, but the hall B is considered as unsatisfactory. In hall C a subjective strong drop of loudness along its length is observed; this accounts for poor audibility in the back half of the hall.

While being simple the described method gives one of the substantial criterions for evaluating the acoustic quality of the rooms.
An experimental method for the evaluation of acoustics of rooms.

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**x** Istituto Nazionale di Ultracustica "O. M. Corbino" Roma

**xx** Fondazione "U. Bordoni" - Roma

The method is based on the following principle: the delay in the various reflections of the reverberated energy brings about a loss of correlation between the resulting and the original signal, as well as an overlapping of informations; that is to say the acoustic signal related to a given information can still be heard during the flow of the following information. It is therefore to be thought that, in addition to the sound wave coming directly from the source, the useful signal also includes by that portion of reverberated power which is connected to the first reflection weighted, maybe, according to a reduction factor, to account for the loss in correlation. It is moreover to be thought that the reverberation acts somehow like a masking noise on the subsequent signals (reverberation noise).

The balance between the useful and the disturbing portions of the reverberated power are mainly depending on the physical and geometrical characteristics of the room, as well as on the characteristics of the signal and the rate of the information flow.

The ratio of useful signal (direct sound wave plus useful portion of reverberation) to reverberation noise is a parameter quite suitable as a criterion for the evaluation of the listening quality. The overall reverberation to useful signal ratio and the overall reverberation to reverberation noise ratio are convenient parameters to indicate the acoustic quality of a room. A knowledge of such parameters will also permit an appreciation of the various existing criteria for room acoustics.

The method we propose, at present only experimented on speech signals, is
Evaluation of acoustics of rooms

based on the determination of the level of the useful signal as well as of the reverberation noise level, by means of subjective tests similar to those successfully adopted to ascertain the transmission quality of telephone systems. It is advisable to effect such determinations on speech signals picked up in the room to be tested and recorded on a magnetic tape; such recorded signals can afterwards be reproduced to be listened in laboratory tests by means of hi-fi earphones.

The results thus obtained are suitable for an evaluation of the acoustics of rooms, particularly from the point of view of sound recording.

To evaluate the quality of live signals, the influence of binaural listening should be considered. The technique for the evaluation of the three above mentioned ratios calls for a recording of the signals in the room to be tested and for a subsequent analysis in a laboratory.

The speech signals must be recorded by means of a two-channel tape recorder; the two channels must be identical and independent. In order to obtain a reference condition, the microphone leading to channel 1 must be placed near the sound source where the level of the direct sound field is prevailing. The microphone of channel 2 must be placed at various distances, including a position quite near the first microphone for a preliminary balance between the two channels. By using two microphones, one of which to be kept at a fixed distance from the sound source, it is possible to record on a single tape two signals directly comparable, thus eliminating any influence from the alterations deriving from the emission and recording systems. The microphones must be identical, panoramic, with flat frequency response up to about 8000 Hz. The test material can consist in short passages, to be read in a constant volume of voice. It is advisable to record the test signal on a magnetic tape and to reproduce it by means of a hi-fi apparatus.

The following analysis must be effected on the recorded signals:

a) The level $L_{ts}$ of the overall sound field (direct sound plus reverberated sound field) is measured as a function of the distance $s$ from the sound source. This makes it possible to check whether the distribution of the sound field within the room is or not regular. To this purpose the signals from channel 2 are used; all signal levels are referred to the direct sound level, obtained from channel 1.

b) The level $L_{us}$ of the useful signal is determined at various distances from the
sound source by means of a loudness balance test method. To this purpose
channel 2 is employed using as reference the direct sound level from channel 1.
c) The reverberation noise level \( L_{es} \) is determined by isopreference test method.

The comparison is made between the signals from channel 2 and from channel 1
previously balanced in loudness. A white noise is sent in channel 1 in addition
to the signal to obtain a quality equivalence to the signals affected by reverberation.
The level of white noise satisfying to this condition is taken as measure of the reverberation noise. In our tests the white noise has been limited
to frequencies \( \leq 4,000 \) Hz.

From the results of the various determinations described in a) and b) the ra-
tio of overall sound field/useful signal \( (L_{ts} - L_{us}) \) is obtained, versus distance \( s \)
from the sound source. In those parts of the room where the contribution of the direct
sound can be neglected, the above mentioned ratio takes over the meaning
of reverberation/useful signal ratio \( (L_r - L_u) \). On the other hand, from the re-
sults obtained under points a) and c) the ratio of overall sound field to reverbera-
tion noise \( (L_{ts} - L_{es}) \) is obtained, which for the diffuse field takes the meaning
of overall reverberation/reverberation noise \( (L_r - L_e) \).

At last, from the results of tests under points b) and c) the ratio of the use-
ful signal to reverberation noise versus distances \( (L_{us} - L_{es}) \) can be obtained.
In the diffuse field such ratio becomes \( (L_u - L_e) \).

The final values of the three above mentioned ratios \( (L_r - L_u, L_r - L_e \) and
\( L_u - L_e) \) for a speech signal with a given rate of the information flow, are depend-
ing on the physical and geometrical characteristics of the room and can therefo-
re be used to evaluate the quality of the room in respect of the signal to be sent
forth.

The results of the first applications of our method in rooms with different acous-
tical characteristics have appeared to be in good agreement with direct quali-
tative judgements.

Figs. 1 and 2 show a sample of the results obtained in typical rooms.
Evaluation of acoustics of rooms

References


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Fig. 1: Room Volume = 400 m$^3$; Surface = 316 m$^2$; Reverb. time = 1.6 sec

Fig. 2: Room Volume = 202 m$^3$; Surface = 260 m$^2$; Reverb. time = 4.8 sec

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Relation between Articulation and Primary Reflected Sound in Auditorium.

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Introduction

Lately, application of diffusion and perturbation techniques has become more remarkable in the field of acoustical designing of room structure. It is, however, obviously important for speech articulation to cover every seats in the room with plenty of primary reflected sound, which reaches to them immediately after direct sound. Author has designed many auditoria and conference halls from this point of view. But there encountered many cases that circumstance do not allow to satisfy these conditions due to the architectural restriction. I wish to report in this paper the study of the difference of PA in these two different types of halls.

We can classify reflected sound into two groups, that is, the one is regular reflected sound, which reaches to audience after one or two reflections, the second one reverberant sound, which is chiefly produced by random or repeated reflection. The latter gives fullness in sounds of music, but noxious to articulation of speech, since the delay is over 50 ms.\(^{(1)}\) The shorter the time difference there is, the better is the articulation. We experience frequently, some seats are not satisfactory in audibility where receiving no primary reflection. This often happens under the gallery, even though no fault is
Relation between Articulation and Primary Reflected Sound in Auditorium

<table>
<thead>
<tr>
<th>No.</th>
<th>Name of Auditorium</th>
<th>V(m³)</th>
<th>PA(Max.-Min.)</th>
<th>Seats.</th>
<th>RT</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Shinjuku Koma Stadium (early)</td>
<td>22000</td>
<td>80.3</td>
<td>2923</td>
<td>1.8</td>
</tr>
<tr>
<td>2</td>
<td>Asahi Festival Hall</td>
<td>17000</td>
<td>89.8(97.0-85.0)</td>
<td>2800</td>
<td>2.9</td>
</tr>
<tr>
<td>3</td>
<td>Umeda Koma Studium</td>
<td>13900</td>
<td>81.8</td>
<td>2045</td>
<td>1.25</td>
</tr>
<tr>
<td>4</td>
<td>Kobe Kokusai Kaikan Hall</td>
<td>11100</td>
<td>90.5</td>
<td>1892</td>
<td>1.7</td>
</tr>
<tr>
<td>5</td>
<td>Tokyo Sankei Hall</td>
<td>10300</td>
<td>87.9(97.0-77.0)</td>
<td>1704</td>
<td>1.85</td>
</tr>
<tr>
<td>6</td>
<td>Osaka Shin-kabuki Theater</td>
<td>6900</td>
<td>87.6</td>
<td>1883</td>
<td>1.13</td>
</tr>
<tr>
<td>7</td>
<td>Higashi-Osaka Citizen Hall</td>
<td>6570</td>
<td>92.5(98.8-85.3)</td>
<td>1504</td>
<td>1.81</td>
</tr>
<tr>
<td>8</td>
<td>Meiji kabuki Theater</td>
<td>5670</td>
<td>86</td>
<td>1545</td>
<td>1.02</td>
</tr>
<tr>
<td>9</td>
<td>Sankei International Cong. Hall</td>
<td>4541</td>
<td>85.2</td>
<td>600</td>
<td>1.3</td>
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<tr>
<td>10</td>
<td>Tokyo City Children’s Hall</td>
<td>3680</td>
<td>96.4(99.1-92.9)</td>
<td>708</td>
<td>1.8</td>
</tr>
<tr>
<td>11</td>
<td>Kobe Kokusai-Kaikan, Small Hall A</td>
<td>3037</td>
<td>88.5</td>
<td>800</td>
<td>1.3</td>
</tr>
<tr>
<td>12</td>
<td>&quot; B</td>
<td>3037</td>
<td>89.2</td>
<td>800</td>
<td>1.3</td>
</tr>
<tr>
<td>13</td>
<td>Toshiba Cent. Research H., State A</td>
<td>1700</td>
<td>96.0(99.1-92.9)</td>
<td>265</td>
<td>1.1</td>
</tr>
<tr>
<td>14</td>
<td>&quot; State B</td>
<td>&quot;</td>
<td>94.4(99.0-89.8)</td>
<td>&quot;</td>
<td>0.92</td>
</tr>
<tr>
<td>15</td>
<td>Tokyo Gas Hall</td>
<td>1060</td>
<td>94.3</td>
<td>367</td>
<td>1.1</td>
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<tr>
<td>16</td>
<td>Toshiba Ginza Hall</td>
<td>425</td>
<td>93.4(94.8-90.4)</td>
<td>118</td>
<td>0.75</td>
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<tr>
<td>17</td>
<td>Nikkatsu Motion Pict., Audition R.</td>
<td>119</td>
<td>96.8(99.0-94.0)</td>
<td>100</td>
<td>0.7</td>
</tr>
</tbody>
</table>

Table 1, Constants of auditorium, RT shows at 500 Hz, no audience.

Observed in the decay curve. Sound reflected several times can not be perceived as regular reflection, because it gradually diverges unless the reflection surface are extremely large.

Relation between PA and types of auditorium

Table 1 shows constants of auditoria and respective PA value. Some of these data in this table are not quite true as they are somewhat disturbed by weak noise caused by building construction works while taking data. But I acted as a speaker in all tests, so general condition of the test is fairly constant. Fig. 1 shows PA with respect to volume of auditoria. In this figure it is evident that there is difference in these two types of auditoria. Group one (black points) is designed positively to distribute plenty of primary reflection, the other group (white points) is the one that such techniques are not applied.

The scope of this paper is to show that the difference of
Relation between Articulation and Primary Reflected Sound in Auditorium

articulation is caused by the difference of architectural conditions, by referring existing auditoria which I served as a consultant for each architect. Then simulation tests by electrical circuit and earphone are not performed yet. Very high PA value is noticeable in former table, but this is something which occurs due to the Japanese characteristics, as every syllable in Japanese has a vowel-like ending.

The auditorium of The National Education Center (Toranomon Hall, now ICA is holding) is one of the primary reflection type one. I wish to see everybody pay attention to its audibility. The largest conference hall on the sixth floor in this building, and six halls in The Kyoto International Conference Hall (2) (The Speech Symposium is to be held) are another type. These are consulted acoustically by me, but PA test is not performed yet, therefore these data are not added in this paper.

Practical designing of primary reflection surface

In general, only ceiling is normally considered as available surface of primary reflection. Most of the group one auditoria in the Fig. 1 are designed in the similar way as those in Fig. 2. Reasonable ceiling shape is ABCD to act as primary reflection to audience from the original point P. Waved ceiling ABEFGH is made by lifting the rear portion of it. When the depth of the stage is very deep, original point P moves between QR, then plane EF must be adjusted in angle as shown in the figure. The curved ceiling illustrated in the upper
Relation between Articulation and Primary Reflected Sound in Auditorium

margin above the figure includes these angles. The surface must be larger enough comparing with sound wave, and also time difference must be suitable.

Generally saying if side walls are required to be as primary reflector, they must be inclined by more than 40°. This condition is not quite acceptable in reality to architectural designing. However, Toshiba Central Research Hall (3) (object No. 13 and 14) is taken up as an example as very rare case. Its side walls are constructed acoustically transparent. And behind the transparent walls there exist regular reflection walls. I examined the side wall effect by covering the side walls with absorbing curtains. Since the PA of ceiling reflection alone is very high value as 94.4 (99.0-89.8)%*, the merit of side walls did not appear so evidently, but still considerable improvement was found at the worst seat area. The value is 96.0 (99.1-92.9)%. But I can hardly make definite conclusion of the side wall effect by this only one example. On the contrary, I can introduce Kabuki theater (object No. 6 and 8) to you as a typical example which has acoustically no suitable reflection wall. This is because the traditional performance of Kabuki play requires extremely classical atmosphere in architecture at the sacrifice of all acoustical effect. As the result its PA is always poor.

* A(B-C): A = mean value of the whole seats, B = PA of highest seat, C = PA of worst seat.

(1) H. Haas, Acustica, Vol. 1, 1951
(2) S. Otani, M. Shizahara, Preprint text of ASJ, Nov., 1966
(3) M. Shizahara, A. Ozawa, " " Oct., 1962
Concert Hall Shapes for Minimum Masking of Lateral Reflections

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Introduction

The perceived quality of musical sounds in an auditorium is affected by the details of the reflection sequence at the listener’s ears during the first 100-200 Milliseconds after the arrival of the direct sound (1,2).

Not all reflections which occur during this time are perceptible however, either as discrete acoustical events (echoes) or as making an integrated contribution to the qualities of the perceived sound (reflections). Masking levels of the order of -10dB relative to either the level of direct sound or a strong reflection are set up for frontal reflections and are sustained for about 60 Ms for musical sounds, thereafter dropping off a further 5 dB during the next 50 Ms (3). These levels are up to 10 dB lower for reflections from the side. Marshall (4,5) has suggested that a major qualitative acoustical difference in Concert Halls is due to the masking of the lateral reflections in some of them, and has discussed some of the room parameters which affect the occurrence of this effect. It is unnecessary to repeat the argument for the importance of lateral reflections in the perception of a reflection sequence but one notes that subsequent papers by other authors have drawn attention to the same phenomenon (6,7).

It has also been shown that if boundary absorption can be ignored, major reductions in level during transit from source to receiver occur only by spherical
Concert Hall Shapes for Minimum Masking

divergence and the effect of the audience seating upon sound propagating at grazing incidence across it (8). This interaction is frequency selective in the range 100-500 Herz and may result in levels as much as 15-20 dB below the spherical divergence level. Reflection paths remote from the audience are not subject to this interaction. It follows that reflections via an over-head stage reflector may arrive with levels up to 15 dB in excess of the corresponding side wall reflections which have propagated across the audience, and if they precede them will produce the masking alluded to earlier.

The architectural studies summarised in this paper have been directed toward the design of room shapes which minimise the masking of lateral reflections. The architectural design of a concert hall is however a much more complex process than the total solution of its many acoustical requirements and here we are dealing with but one of these. The shapes indicated, then are to be regarded in this light, as partial solutions.

Areas of Audience for Consideration

Not all seats in an auditorium need to be considered in terms of reflection masking. Areas in which it is irrelevant are:

(a) Those near the stage, before the audience interaction dip in the transmission characteristic is fully established, receive so much direct sound that all but very early or very late reflections are masked (figure 1).

(b) Seats for which side wall reflections have a shorter path than overhead or ceiling reflections.

(c) Seats in the front three rows of galleries. Here the audience interaction is missing so masking does not occur.

The areas remaining comprise usually, the centre 2/3 of ground floor seating and a similar area of the gallery.

Fig. 1 Masking for a seat only 10 Ms from the source, before the audience interaction with the sound is established.
Concert Hall Shapes for Minimum Masking

Orchestral Locations for Considerations

Halls which exhibit reflection masking are invariably criticised for "lack of bass" or "lack of acoustical warmth". These subjective assessments correlate very well with the maximum dip at 100-200 Hertz in the transmission curves published by Shultz & Watters (8). The instruments for which reflection masking will be particularly important, therefore are the cellos, trombones, bassoons.

Architectural Solutions for Minimum Masking

1. Classical Concert Hall Shape (Figure 2)

   Characteristics: Near square cross section, substantially flat floor, no overhead reflectors other than the ceiling. Proportion and architectural limitations have been discussed extensively elsewhere (4,10). Renowned for reverberance, and warmth of tone, but tend to lack of clarity due to high ceiling. Existing halls such as Vienna, Grosser Musikvereinsaal are already at the limiting ceiling height for adequate clarity, so that the same shape in cross section cannot simply be "scaled up" to accommodate more people. These halls produce unmasked lateral reflection at most seats. Examples: Boston Symphony Hall, Concertgebouw.

   Fig. 2 "Classical" Hall Cross Section

2. Modified "Modern" hall:

   If overhead surfaces can be contrived to produce lateral reflections remote from the grazing incidence path, masking is unlikely and stage reflectors are permissible. Reflector dimensions must be chosen to produce reflections of 'cellos at all audience areas where masking might take place. Example: Christchurch (N.Z.) Town Hall, Architects, Warren & Mahoney, Christchurch, N.Z., Acousticians, Engineering Design Consultants, London, in association with the author of this paper.

3. Subdivided Audience: (Figure 3)

   An initial attempt has been made to surround a major portion of the audience with walls placed to provide correct reflection sequences. (Fig. 3). This involves the
subdivision of the peripheral audience into open "boxes"
seating 2-300 people and opening into the main volume.
Masking levels are reduced by splaying the box walls so that
the reflections propagate approximately parallel to the seat rows,
and by a steep rake to the floor.
In this sketch the central floor seating has the same width as the
Grosser Musikvereinsaal. No complete example can as yet be quoted but in parts, (and fortuitously,) the Berlin
Philharmonic fulfills the acoustical requirements for avoiding masked lateral reflections by subdivision of
seating areas.

Fig. 4 is a photograph of a study model which further shows the form of this solution.

In the particular case illustrated, the subdividing walls also form the room boundaries but clearly they need not do so. Suitable surfaces could be disposed irregularly or regularly according to other architectural constraints to produce suitable reflection sequences within a much larger space than that illustrated.

The Influence of Early Lateral Reflections on the Spatial Impression

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Introduction

In an earlier paper Lochner and Keet compared stereophonic and quasi-stereophonic systems with respect to the apparent source width perceived by an observer listening to a two-loudspeaker arrangement in an anechoic room. They suggested the term "Ambience", relating to the perception of sound in a concert hall where directional separation of the primary sources is absent, and concluded that "... the amplitude and time difference in the echo patterns at the two ears create an apparent broadening of the sound source. The extent of broadening depends on the magnitude of ... (these) ... differences".¹

The author is of the opinion that this "broadening of the sound source" plays a very important rôle in concert hall acoustics and that in fact it represents a quality which by its presence or absence can make the difference between excellent or mediocre concert hall acoustics.

Measurements by Schroeder et al. in Philharmonic Hall,² as well as work by Marshall,³) strengthen this opinion. The author believes that "Ambience" is essentially the same as "Rumöglichkeit", "Schwimmbad-effekt" or "Spatial Responsiveness". It is proposed to use the term Apparent Source Width (ASW) in this paper for that part of the spatial impression which is conveyed by binaural as opposed to monaural listening.

Experimental Procedure

Two-channel stereo recordings were made in 3 concert halls during
The Influence of Early Lateral Reflections

a visit to Germany in 1965, using pre-recorded music reproduced through a single speaker system on the stage, so that the only spatial information would be furnished by the reflections. The music consisted of a 10 second "dry" excerpt from a light orchestral recording using the standard "close-microphone" technique in vogue at present, and was virtually reverberation free. Recordings were made in 4 positions in each hall. Two cardioid microphones were used, at an angle of 90° to each other and with their diaphragms 21cm apart. Recordings from one of the halls were dubbed onto an endless loop, adjusting the respective recording levels for approximately equal loudness by means of a sound energy integrator and "A" weighting network. These recordings were played to an observer in an anechoic chamber, seated equidistantly from two loudspeakers behind a horizontally numbered screen. The loudspeaker axes crossed at the observer's head at an angle of 100° which was experimentally found to give the most natural reproduction. It is interesting to note that colouration became less noticeable as the speakers were moved further apart.

The observer, who kept his head position constant by means of visual sighting marks, was asked to indicate the average source boundaries in terms of the screen numbers. As it was expected from experience in concert hall listening that the ASW would depend on level, the recordings were played back at a series of 8 randomly varied levels, 5dB apart, over a range of 35dB.

Results

Fig. 1 shows the average curves obtained for 10 observers. Standard deviations were of the order of 21°, giving probable errors of the order of 7°. It is seen that the ASW increases with level, the average slope being about 1.6°/dB.

If we regard the ear as carrying out cross- and autocorrelation after Sayers and Cherry 4) a possible explanation for this phenomenon may be given by assuming that at low levels, i.e. under difficult listening conditions, the hearing mechanism suppresses unfused sound, and that this suppression is relaxed as conditions improve.
Of even greater importance is the marked difference in ASW between the different positions. The above-mentioned work as well as the work of Burgdorf and associates\(^5,6\) suggested an analysis of the channel signals on a cross correlation basis. The histograms of fig. 2 show the short-time cross-correlation coefficients between the two channel signals for a 5ms swept-frequency pulse starting at 1kHz and ending at 1.2 kHz, which was played back over the loudspeaker on the concert hall stage and recorded stereophonically in the same positions. The short-time cross-correlation coefficient \(K\) is defined as

\[
K_{t_2} = \frac{\int_{t_1}^{t_2} A \times B \, dt}{\sqrt{\int_{t_1}^{t_2} A^2 dt \cdot \int_{t_1}^{t_2} B^2 dt}} \quad \cdots \cdots \cdots \cdots \cdots \cdots \cdots (1)
\]

where \(A\) and \(B\) are the two channel signals.

The integration was carried out in 50ms steps. It is seen that in the interval 0-50ms there are marked differences in \(K\) for the four positions, and in the correct sequence, i.e. a high coherence corresponds to a narrow source. In the other intervals the sequence is lost, except in the interval 150-200ms. Fig. 3 shows the fraction of laterally incoherent energy \(1-K_{50}^0\), for the first 50ms, plotted against the ASW, with listening level as parameter. It is seen that the relationship remains linear for all levels used.

It is seen that \(1-K_{50}^0\) is a measure of the ASW, at least for the positions investigated.
The Influence of Early Lateral Reflections

Summary of Results

1) Stereophonic recordings of single source music reproduction in a concert hall, made at different seat positions, show considerable variations in apparent source width.

2) The apparent source width increases with level, with an average slope of about $1.6^\circ$/dB.

3) For a given level, the apparent source width depends linearly on the incoherent lateral energy fraction $1-K_0^{50}$ where $K_0^{50}$ is the cross-correlation factor between the channel signals for an impulse recorded at the same seat position, integrated over the first 50ms.

Conclusions

The importance of early lateral reflections for the spatial impression seems to have been proved. The fact that these reflections fall within the integrating period of the hearing mechanism for speech means that they will contribute to the clarity. An objective measurement criterion for the ASW has been found, but it may have to be modified for other types of music. It is also likely that the critical interval does not end abruptly at 50ms, but that a weighting factor operates as with speech integration, but depending on different parameters. It should however be noted that not all lateral reflections need contribute to the ASW. With source and receiver on the symmetry axis of a laterally symmetrical hall, for example, the coherence of lateral reflections would be high.

Acknowledgements

The author wishes to convey his sincere appreciation to the many people in Germany who gave him assistance in connection with this work. He is especially grateful to Prof. L. Kremer for valuable discussions, encouragement and laboratory facilities, and to Messrs AEG-Telefunken who generously provided apparatus and assistance.

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1) J.P.A. Lochner and W. de V. Keet, JASA 32, 3, Mar '60, 393-401.
4) B. McA. Sayers and E.C. Cherry, JASA 29, 9, Sept '57, 973-987.
Detectable Threshold of Single Echo and Sound Quality in Rooms

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Tadamoto Nimura
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Introduction
The auditory character for the single echo has been investigated by Haas(1), Lochner and Burger(2) and authors(3). It is common to those results done by them that the single echo to 30 msec delay can not be detected easily and it is useful for increasing of the loudness of the direct sound. The relationship between the auditory character for the single echo and an articulation in a room has been investigated by Lochner and Burger(3), but this question of the sound quality in a room has not been still developed.

This paper consists of the part of the results that authors have carried out experiments related to this question. On the basis of the results in this paper, the relationship between the sound quality and the detectable threshold is discussed, and it can be found that they have the similar tendency to the echo delay.

Relationship between sound quality and echo delay
(1) The case of the echo with the different reverberation. The psychological scales of "splendid and vast" shown in Fig.1 were measured with the method by Scheffé's paired comparison after following experimental arrangements: In the first place, for the sake of taking the time interval between the direct sound and the first echo, the signal sources recorded on the tape were reproduced with the speaker placed on the stage in the hall(V=13,400 m², R.T=2.0 sec in 500 cps), and the reproducing sounds were recorded with the 2ch tape recorder; The signal sources corresponding to the direct sound in the psychological measurement were electrically recorded on the one
channel, and at the same time the signal sources corresponding to the first echo and the reverberant sounds were recorded on the other channel through the nondirectional microphone put on the position corresponding to the first echo delay. The signal sources consist of the short composition (news) announced by the young woman, the part of Schumann's piano variation and the part of the 3rd movement of Beethoven's the 6th symphony. In the second place, the artificial sound field for the psychological measurement was composed with seventeen pieces of the speaker and the 2nd tape recorder in an anechoic room with dimensions of 4.5×4.5×2.8 m³.

The results were checked with F-test and the main effect for the psychological scales was allowed in the level with the significance of 1%. The yardstick for the comparison of the main effect, i.e., \( Y(0.01) \) was written in Fig.1.

(2) The case of the echo with the reverberation and without. The psychological scales of "delicious" and "splendid and vast" shown in Fig.2 were measured with the same method in (I). Fig. 2-a shows the case of the echo with the same reverberation, and Fig.2-b shows the case of the echo without the reverberation. The same symphony in (I) was used as the signal source. The time interval between the direct sound and the first echo was practised with the delay equipment, so that only in the case of adding the reverberation to the echo the signal source was reproduced in the lecture room (V=263 m³, R.T=2.2 sec in 500 cps) and then recorded on the 2nd tape in the only one point of the nondirectional microphone.

The main effect in this case was also checked with F-test and

---

**Fig.1** Relation between psychological scale and echo delay
SPL of DIRECT SOUND, 75 dB. SPL of ECHO, 70 dB
Detectable Threshold of Single Echo and sound quality in Rooms

Fig. 2 Relation between psychological scale and echo delay

allowed in the level with the significance of 1%. Regardless of the reverberation, it is useful to the scale of "delicious" that the echo to 30 msec delay follows on the direct sound, but this thing is contrary in the case of "splendid and vast".

Detectable threshold of single echo

The results shown in Fig. 3 were measured with the method of the limiting. The full lines show the results as a function of the duration, the dashed lines show the results as a function of the hearing level. Using two speakers close to each other in an anechoic room, the detectable threshold level (DTL) was defined as the mean values of "jnd" and "jnnnd" of the echo from the speaker for the echo. The delay was defined with the time interval between the beginning of the direct sound and the echo.

From the results shown in Fig. 3, it can be found that the DTL represented as a function of the logarithm of the echo delay decrease within 30 msec slowly and decrease after it quickly, regardless of
Detectable Threshold of Single Echo and Sound Quality in Rooms

the duration and the hearing level of the tone burst.

Discussion

For emphasizing the special sound quality in a room, it may be necessary that the echo with the proper delay follows on the direct sound regardless of the kinds of the signal source, the reverberation and the hearing level. From the common results in above measurements, that is to say, both the psychological scale and DTL are different before and after 30 msec delay, we can think that the selection of the echo delay and the detectable threshold of the single echo may relate to each other closely. There may be the question of the perception of the echo in the foundation of the tonal evaluation, and the relation between DTL and the echo delay can play an important role in the factors relating to the tonal evaluation in a room.

Conclusion

We ought to notice that both the sound quality and the detectable threshold as a function of the echo delay change greatly before and after 30 msec delay regardless of the method of the measurement.

1000 CPS, TONE BURST
RISE AND FALL TIME, 5 MSEC
SPL OF DIRECT SOUND, 70 dB CONST,
DURATION, ——— 600 MSEC
  ——— 400 "
  ——— 200 "
  ——— 100 "
  ——— 50 "
  ——— 25 "
DURATION, 100 MSEC CONST,
SPL OF DIRECT SOUND,
—-(1)— 76 d.
—-(2)— 64 "
—-(3)— 58 "

Fig. 3 Relation between DTL and echo delay

References:
(1) H. Haas, Acustica, 1, I95I.
(3) T. Yanagisawa and T. Nimura, Reports of the I964 Autumn Meeting, J.A.S.J.
ZUR KLANGFÄRBUNG DURCH KURZZEITREFLEXIONEN
BEI RAUSCHEN, SPRACHE UND MUSIK

Ludwig Müller
Westdeutscher Rundfunk Köln

Die in Räumen häufig beobachtete Klangfärbung von Sprache und Musik wird in erster Linie durch Kurzzzeitreflexionen /1/ und Raum-eigenschwingungen (stehende Wellen) verursacht. In viel geringerem Maße als diese tragen nichtlineare Nachhallkurven zur Klangentstel-
lung bei. Zunächst wurde durch subjektive Hörvorliebe festgestellt, daß die im Raum beobachteten Klangfärbungen und Klangentstellungen vollständig durch elektroakustisch simulierter Reflexionen mit einer Verzögerungszeit von etwa 0,5 bis 15 ms nachgebildet werden können. Außerdem ergaben im reflexionsarmen Raum mechanisch hergestellte Re-
flexionen die gleichen Klangfärbungen. Alle im folgenden über Hör-
ereignisse bei Kurzzzeitreflexionen gemachten Aussagen gelten auch
für Raumschwingungen, weil für jede am jeweiligen Empfangsort die
Schallereignisse identisch sind, und weil die Empfindung der Klang-
färbung nicht vom Einfallswinkel zwischen Direktschall und Reflexion
abhängig ist.

Bei Verwendung von weißem Rauschen werden Klangfärbungen im Cha-
rakter von Tönen hörbar, die dem Rauschen überlagert sind. Bei Verzö-
gerungszeiten (τ) zwischen 0,5 und 15 ms werden Töne zwischen 2000 Hz
und 60 Hz gehört (f = 1/τ = Wiederholungstonhöhe = repetition pitch).
Sie werden nicht nur bei Wiederholungen, sondern überhaupt bei perio-
dischen Spitzen im Frequenzbereich gehört, wie sie durch die Interfe-
renz des Direktschalles mit einer Reflexion oder durch die Interferenz
zweier kohärenter Reflexionen entstehen /2, 3, 4/. Bei einer Phasen-
drehung des verzögerten Schalles, im elektroakustischen Versuch, ent-
stehen Tonhöhenabweichungen von dem Wert 1/τ. Bei einer Phasendrehung
von 180° z.B. liegt die Wiederholungshotlänge bei 0,88\(\frac{1}{2}\) und bei 1,14\(\frac{1}{2}\) /2,4,5/. Gerade auch unter Heranziehung dieser Tonhöhenverschiebung soll gezeigt werden, daß die gebührte "Wiederholungshotlänge" nichts anderes ist als ein residualer Grundton, welcher im Ohr aus den Spitzen des Interferenzspektrums gebildet wird, wenn diese in Naturtonabständen (1:2:3:4 usw.) auftreten, was bei Interferenz im Raum immer der Fall ist. Die Spitzenwerte des Interferenzspektrums und der residuale Grundton sind in der oberen Hälfte der Abb.1 physikalisch und musikalisch für den Fall \(\tau = 13,7\) ms dargestellt. In der unteren Hälf-

![Interferenzspektrum](image)

Abb. 1

Westdeutscher Rundfunk

Entscheidung der Wiederholungshotlänge (Repetier-Pitch) bei \(\Delta \varphi = 180°\) und \(\Delta \varphi = 180°\) im Falle (L, J. 1976)

der Abb.1 ist das Interferenzspektrum bei \(\Delta \varphi = 180°\), elektrosku-

stisch erzeugt, dargestellt. Da bei \(\Delta \varphi = 180°\) die Interferenzspitzen nicht im Naturtonverhältnis, sondern im Verhältnis 0,5:1,5:2,5:3,5 usw. zueinander liegen, kann aus den Spitzen selbst auch kein residu-
aler Grundton entstehen. Das Ohr (Basilarmembran) bildet statt dessen einen residualen Grundton aus solchen möglichst hohen Schalldruckwer-
ten in Spitzenhöhe, die exakt im Naturtonverhältnis zueinander liegen, wobei die harmonisch stärksten Intervalle wie Quinte (2:3), Quarte (3:4), Terz (4:5) vorrangig die Frequenzlage dieser "möglichst hohen" Schalldruckwerte bestimmen. (Der in Notenschrift aufgezeichnete Hör-
vorgang bei \(\Delta \varphi = 180°\) erinnert lebhaft an die Darstellung von J.F. Schouten zur analogen Erklärung des Glockenschlagtones als Residuum /5/. Wie in Abb.1 graphisch ermittelt, gibt es dabei zwei Alternati-
ZUR KLANGFÄRBUNG DURCH KURZZEITREFLEXIONEN

ven, nämlich die Bildung einmal des Grundtones $0,88 \frac{1}{2}$ aus der 3. bis 6. Harmonischen, und zum andern des Grundtones $1,14 \frac{1}{2}$ aus der 2. bis 5. Harmonischen. Wird das Interferenzspektrum, durch einen Hochpaß im Wiederholungskanal, etwa bis zur vierten Harmonischen unterdrückt, und müssen dann höhere Harmonische zur Grundtonbildung herangezogen werden, so wird die Abweichung des Grundtones von der Frequenz $\frac{1}{2}$ geringer werden, weil mit wachsender Frequenz die Abweichung der Interferenzspitzen vom Naturtonverhältnis fortschreitend kleiner wird.

Bei einer Kurzzeitreflexion von Rauschen tritt also immer eine Klangfärbung durch das Residuum auf, auch wenn die Bandbreite des Rauschens auf wenige Naturtonabstände beschränkt ist. Die einzelnen Spitzen und Auslösungen im Interferenzspektrum werden als solche nicht herausgehört. Der residuale Grundton bei der Frequenz $\frac{1}{2}$ wird eindrucksvoll als gleitende Färbung hörbar, wenn τ kontinuierlich verändert wird.

Auch bei Kurzzeitreflexionen von Sprache entsteht eine Klangfärbung durch das Residuum, wenn deren Spektrum teilweise ein dem Rauschen angenähertes Bandspektrum enthält. Solche Band Spectrum sind mit erheblicher Breite bei allen Zisch- und Frikativlauten (z.B. z, s, sch, f) gegeben. Bei Verzögerungszeiten zwischen 0,2 und 0,8 ms wurden diese Laute infolge des residualen Grundtones ($\frac{1}{2}$) zwischen 5000 Hz und 1250 Hz besonders hart und zischend gehört und aus einem Sprachtext stark herausgehoben. Bei Verzögerungszeiten längere als 0,8 ms verursachen die Frikativlaute dann Residualtöne in tieferen Tonbereichen, wodurch zunächst eine Baßbetonung und schließlich, die etwa 10 ms, ein Dröhnen, ein topfiger Klang, entsteht. Bei Verzögerungen $\tau > 10$ ms stellt sich mehr und mehr die Empfindung der Raumhelligkeit ein. Andere, nicht durch Residualtöne verursachte Färbungen und Entstellungen treten bei allen Vokalen und stimmhaften Konsonanten auf, besonders, wenn Auslösungen durch Interferenz mit entscheidenden Formantfrequenzen zusammenfallen. Die Abb. 2 zeigt z.B., daß die Vokale o und u nicht mehr unterschieden werden können, wenn bei $\tau = 1$ bis 2 ms die Formantfrequenzen $F_1$ ausgelöscht werden. Die verbleibenden Formantfrequenzen $F_2$ sind für beide Vokale nahezu gleich, so daß dann eine o/u-Verschmelzung bzw. eine o/u-Verwandlung auftritt. Abb. 3 zeigt die Klangveränderung der Silbe "Mön" infolge der 1. und 2. Auslösung bei $\tau = 1$ ms. Auch bei Sprache wird die Klangänderung bei gleitendem τ besonders eklatant.
Kurzzeitreflexionen bei Musik können sowohl melodische als auch harmonische Entstellungen verursachen. Einerseits können durch Nullstellen Melodietöne ausge- löscht werden, andererseits können durch Residualtöne Harmonien verändert oder disharmonische Klänge verursacht werden. Wie bei Sprache ist hier das Auftreten von Residualtönen vor allem an breitbandige Spektren, also an eine dichte Instrumentierung und ganz besonders an Schlagzeug, gebunden. Eine Wiedergabe mit gleitendem \( \tau \) läßt fatale melodische und harmonische Veränderungen erkennen.

Bauliche Maßnahmen zur Verhinderung der geschilderten Klangfärbbungen bestehen einfach in der Unterdrückung der gefährlichen Reflexionen durch Absorptionsmaterial für alle Frequenzen oberhalb etwa 300 Hz. Der Schluckgrad ist so zu bemessen, daß die Wahrnehmbarkeitsschwelle, bei Rauschen zu \(-17\) dB, bei Sprache zu \(-12\) dB gemessen, unterschritten wird.

Acoustical Problems of Some Modern Theatres

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Introduction

During the past few years two modern municipal theatres have been built in Finland, those in the cities of Turku and Helsinki. The architects of the former were R.V. Luukkonen and H. Stenros, whereas T. Penttilä was the architect of the latter.

When these theatres were planned, I collaborated with the architects as an acoustical expert together with Messrs. J. Borenius and A. Halme, the former being in charge of electroacoustics and the latter of architectural acoustics.

The seating capacity of the Turku Theatre is 680, that of the main auditorium of the Helsinki Theatre 920 and of the smaller auditorium 300. The following refers to the main auditorium of the Helsinki Theatre if not mentioned otherwise.

Requirements for the solutions

1. The stage enters deep into the auditorium so that the audience listens to the actors from the side, sometimes almost from the back (figure 1).
2. The number of useful reflections from the ceiling was limited since space had to be reserved for the lighting devices and ventilation (figure 2).

3. Because of this the noise level had to be made very low. External noise, noise from the foyers, the scenery workshop, ventilation and many machineries on the stage had to be considered.

4. Owing to the character of the theatre architecture a traditional prompter's box could not be used.

5. The electroacoustical system had to be as complete as possible.

Architectural acoustics

There are concert halls both in Helsinki and Turku and besides that an opera in Helsinki; therefore the task was to create a speech auditorium. The upholstered seats are strongly absorbing, and additional absorption was necessary only on lower frequencies.

The reverberation time of the 5700 m³ Helsinki auditorium is 1,1 sec. at 1000 Hz when the fire-resistant curtain as well as the stage curtain are down. The reverberation time of the 19000 m³ stage tower when empty is 1,6 sec. It is strongly attenuated since no useful reflections occur there. The common reverberation time is 1,3 sec. measured on the stage opening facing the stage.

The audience-stage parts have been separated from other parts of the building from ground to ceiling without connecting points. Between the stages as well as between them and the scenery workshop there are double 16 + 16 cm concrete walls the air space between them being 10 cm and containing poly-
Acoustical Problems of Some Modern Theatres

styren foam plastics. The doors used here are of heavy double sheet construction (figure 2). The automatically opening shutters on top of the stage tower leading out the smoke in case of fire are heavy and tightly closing. The doors leading from the foyers to the auditorium are single, but the audience has to pass through a long and strongly absorbing corridor.

Noise reduction

The ventilation machines are isolated from the building construction with the aid of steel springs and rubber cushions and are situated relatively far from the audience. The bends of the ventilation ducts have been coated with a 7 cm layer of mineral wool covered with a perforated plate. The speed of the incoming air is only about 2 m/sec. thanks to the relatively big inlet holes. The noise caused by the ventilation is with full speed (during intermissions) 30 db(A) and with half speed (during performances) 27 db(A).

When the theatres referred to were built, one of the requirements was a very complicated stage technique, which on its part is apt to cause difficulties in the acoustical design.

The Helsinki Theatre has a round stage with the diameter of 16 m encircled with a separate peripheral stage with the width of 3 m, both revolving independently. The former holds a number of hydraulic hoisting devices for changing the levels of different parts of the stage. The total weight of the revolving stage with its devices is 120000 kg. The revolving stage rests upon a Teflon plastic bedding on a circular rail. The friction coefficient is minimal and the noise caused by the revolving stage is 31 db(A) measured at the distance of 5 m from the edge of the stage.

The theatre is equipped a.o. with hoisting devices for the lighting bridges, machines for dropping and raising the curtain and with a number of heating, cooling and ventilation mechanisms. All these machines have been isolated from the body with elastic beddings and do not emit any additional noise to the audience.

These are the main steps that have contributed towards making the
intellegibility in the Helsinki Theatre in general very high; even when the actors perform at the back of the stage it is good.

Electroacoustics

The purpose of the sound reproduction system is to feed sound effects to the stage and to the audience with the aid of loudspeakers, to feed music and announcements to the foyers, to amplify sound when necessary and to record and edit sound effects.

Special attention has been paid to the fact that sound effects can be fed at the right moment practically from any desired direction. Together 55 stationary sound effect loudspeakers have been placed in the ceiling and wall surfaces of the auditorium. On the stage itself, on the lighting bridges and in the orchestra pit there are plugs for 48 mobile loudspeakers. Nine output channels feed the loudspeakers. Two so called panorama regulators have been connected with these channels for producing a moving sound image.

For amplifying and recording as well as for editing sound effects there are eight input and three group channels in the sound control desk. The sound can be treated stereophonically. Two wireless microphones can also be used.

A special electroacoustical prompting system had to be planned for the Helsinki Theatre. The prompter sits in a room beside the control room with good visual contact to the stage and uses a headset with a microphone and earphones. With switches the prompter guides the prompting into small special loudspeakers. The front stage is taken care of with five small dynamic loudspeakers on the edge of the stage, the middle and back stage is covered with small horn loudspeakers on the lighting bridges. A limiting amplifier is used. Frequencies under 1000 Hz are cut off. In rehearsals it is possible to increase the sound volume by over 20 db. The prompter follows the events on the stage stereophonically from the earphones with the aid of microphones hanging from the lighting bridges.
Notwendige Eigenfrequenzdichte zur Vermeidung der Klangfärbung von Nachhall

Eigentone density and colouration of reverberant sound

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Colouration of reverberant sound also with broad-band noise is avoided if the density of eigentones of rooms, plates and other devices is higher than 3 per Hz. This is true for about 1000 Hz. At lower as well as higher frequencies, a much smaller density is sufficient. Diffusion and reverberation time have no influence on colouration.

Einrichtungen für künstlichen Nachhall bewirken meistens eine metallische Klangfärbung, wenn der zu verhallende Schall ein kontinuierliches Spektrum hat, wie bei Sprache und Rauschen. Verhältnes Rauschen klingt so, als ob Töne zugefügt seien. Auch im stationären Zustand, bei Mischung von direktem Schall und Nachhall ist die Frequenzkurve unregelmäßig. Ihre Maxima bewirken eine Klangfärbung, obschon das Spektrum kontinuierlich ist. Die freien Eigenschwingungen der Verhallungseinrichtung (Hallraum, Nachhallplatte, Nachhallfeder oder Magnettonverzögerungseinrichtung), die nach dem Abschalten des direkten Schalls übrig bleiben, haben dagegen ein Linienspektrum und eine stärkere Klangfärbung. Wie dicht müssen nun die Linien, also die Eigentöne sein, damit verhalltes Rauschen wie das Original klingt?

Bei der Entwicklung der Nachhallplatte wurde bereits klar, daß die Klangfärbung mit der Eigenfrequenzdichte \( \Delta N/\Delta f \) zusammenhängt. Bei einer zu Biegeschwingungen angeregten Platte ist \( \Delta N/\Delta f \) proportional dem Quotient von Fläche und Dicke, unabhängig von der Frequenz. Bei dünnen gespannten Platten nimmt \( \Delta N/\Delta f \) zu tiefen Frequenzen hin ab. Bei einem Raum ist \( \Delta N/\Delta f \) dagegen proportional dem Volumen und dem Quadrat der Frequenz. Bisher war es technisch nicht möglich, die Eigenfrequenz-
Eigenfrequenzdichte und Klangfärbung von Nachhall

dichte der Nachhallplatte zu vergrößern. Wir haben aber vor einigen Jahren untersucht, wie groß die Dichte sein muß, um eine Klangfärbung zu vermeiden.


Wir haben dagegen die mit Rauschimpulsen angeregt, freien, abklingenden Eigenschwingungen eines Raumes und verschiedener Nachhallplatten objektiv und subjektiv untersucht. Der Nachhall wurde auf Magnettonband aufgezeichnet, der direkte Schall abgeschnitten und der abfallende Pegel durch einen Regelverstärker konstant gehalten. Das Spektrum wurde schmalbandig analysiert. Das Filter selbst hatte eine Bandbreite von 6 Hz. Durch größere Bandgeschwindigkeit bei der Wiedergabe wurde eine effektive Bandbreite von 3 bzw. 1,5 bzw. 0,75 Hz erzielt. Die so erhaltenen Frequenzkurven waren fast gleich. Die Abstände der Maxima sind eine statistische Größe, ebenso ihre Pegel, die zeitlich umso stärker schwanken, je geringer die Bandbreite ist (bei 1 Hz Bandbreite bis zu 10 dB). Die Frequenzen der Maxima sind zeitlich konstant.


Der Hallraum wurde mit und ohne Diffusoren benutzt. Von zwei Nachhallplatten mit gleichem ΔN/Δf hatte die eine gerade Kanten, die andere bogenförmige, also schallzerstreuende Kanten. Die Änderung der Diffusität hatte ebenso wie eine Variation der Nachhallzeit von Hallraum und Platten im Bereich 1:4 keinen oder fast keinen Einfluß auf die Frequenzkurven und den Klang. Wenn man dagegen die Frequenzkurve im eingeschwungenen Zustand mit einem gleitenden Sinuston aufnimmt, erhält man bei Platten und Hallraum die nach Schroeder und Kuttruff zu erwartende Zahl der Maxima, abhängig von der Nachhallzeit, und die
mittlere Höhe der "Berge".

Fig. 1 zeigt als Beispiele die mit einer Bandbreite von 1,5 Hz aufgenommenen Frequenzkurven des Nachhalls eines Hallraums \( V = 56 \text{ m}^3 \) und dreier Nachhallplatten \( \Delta N/\Delta f \text{ gleich 3,7/Hz, 1,0/Hz und 0,4/Hz} \). Beim Hallraum ist \( \Delta N/\Delta f \text{ gleich 15/Hz bis 22/Hz. Die größte Pegelschwankung beträgt bei der untersten Kurve 39 dB, bei der obersten 18 dB.} \)

Bei den unteren beiden Platten stellten wir eine sehr starke oder starke Klangfärbung fest, bei der oberen Platte und dem Hallraum keine.

Für Klangfärbungsteste standen die Aufnahmen des verhallten Rauschens vom Hallraum und von fünf Nachhallplatten mit unterschiedlichem \( \Delta N/\Delta f \) zur Verfügung. Außerdem wurde durch größere Bandgeschwindigkeit bei der Wiedergabe die Frequenz erhöht und \( \Delta N/\Delta f \) verkleinert. Bei kleinerer Geschwindigkeit geschieht das Umgekehrte. So wurden fünf weitere Nachhallplatten mit anderen \( \Delta N/\Delta f \) simuliert. Wenn man bei einem Raum die Frequenz 2:1 erhöht, ist \( \Delta N/\Delta f \text{ 4:1 vergrößert. Wenn das Tonband 2:1 schneller läuft, ist die Frequenz auch verdoppelt, } \Delta N/\Delta f \text{ aber halbiert. Gegenüber dem Originalraum ist also } \Delta N/\Delta f \text{ 1:8 verkleinert. Der so simulierte Raum ist achtmal kleiner als der Originalraum. Durch doppelte und vierfache Bandgeschwindigkeit wurden so Hallräume mit } 7 \text{ m}^3 \text{ und } 0,88 \text{ m}^3 \text{ simuliert.}

In Fig. 2 geben die Geraden parallel zur Abszisse das frequenzabhängige \( \Delta N/\Delta f \) der wirklichen und simulierten Nachhallplatten an, die proportional \( f^2 \) ansteigenden Geraden \( \Delta N/\Delta f \) des wirklichen und zweier simulierter Hallräume. Die Geraden sind in den Oktaven durchgehend gezeichnet, in denen kein Unterschied zwischen verhalltem Rauschen und Originalrauschen zu hören war. In den Oktaven mit metallischer Klangfärbung sind die Geraden gestrichelt. Der gesamte Bereich der Klangfärbung ist schraffiert. Die gestrichelten Geraden für den Hallraum und die Nachhallplatten liegen in dem gleichen Bereich. Die Klangfärbung hängt also von der Eigenfrequenzdichte und der Frequenz ab, nicht von der Verhallungseinrichtung. Bei 300 Hz und etwa 10 kHz ist ein \( \Delta N/\Delta f \text{ von nur 0 1/Hz erforderlich, in der Oktave mit der Mittelfrequenz 1200/Hz dagegen 3,2/Hz. Nachhallplatten müssen also nur wegen eines schmalen Frequenzbereichs eine relativ große Eigenfrequenzdichte haben. Die Nachhallplatte EMT 140 hat } \Delta N/\Delta f \text{ gleich 1,3/Hz und dadurch bei unseren Testbedingungen eine Klangfärbung in zwei Oktaven. Bei Sprache ist die Klangfärbung weniger zu merken als bei Rauschen, bei Musik in den meisten Fällen gar nicht. In einem Hallraum mit } 56 \text{ m}^3 \text{ hat verhalltes Rauschen keine Färbung. Bei } 7 \text{ m}^3 \text{ ist eine Oktave gefärbt, bei } 0,88 \text{ m}^3 \text{ drei Oktaven. Das gilt für die Anbringung von Lautsprecher und Mikro-}
Eigenfrequenzdichte und Klangfärbung von Nachhall

phon in Raumecken. Wenn beide Wandler im Raum sind, werden nicht alle Eigenschwingungen angeregt bzw. aufgenommen. Deshalb ist ein Volumen von 40 oder 50 m$^3$ ratsam.

Die Hammond-Nachhallfedern haben ungefähr $\Delta N/\Delta f = 1,5$/Hz. Ihre Klangfärbung reicht wahrscheinlich über vier Oktaven. Die British Broadcasting Corporation verwendete früher bei ihrer NachhallEinrichtung mit Transponierung in den Ultraschallbereich einen kleinen Wassertank mit einem mittleren $\Delta N/\Delta f$ von 0,6/Hz. Die Klangfärbung reichte wahrscheinlich über 2,5 Oktaven. Die elektroakustische Einrichtung zur Vergrößerung der Nachhallzeit der Royal Festival Hall in London bei tiefen Frequenzen bis 300 Hz hat $\Delta N/\Delta f = 0,3$/Hz. Nach Fig. 2 ist keine Klangfärbung zu erwarten.

4 Parkin, P.H. und Morgan, K., Journ. Sound Vib. 2 (1965), 74
Psychometric Approach to the Room Acoustics

(Report II: Distribution of subjective evaluations in five auditoriums and discussion)

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Yokohama, Japan

Introduction

One of authors reported at the 5th ICA on the results of an experiment about the subjective evaluations of sound qualities in an auditorium and the result of factor analysis of these qualities.*

In succession to the preceding experiment, we made the same kind of experiments at five auditoriums which were recently constructed in Japan to find out the general pattern of distribution of subjective evaluations, comparing with the results of the preceding experiment, and also to research the relations between these subjective evaluations and acoustical characteristics in room acoustics.

Various Dimensions of Auditoriums and Method of Subjective Evaluations

Various dimensions of auditoriums in which the experiments were made are shown in Table 1. These auditoriums are used for all purposes, i.e. for concert, for opera, for play, for lecture, etc.

We made entirely same experiments at five auditoriums as follow.

Dividing its seating area into 8-12 blocks, we measured reverberation time, distribution of steady state sound pressure level, and response of tone burst, as acoustical characteristics of auditorium.

The fluctuation of sound pressure level in seating area was within 5-10 dB in each block, and any harmful echoes were not seen by the observation of echo time patterns and not received by the sense of hearing.

The subjective evaluations were made on the qualities of "richness", "clearness" and "sensory length of reverberation" by means of the method of constant sum.
Psychometric Approach to the Room Acoustics, (Report II)

As we discussed previously, these traits were represented from words on the two semantic axes of sound qualities in an auditorium.

In addition to the above traits, the score of articulation and a criterion of "easiness of listening" were measured.

As sound sources, recorded music or speech was reproduced from a loudspeaker installed at the center of the stage.

The Results of Experiments

(a) In spite of differences of shapes, volumes and interior materials, the patterns of richness and clearness estimated at the five auditoriums show very similar figures to the results of the preceding experiment. The typical patterns showing the characteristics of these traits are shown in Fig. 1.

(b) The pattern of sensory length of reverberation shows different figures in different auditoriums respectively, which seems to indicate directly differences in characteristics of each auditorium.

But an outline of figures resembles each other. Seats near the stage and the side walls have a tendency to be estimated comparatively long.

(c) The results of the test of coincidence of patterns and the correlation coefficients among traits are shown in Table 2 and 3.

<table>
<thead>
<tr>
<th></th>
<th>R.T.</th>
<th>V.</th>
<th>S.</th>
<th>Seats</th>
<th>Remarks</th>
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<tr>
<td>Hamamatsu City public hall</td>
<td>1.4</td>
<td>9,166</td>
<td>3,482</td>
<td>1,500</td>
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<td>Akita Pref. public hall</td>
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<td>4,771</td>
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<td>Balcony</td>
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<td>3,000</td>
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<td>Obihiro City public hall</td>
<td>1.7</td>
<td>10,520</td>
<td>3,435</td>
<td>1,500</td>
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</tbody>
</table>

Table 1. The dimensions of measured auditoriums.

R.T.; Value of reverberation time at 500 Hz.
V.; Volume of the auditorium (m³)
S.; Total surface area. (m²)
Seats; Seating capacity.
Psychometric Approach to the Room Acoustics, (Report II)

Discussion and Conclusion

(1) As shown in Table 2, The scores of articulation have strong correlation with clearness, and also they depend on the sound pressure level of mid-frequency range. This fact means that they are connected with the ratio of level of direct sound to that of reflected one. So, when we wish to give clear sound to audience in a large hall, it is useful to adopt a sharp directional loudspeaker system whose directional axis is turned to audience.

(2) Richness has no correlation with clearness and also has no correlation with any acoustical quantities mentioned here.

But, thinking of the patterns of distribution of scales concerning richness on each seat, it is evident that the balance between the direct sound and the reflected one is a very important factor.

And it seems to be estimated the richest, when the reflected sound come from all directions around the seat, having nearly equal sound pressure level and time delay, namely, it is "good arranged condition of time-space structure" described in the preceding report.

<table>
<thead>
<tr>
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<th>Obihiro</th>
</tr>
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<tr>
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<tr>
<td>(1) Richness</td>
<td>0.928**</td>
<td>0.962**</td>
</tr>
<tr>
<td>(2) Clearness</td>
<td>0.935**</td>
<td>0.908**</td>
</tr>
<tr>
<td>(3) Sensory length of reverberation</td>
<td>0.741*</td>
<td>0.538</td>
</tr>
<tr>
<td>Sasebo</td>
<td></td>
<td></td>
</tr>
<tr>
<td>(1) Richness</td>
<td>0.839**</td>
<td></td>
</tr>
<tr>
<td>(2) Clearness</td>
<td>0.882</td>
<td></td>
</tr>
<tr>
<td>(3) Sensory length of reverberation</td>
<td>0.412</td>
<td></td>
</tr>
</tbody>
</table>

Table 2
The correlation coefficients among the psychological scales obtained from different auditoriums.

**: over 99% level of significance
*: over 95% level of significance

Fig. 1 Typical patterns of subjective evaluations
(measured at Sasebo City public hall)
This means that the physical quantity related to richness should be treated as
total quantity synthesized of level, delay time, and direction of reflected sound.

(3) According to the result, i.e. richness and clearness are independent
respectively in an auditorium, we think that, in the case of adopting an acoustic
equipment as mentioned in the item (1), we can accomplish our purpose without
loosing richness.

(4) Sensory length of reverberation had negative correlation with richness
in the preceding experiment. But the results derived from the experiments at
different auditoriums do not show the same relation.

The correlation coefficients vary from -0.8 to +0.4. So it is doubtful to con-
clude that their relations are always reverse.

And also, the physical quantity related to this trait should be treated as total
quantity same as mentioned in item (2).

<table>
<thead>
<tr>
<th>Articulation</th>
<th>Easiness of Listening</th>
<th>Richness</th>
<th>Clearness</th>
<th>Sensory length of reverberation</th>
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<td>0.775*</td>
<td>0.469</td>
<td>0.761*</td>
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<td>0.815*</td>
<td>0.255</td>
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<td>0.365</td>
<td>0.918*</td>
<td>0.201</td>
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<tr>
<td>Obihiro</td>
<td>0.773</td>
<td>-0.292</td>
<td>0.973</td>
<td>0.432</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Articulation</th>
<th>Easiness of Listening</th>
<th>Richness</th>
<th>Clearness</th>
<th>Sensory length of reverberation</th>
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<td>0.851*</td>
<td>0.658</td>
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<td>0.498</td>
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Table 3

The correlation coefficients among traits estimated at five auditoriums.

Reference:

*T. Yoshida; Report of 5th ICA

—E-76—
Psychometric Approach to the Room Acoustics

Report III: An improvement of defects of large halls, adopting the methods of psychological measurement

Tomio Yoshida and Yasuo Mafune
Matsushita Communication Industrial Co., Ltd.
Yokohama, Japan

Introduction
In general, it is well known that a large hall has some defects from the acoustical view point. In recent papers \((1)(2)\), we reported that there was no correlation between richness and clearness in a hall. Based on the results of the preceding papers, we attempted to improve the clearness of voice transmission in two large halls (about 30 meters and 70 meters in each depth). Psychological measurement was mainly carried out with subsidiary physical measurement. By fitting up synthesized directional loudspeakers, the two halls were improved on clearness without any architectural repairs.

Outline of Two Halls and Their Defects

One of them is named Koma Stadium (or koma Theatre) after large triple revolving stages of this theatre spinning like Koma, a top in English. The plan and cross-sectional view are shown in Fig.1 and Fig.2 and this hall seats 3,000. This hall was originally designed for a dancing-show applying the reproduced sound of a tape-recorder as a music through a loudspeaker system, and not designed for a microphone use. Following defects have appeared when musicals were staged applying live vocal songs and dialogue through a microphone.

1. The howling grows easily if singers use the microphone.
2. Singer's voice is indistinct in front seats whose number comes to about half of all seats, because it's masked by Orchestra's sound. Here, directivity
Fig. 1: Plan view and arrangement of loudspeakers of Koma Stadium

Fig. 2: Cross-sectional view and arrangement of loudspeakers of Koma Stadium.

Fig. 3: Plan view and arrangement of loudspeakers of Matsushita Hirakata Gymnasium

Fig. 4: Cross-sectional view and arrangement of loudspeakers of Matsushita Hirakata Gymnasium.

- Originally established loudspeakers
- Newly established loudspeakers
of proscenium loudspeaker points in direction of only rear seats and it's hard to turn directivity to front seats, because the stage has a special shape which jits out to front seats.

(3) On the stage, dancers have difficulty in keeping time with music, and player's dialogue is difficult to hear.

Another hall is named Matsushita Hirakata Gymnasium. This gymnasium is sometimes used for lecture and it seats 8,000 or 10,000. The plan and cross-sectional view are shown in Fig.3 and 4. It is measured up to about 70 meters in depth and the reverberation time is about 2.2 second (without audience) and 1.0 second (with audience of 6,000) at 500Hz. Here the absorption of the ceiling is dominant, but walls are reflective. Naturally, the articulation score showed low value.

**Improvement Psychological Measurement and Results**

(1) On the basis of several researches by means of psychological measurement some directional loudspeakers are rearranged. Outlines of the improvement are as follows. The new directional loudspeakers were added as hatched in Fig.1, 2, 3 and 4. Their directions of directivity and acoustical power distributions were adjusted to have the articulation evaluated at maximum and the other psychological items evaluated medial.

In other words, the directions and power distributions of loudspeaker were arranged to make each seat "good arranged condition". For instance, the relation among time, level and direction of sound at point P illustrated in Fig.1 and 2 is shown schematically as Fig.5(a)~(e). But the details of quantitative relation between psychological estimation and physical quantity are not clarified.

(2) By fitting up these directional loudspeakers, the effects of improvement on psychological estimation were indicated as follows. Mean values of articulation score were improved by 1~32 percents at Korna Stadium, and 2~16 percents at Hirakata Gymnasium in comparison with those measured before. Consequently, the articulation score over 90 percents was obtained in every seat, and the amount of rating scales was improved by 0.5~1.5 ranks better than that of original estimation in the both halls.
Fig. 5: Schematic illustration of "good arranged condition". (a, b): sound field
(c): time sequence of (a, b)  (d): subjective recognition of the sound
field (e): and the time sequence recognized.

**Conclusion**

(1) By fitting up some directional loudspeaker systems, two large halls were
improved on subjective estimation of voice transmission without any architec-
tural repairs.

(2) The key of this improvement is to make every seat "good arranged con-
dition", adjusting the directivity of loudspeaker and rearranging the sound
trains caused by the difference of distance from a seat to each loud-
speaker and by the power distribution.

(3) Psychological measurements were repeated several times to find out "the
best arranged condition" at each seat, varying the array of loudspeaker and
the position of loudspeakers, in each hall.

(4) Consequently, the mean values of articulation scores were improved by 1-32
percents and they showed the value over 90 percents at every seat in each hall.
The amount of rating scales was improved by one rank, but another sub-
jective qualities (i.e. richness) were not estimated worse.

**Reference**

(1) T. Yoshida: The report of the 5th ICA "Psychometric approach to the room
acoustics.

(2) Y. Mafune and T. Yoshida: The report of the 6th ICA "Psychometric approach
to the room acoustics." (report II)
Reverberation Control in a Small Hall by Direct Feedback.

Roland W. Guelke
Professor of Electrical Engineering,
University of Cape Town.

Peter R. B. Wilson
University of Oxford.

Introduction

Several methods have been described which can be used to increase the reverberation time of concert halls (ref. 1). They usually rely on a system which is external to the hall either in the form of a reverberation plate, a tape to introduce delay or a separate reverberation chamber, to mention a few.

In one case only, that of the Royal Festival Hall is the increase in reverberation time obtained by direct delayed feedback in the hall itself. However, a considerable number of separate loudspeakers and amplifiers (89) is used to achieve the result, thus forming a somewhat expensive and complex system, each circuit being responsible for a small range of frequencies. Such a system would, therefore, not be suitable for a small hall and the question arises whether it would be possible to cover a wide range of frequencies using a single loudspeaker microphone combination.

A feedback system using one loudspeaker and one microphone

If a single feedback system is to be used to cover a wide range of frequencies, it is necessary for the whole system to have a frequency response that is level within close limits. These limits have to be such that at the frequency of minimum response the feedback is still sufficient to increase the reverberation time significantly without the whole system going into oscillation due to feedback at the
frequency of maximum response. If $t$ is the delay in seconds and $A$ the attenuation in db experienced after this delay, the reverberation time $T = \frac{60t}{A}$. An estimate of these limits can be made as follows. Let it be assumed that a path of 20 m. is available for the delay, this would represent a time delay of 58 milliseconds. If the feedback is so arranged that the intensity drop over the loop is 2 db, then a reverberation time of $\frac{60}{2} \times 58 \times 10^{-3} = 1.74$ seconds, would be produced. If at one frequency the drop is only 1 db, the corresponding reverberation time would be 3.4 seconds and if at another frequency it is 3 db, the reverberation time at this frequency would be 1.16 seconds. A frequency response curve flat within ±1 db would, under these conditions, result in a considerable variation in the reverberation time but would probably still be satisfactory because under normal conditions such variations are quite usual. It is obvious that if the delay is increased, the demands on the constancy of the frequency response become less stringent. Too large a delay would, however, give rise to flutter echoes distinguishable as separate bursts instead of a continuous decay. There are further complications due to phase relationships of the feedback loop (Schroeder ref. 2).

A method of reducing interference

If such a system is placed in a hall, one of the major difficulties is due to interference patterns. Even if microphones and loudspeakers were available with a sufficiently level frequency response, it would not be possible to place a microphone in a position in the hall where it would be free from interference effects. This would mean that at certain frequencies the loop gain would be much greater than normal thus making it impossible to obtain a reasonable increase in reverberation time without oscillation of the system at that frequency.

This difficulty has been overcome by the use of a highly directional microphone constructed with the help of a parabolic reflector.

When this microphone is directed towards a loudspeaker the effect of interference is minimized because the direct sound picked up from the speaker is much more intense than any reflections, due to the fact that the reflections come from
a different direction. Although this still does not overcome the difficulties due
to certain frequencies being reinforced in phase, whereas others are subject to
destructive interference, it was found that a system using this principle could be
adjusted to give a useful increase in reverberation time.

Test in a small theatre

The Beattie Theatre, a hall of about 3,000 cubic metres capacity
(110,000 cu. ft.) at the University of Cape Town, had been designed primarily as a
lecture theatre. For this purpose the reverberation time had been kept fairly short
(about .9 secs. at 500 hz.). As a consequence it was not well suited for music.
A stage was available for performances and it was decided to attempt reverberation
control by the method described above.

Eventually the system, shown diagrammatically in fig. 1 was adopted using two
loudspeakers and two close talk microphones in parabolic reflectors.

Fig. 1. Plan of Beattie Theatre with Reverberation Control.

This arrangement effectively doubles the delay for the feedback loop while
at the same time avoiding distinct echoes. The main area of the hall receives
sound alternately from each loudspeaker at the same interval as before so that the time delay between pulses is not increased.

In order to avoid emphasizing peaks due to the use of two similar loudspeakers, different ones were chosen. The one was a straight horn and the other a cone and horn combination. Both were selected to have as level a frequency response as possible. It is considered that one of the chief difficulties of a system such as this, is the provision of loudspeakers with sufficiently level frequency characteristics. In fact resonant circuits were introduced into the system in order to eliminate certain prominent peaks. The increase in reverberation time obtainable is shown in fig. 2. The upper curve represents the maximum reverberation time which can be obtained without oscillations. The central curve shows the reverberation time as adjusted for a practical case.

Performances of a string quartet were given using this system. The performers and members of the audience who were informed about the system agreed that a considerable improvement had been effected in the acoustics of the hall. The use of the loudspeakers as such was, however, not apparent.

References
(1) Axon, P.E., Gilford, C.L.S., Shorter, D.E.L.
On the Sound System for Architectures

Tadamoto Nimura
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Toshio Sone
Associate Professor of Faculty of Engineering, Tohoku University

In designing sound systems to be installed in halls and other kinds of rooms, we should properly take into consideration not only its function of sound amplification but also improvements in acoustical properties of rooms. This paper discusses from a room-acoustical point of view what characteristics a sound system has to be equipped with, its functions and especially directivity of a loudspeaker system.

1. Functions of a Sound System

Recent progress of electro-acoustical devices make it possible not only to reinforce speech or music, but also to cover up defects of acoustics of a room by means of an electro-acoustical installation, to improve them positively and further to control them at our option. Moreover, it is sometimes put into practice to produce the special sound effects by means of electro-acoustical installation. These functions of a sound system is discussed with reference to acoustical characteristics of rooms.

(1) Sound Pressure and its Distribution

As far as a sound system is used merely as a reproducing system for recorded sounds such as a disc or a magnetic tape, sound pressure is little worth of consideration, but when it serves as a public address system, a distribution of sound pressure can be improved by means of directivity of a loudspeaker system. (See Sec.2)

(2) Frequency-Response Characteristic of a Room

Frequency-response characteristic of a room is able to be covered up by operating a tone control circuit in a system, but an attention should be paid to the fact that a frequency-power characteristic for sounds radiated from the sound system is important.
(3) Reverberation Time

Reverberation time is more effectively varied by electro-acoustical means than by material means, i.e. reversible wall pannels. Reverberation equipment of a magnetic tape recording-type, however, has room for improvement on the point of noise level, and that of a steel plate-type, that of a spring-type and echo room have not a sufficiently good quality of sound. It must be remembered that a sensation of reverberation is influenced by a spatial feature of reflected sounds as well as by a temporal feature. Apparent reverberation time can be reduced by an appropriate directivity of a loudspeaker system.

(4) Diffusion of Sound

A distributed arrangement of loudspeakers and various time differences among outputs from different loudspeakers are helpful to a diffusion of sound like so-called diffusers, prevent sound from focusing, and accordingly get rid of a dead point.

(5) Decay Forms of Sound and Echoes

An arrangement of loudspeakers for synthesis of directivity, a distribution of loudspeakers, a use of a reverberation equipment, time differences among loudspeakers and so forth made it possible to produce or remove echoes, and consequently to control decay forms of sound waves.

(6) Appreciation of Sound Direction and its Movement

It is possible for speech sounds and sound effects to be moved by means of a sound-running system and the like.

(7) Speech Intelligibility

Speech intelligibility in a distant place from sound sources in a large room can be improved by means of increasing the sound intensity through a sound amplification system. On the contrary, reinforcement of the sound through distributed loudspeakers is liable to introduce confusion into listening to the speech and result in a worse intelligibility. In such a case, a directivity ought to be utilized as stated in Sec.2. An automatic control system will be available so that sound intensity in the room may not change dependent on distance and directional relation between a speaker and a microphone. Field-monitoring should be stereophonically practiced so that a monitor in a sound control room may observe a sound condition in the hall.

2. Directivity of a Loudspeaker System

It is required for a loudspeaker system in a sound system to be equipped with a good directivity suitable to room-shape and acoustics of the room, because the following effects can be expected of such a loudspeaker system:
(1) Rise of Howling Threshold

If a directivity of a loudspeaker system is synthesized as shown in Fig.1, a large percentage of radiated sound power is absorbed in seats which have large absorption coefficients, sounds coming from a loudspeaker system up to a microphone and scattered sounds are reduced, and howling threshold rises, so that sound pressure in the room is possible to increase. Then, if a directivity of the microphone is suitable for the occasion, this effect will be enhanced still more.

(2) Improvement of Deutlichkeit

Increase of direct sound from a loudspeaker system and decrease of scattered sound are able to improve "Deutlichkeit" and so speech intelligibility.

(3) Unification of Sound Pressure Distribution

Sound pressure distribution is nearly determined by a direct sound from loudspeakers. Uniform pressure distribution, therefore, ought to be theoretically obtained along with directivity of a loudspeaker system as shown in Fig.1, because a direct sound from a speaker has less intensity than sound radiated from loudspeakers. At the same time "front loudspeakers" had better be prepared for the audience seated in the front row. Sound pressure on the stage being taken into consideration, it is proper to provide supplementary loudspeakers for the stage if necessary.

(4) Effects of a Linear Array and a Curved Array on Synthesis of Directivity

When loudspeakers are formed in a line, directivity at higher frequencies is more sharpened as compared with that at lower frequencies, and so sound quality greatly depends on places in the room. To improve this point, loudspeakers should be placed in a curve or arranged in a curved surface so that synthesized directivity may less depend on...
3. Conclusion

In a word, a sound system should have acoustical characteristics matching so well with those of the room. We should aim at the condition under which sounds radiated from loudspeakers reinforce the direct sounds from sound sources without the audience being aware of it and a sound system suits its own purposes.
Near-field Characteristics of Curved Array for Sound Reinforcement System

Tadashi Wakana (graduate Course, Tohoku University)
Masami Gotoh (Technician, Res. Inst. of Elect. Comm., Tohoku University)
Ken’iti Kido (Professor, Res. Inst. of Elect. Comm., Tohoku University)

Introduction

It is well-known that a directive array is useful for a sound reinforcement system, and such an array is often used. That is, the transmission gain of the acoustic feedback loop in a sound reinforcement system can be reduced and the howling level is largely raised by the use of the directive array, if the greater part of the radiated acoustic power is absorbed by the seats or audiences having a large absorption coefficient. In order to synthesize the sufficient directivity for the array, the length of the array must be much longer than the wave length. Therefore, a directive array of several meters is often used. In the case of the long array, the range of the near field of the array spreads unexpectedly, and covers the greater part of the room. The sound pressure distribution and transient effect of the array differ radically from those of a point source.

First, the results of the theoretical analysis of the sound field of the linear array are presented, and the results of the numerical computation are compared with the results of the experiment.

Next, a method of designing the shape of the curved array is proposed. And it is shown as a result of numerical computation that an adequate sound pressure distribution can be obtained by a curved array designed by the proposed method.

Near field sound pressure distribution and near field transient phenomena of the linear array

The continuous line source is substituted for the linear array for the sake of simplicity. If the sound field in the near field is used,
Near-field Characteristics of Curved Array for Sound Reinforcement

length of the array must be shorter as the frequency increases in order to avoid the sharpening of the directivity pattern. But if the length of the array is long, the concept of directivity is of no use. The model of the continuous line source and the coordinate system are shown in Fig.1. An example of the results of the numerical computation of the steady state near field pressure distribution is shown in Fig.2, and an example of the experimental results is shown in Fig.3 in comparison with the numerical result. In Fig.2, it is shown that the smoothed sound pressure curves in the near field are nearly in inverse proportion to the square root of the distance in the range of \( \frac{k}{\lambda} > 50 \), \( \frac{R}{\lambda} < 10 \), as expected.

If the line source is inclined to the observing plane, it is possible to make a sound pressure level nearly constant within a limited range as shown in Fig.4. This is very useful for a sound reinforcement system.

But the transient phenomena may not be ignored in the near field, if the length of the array is long. An example of the results of the numerical computation of the transient phenomena is shown in Fig.5. In this figure, the zero points of abscissa (t/\( \lambda \) = 0) show the time at which the first sound wave reaches the observing point, or the time at which the steady state ends, and sound pressure in ordinate is normalized by the steady state sound pressure. But such a long array cannot always be used. The curved array had better be used instead of the linear array.

A method to design the curved array

The shape of the curved array is decided by the shape of the room and the position of the loudspeaker. That is, one makes the equidistance dividing positions \( P_1, P_2, \ldots, P_n \) in the range of audiences’ seats in the sectional plane of the room, as shown in Fig.6. Next, one connects the source point s to the points \( P_1, P_2, \ldots, P_n \) by straight lines. The angles between the lines and the horizontal line are denoted by \( \theta_1, \theta_2, \ldots, \theta_n \). The shape of the loudspeaker enclosure is made as shown in Fig.7, using \( \theta_1, \theta_2, \ldots, \theta_n \) as the angles between the corresponding baffle board and the vertical line. In Fig.6, n is the number of loudspeakers used. The distances between the centers of the nearest loudspeakers should be smaller than half wavelength at the highest frequency.

Near field sound pressure distribution of the curved array

The continuous curved array is substituted for the curved array for the same reason as in the previous case. An example of the results of the numerical computation of the sound pressure distribution in the respective area is shown in Fig.8.
Near-field Characteristics of Curved Array for Sound Reinforcement

Conclusion

From the above considerations, it is concluded that the directive array is useful for sound reinforcement system, and its transient phenomena can be neglected, when the length of the array is smaller than about $10^m$. And the curved array is more useful for sound reinforcement system than the linear array. The curved arrays designed by this method are freely used in many auditoria in Japan.

Fig. 1 Model of line source and coordinate system

Fig. 2 Sound pressure distribution of line source ($a=\theta, b=\theta, d=90^\circ, z=0$ in Fig. 1)

Fig. 3 Numerical result and experimental result of linear array. Its array used in the experiment is composed by $9^m$ loudspeakers and its length is $1.2^m$. 
Fig. 4 Numerical result of sound pressure distribution of line source

Fig. 5 Numerical result of transient phenomena of line source \( a=0, b=\xi, \)
\( d=90, z=0, \frac{x}{\xi}=1 \) in Fig. 1

Fig. 6 A method to design the curved array

Fig. 7
a) Continuous curved source used in the numerical computation
b) Numerical result of sound pressure distribution of curved source A-I
Introduction

The well known instability caused by acoustic feedback in public address systems occurs at a peak in the frequency response between a loudspeaker and a microphone.

In order to lower the peaks and improve the acoustic feedback stability of a public address system, an effective method has been studied, namely, sound signals are amplified with frequency modulation.

In this paper, the optimum conditions of frequency modulation in relation to acoustic feedback stability and tone quality are described. Physical consideration on optimum condition of frequency modulation to improve instability

By rapidly swept input signals, the peak value of a single resonance can be reduced. Fig. 1 shows calculated results of the reduction of peak level of a single resonance as a function of modulation index $\rho$ (= frequency deviation $\Delta f$/modulation frequency $\nu$).

Since a frequency response between two points in a room is analogous to that of a multi-resonance system, the peaks of the frequency response in a room are also reduced by frequency modulated input signals. But in the multi-resonance system, there is a possibility that frequency modulation of large deviation causes the neighbouring peaks to build up. As the greater part of energy distribution of a frequency modulated signal is included in the band width $2\Delta f$, it may be said that the frequency deviation should be wider than the band width between the adjoining valleys and narrower than that of the peaks on
either sides, namely \( \Delta f < 2 \Delta v \), where \( \Delta v \) is spacing between a peak and a valley of frequency response in a reverberant sound field. As shown in a previous paper, \( \Delta v \) is related statistically to reverbereation time \( T \) by \( \Delta v = 4.9/T \), so the optimum frequency deviation is about 3 Hz to 20 Hz in a music hall and a lecture room.

A optimum condition is furthermore studied experimentally in various rooms in relation to additional stable gain in a public address system, as shown in Fig. 2 and Fig. 3.

Results confirmed by these experiments are as follows:

(1) The optimum modulation index to obtain the maximum additional stable gain is found about 2.0 to 3.0 in ordinary sound fields such as a music hall and a lecture room.

(2) The larger the reverbereation time of a room is, the more additional stable gain is obtained and the lower the optimum modulation frequency is, because the sound field of large reverbereation time has a high Q factor as resonance, and because of crowded normal modes.

(3) The direct sound transmission from loudspeaker to microphone limits the additional stable gain, because the fluctuation between peaks and valleys in frequency response is reduced.

For reference, peak factors of a sound field measured with a swept warbling tone are shown in Fig. 4, where the peak factor is defined as the difference between maximum peak level and the average level of the frequency response between two points in a reverberant sound field. These results suggest that the obtained additional stable gain agrees with reduction of peak factor.

Consideration on aural problems

In general, sound signals amplified with frequency modulation are injurious to tone quality and seem to limit the practical use of this method, though stability
of the acoustic feedback of public address systems is improved by frequency modulation.

In Fig. 5, the detectability of "warble" in speech as a function of modulation frequency is shown at the modulation index of 2.0.

According to these results, the threshold of detectability of "warble" is 5 Hz in direct sound field and 6.5 Hz on the average in a reverberant sound field when the modulation index is 2.0. Therefore, a large majority of audiences are annoyed by warbling tones.

To avoid the difficulty, a limited frequency modulation band has been experimented with. The lower components of speech signals are amplified without frequency modulation and the only frequency range in which the feedback loop is apt to oscillate should be modulated.

Fig. 6 shows the undetectable limit of frequency range to be modulated in direct and reverberant sound fields. It can be generally said that the optimum crossover point of frequency range with and without frequency modulation in a music hall and a lecture room is twice as much as the pitch of vowels, that

---

**Fig. 2.** Pattern of additional stable gain as a function of modulation frequency and deviation in a music hall (volume 8000 cm³, reverberation time T = 1.5 sec).

**Fig. 3.** A section of patterns of additional stable gain in various sound fields (modulation index β = 2.0).
Improvement of Acoustic Feedback Stability

is, about 500 Hz.

With this limited frequency modulation band, the acoustic feedback stability of public address systems may be improved and the warbling tones of speech are hardly perceptible.

Acknowledgement

The author is much indebted to Dr. H. Nakazima for valuable suggestions and to the members of the Acoustics and Audio Frequency Research Division of the NHK Technical Research Laboratories for discussion in this development.

Reference:


Fig. 5. Percent recognition of "warble" in various sound fields as parameter of male and female voice.

Fig. 6. Undetectable limit of "warble", normalized by pitch of vowels in various sound fields.
Concrete blocks are widely used in the U.S.A. for both interior and exterior 
wall construction, sometimes alone and sometimes as back-up for brick, stone, etc. 
Over 2,000,000,000 block are produced annually on the basis of an 8 x 8 x 16 inch 
hollow block. They have the desirable qualities of being incombustible, economical, 
inorganic, etc. The thermal conductivity and the density depend upon the type of 
aggregate used, ranging in density from lightweight pumice to heavy sand and gravel 
with various types of porous products in between, such as slag, cinders, etc.

A desirable quality of a wall, especially interior walls, is both a high STL 
(Sound Transmission Loss) and high sound absorptivity on the interior surfaces, such 
as when used between classrooms. A problem exists with ordinary hollow concrete 
block, which normally has two or three cavities. In a single block wall, one cannot 
obtain both a good STL and high absorption of the surface. A dense sand and 
gravel block has good STL but low sound absorptivity. A lightweight porous aggregate 
block has good sound absorption but a poor STL. Blocks are normally painted 
in the U.S.A. on most installations. Painting or sealing the porous surfaces im-
proves the STL but materially reduces the sound absorption.

As an example, the following reverberation room tests illustrate the effect of 
painting a concrete block of cinder aggregate:
Sound Absorption of a Slotted Concrete Block

SOUND ABSORPTION COEFFICIENT AT VARIOUS FREQUENCIES (Hz)

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Symbols: CB = Cement Base Paint; a = First coat coverage; b = Second coat coverage.
NRC = Noise Reduction Coefficient - Average of 250, 500, 1000 and 2000 Hz sound absorption coefficients to the closest 5 per cent (0.05)

The problem is one inherent in the composition of the type of aggregate used. However, a new design of block has overcome some of these conflicting problems by slotting the face of the block with 2 or 3 narrow slotted cavities. The sound absorption is mainly due to the Helmholtz resonator principle. The block has one end closed, so when the block is laid up in a wall, each cavity is an individual cavity. In contrast, ordinary hollow block has continuous cavities the height of the wall.

The type block was originally designed for quieting enclosures housing electrical transformers or other mechanical equipment which usually have a sound frequency spectrum with a peak in the low frequency region. The cavities and slots were designed to give peak sound absorption around 125 Hz or between 125 Hz and 250 Hz. Typically this peak was pronounced and the sound absorption fell off rapidly at the upper frequencies. Experience indicated the product was useful for acoustical treatment in many other types of rooms. Many types of acoustical tile and carpet when installed have good sound absorption at the middle and upper frequencies but are relatively inefficient in the 125 Hz and 250 Hz region. By combining the slotted concrete block with acoustical tile or carpet, the result could give a balanced absorption curve of the type desired. In a room designed only for speech a flat absorption curve is desirable, as the prolongation of low frequency sound does not aid in the intelligibility of speech.
Sound Absorption of a Slotted Concrete Block

For more general use, it seemed desirable to modify the absorption curve of the block in order to flatten out the peak and raise the absorption at the upper frequencies. Experiments showed that by changing the width and length of the slots, modifying the shape and size of the cavity and adding a sound-absorbing mineral wool batt, it was possible to obtain a flatter absorption curve. The table below illustrates the varying sound absorption coefficients obtainable from existing types of commercially available blocks.

### Sound Absorption Coefficients

<table>
<thead>
<tr>
<th>Size</th>
<th>Type</th>
<th>Surface</th>
<th>Cavities/Slots</th>
<th>125</th>
<th>250</th>
<th>500</th>
<th>1000</th>
<th>2000</th>
<th>4000</th>
<th>NRC Range</th>
</tr>
</thead>
<tbody>
<tr>
<td>4&quot;</td>
<td>A</td>
<td>Unpainted</td>
<td>2/2</td>
<td>.19</td>
<td>.83</td>
<td>.41</td>
<td>.38</td>
<td>.42</td>
<td>.40</td>
<td>.45-.55</td>
</tr>
<tr>
<td>6&quot;</td>
<td>A</td>
<td>Unpainted</td>
<td>2/2</td>
<td>.50</td>
<td>.68</td>
<td>.28</td>
<td>.30</td>
<td>.48</td>
<td>.56</td>
<td>.40-.50</td>
</tr>
<tr>
<td>6&quot;</td>
<td>A</td>
<td>Painted</td>
<td>2/2</td>
<td>.62</td>
<td>.84</td>
<td>.36</td>
<td>.43</td>
<td>.27</td>
<td>.50</td>
<td>.45-.55</td>
</tr>
<tr>
<td>8&quot;</td>
<td>A</td>
<td>Unpainted</td>
<td>2/2</td>
<td>.75</td>
<td>.44</td>
<td>.24</td>
<td>.27</td>
<td>.29</td>
<td>.33</td>
<td>.30-.40</td>
</tr>
<tr>
<td>6&quot;</td>
<td>B</td>
<td>Painted</td>
<td>2/2</td>
<td>.31</td>
<td>.97</td>
<td>.56</td>
<td>.47</td>
<td>.51</td>
<td>.53</td>
<td>.60-.70</td>
</tr>
<tr>
<td>8&quot;</td>
<td>B</td>
<td>Painted</td>
<td>2/2</td>
<td>.74</td>
<td>.57</td>
<td>.45</td>
<td>.35</td>
<td>.36</td>
<td>.34</td>
<td>.40-.50</td>
</tr>
<tr>
<td>8&quot;</td>
<td>AA</td>
<td>Painted</td>
<td>3/3</td>
<td>.49</td>
<td>.48</td>
<td>.32</td>
<td>.32</td>
<td>.36</td>
<td>.46</td>
<td>.35-.45</td>
</tr>
<tr>
<td>8&quot;</td>
<td>BB</td>
<td>Painted</td>
<td>3/3</td>
<td>.60</td>
<td>.72</td>
<td>.56</td>
<td>.48</td>
<td>.46</td>
<td>.47</td>
<td>.50-.60</td>
</tr>
</tbody>
</table>

Type A (2 narrow slots) and Type AA (3 narrow slots) have no filler.

Type B (2 wider slots) and Type BB (3 wider slots) have a mineral wool filler.

The blocks are nominally 8" (20.3 cm) high and 16" (40.6 cm) wide. The test data was obtained from reverberation room tests at the acoustical laboratory of Geiger and Hamme, Inc., Ann Arbor, Michigan in accordance with ASTM C-423-66. The aggregate was expanded clay and approximately 75% light slag.
Sound Absorption of a Slotted Concrete Block

Installations have been made in auditoriums, music rooms, swimming pools, rifle ranges, factories, classrooms, gymnasiums, mechanical equipment rooms, churches, etc.

The durability of the product provides an acoustical wall treatment able to stand abrasion and abuse. In a gymnasium for example, the block are not injured by impact of basketballs, volley balls, etc. For wainscots from floor to door height, the Type A with narrow slots is commonly used. The Type B with wider slots is not suitable below 6 ft. (1.6m) to 7 ft. (2.1m) above the floor if there is a problem of persons placing pencils, cigarettes, etc., in the slots. This may occur in some schools or public areas.

Since there are locations where appearance and light reflection are not a factor, the block may be used unpainted. For this reason sound absorption tests have been made on some types unpainted.

By using this special block for its sound-absorbing qualities in conjunction with ordinary block, it is possible to use ordinary masonry construction to get a wall with a definitely calculated amount of sound absorption. One can balance the total absorption in a room by a combination of acoustical materials.

It is desirable in many rooms to have sound-absorbing materials on the walls as well as on the ceiling and floor. Note the papers by Fitzroy\textsuperscript{1} and Knudsen\textsuperscript{2} on this subject.

Further development is proceeding to achieve new designs of slotted block with varying sound absorption characteristics.

This is a patented design.

References


Hollow concrete blocks, with two or three cavities, are commonly used in the U.S.A. for wall construction. Lightweight porous aggregates are popular as this type of block provides a certain amount of acoustical absorption. This is advantageous in many rooms.

However, it is often desirable to have the concrete block wall provide good STL (Sound Transmission Loss). To attain the best STL of a porous block, it is necessary to seal the surface as by painting to reduce the surface porosity. Unfortunately, this is harmful in regard to the sound absorbing properties.

The following table gives the STL of an ordinary 8 inch hollow block of cinder aggregate as tested in 1954. The block was a nominally 8 inch (20.3 cm) high and 16 inch (40.6 cm) wide and the wall weight was 28.8 lbs. per sq. ft. (140 kg/m²) as laid up with mortar joints. The first line gives the STL of the unpainted block and the second line the STL of the block after being painted on both faces with one coat of cement base paint, using a coverage of 57 sq. ft. per gallon (1.4 m² per liter).

<table>
<thead>
<tr>
<th>SOUND TRANSMISSION LOSS</th>
<th>Decibels at Various Frequencies (Hz)</th>
<th>Average Loss</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>125</td>
<td>175</td>
</tr>
<tr>
<td>Unpainted</td>
<td>18.5</td>
<td>22.0</td>
</tr>
<tr>
<td>Painted</td>
<td>31.0</td>
<td>35.5</td>
</tr>
</tbody>
</table>
Sound Transmission Loss of a Slotted Concrete Block

It will be noted that the average STL increased 11.6 db, from 33.3 db to 42.9 db. However, the paint coat decreased the effective sound absorption of each face.

A new type of concrete block has two or three slots in one face which open into individual cavities. Since one end of the block is closed, each cavity becomes an individual cavity when the block is laid up in a wall with mortar joints. The slotted block is heavier than the ordinary block of the same size and aggregate, because the closed end and increased wall thickness add more weight than the slots subtract.

In order to test the STL, three sets of blocks were made up at the same time in the same plant with the same aggregate and portland cement mix. The blocks were a nominal 8 inch (20.3 cm) height and 16 inch (40.3 cm) width of the face, and 6 inch (15.2 cm) thick, to give a 6 inch wall. The Type A block had two narrow slots of different lengths into the two empty cavities. The Type B block had two wider slots of equal length into two cavities filled with a mineral wool batt. The ordinary block had no slots to the two continuous empty cavities.

![Graph showing Sound Transmission Loss (STL) in dB vs Frequency (Hz)]

STL-db 38 41 38 43 44 48 51 56 58 53 58

Figure 1. 6 inch Type A Slotted Concrete Block
Sound Transmission Loss of a Slotted Concrete Block

Figure 2. 6 Inch Type B Slotted Concrete Block

Figure 3. 6 Inch Ordinary Two-Core Block
Sound Transmission Loss of a Slotted Concrete Block

The STL tests were made in accordance with ASTM E90-61T by the Kodaras Acoustical Laboratories in New York City. The lightweight aggregate used was 50% Waylite and 50% sand. The walls on the unslotted side were painted with two coats of cement base paint before testing, conforming to normal practice. The slotted face side was purposely left unpainted.

It will be noted the Type A block had an STC (Sound Transmission Class) 49, the Type B an STC 47 and the ordinary block an STC 43. If a wall is laid up entirely in the slotted block, the slotted side will furnish a definite amount of sound absorption and the wall will have a higher STL than ordinary block would have. When a wall is laid up with both ordinary block and slotted block, the slotted block can furnish a predetermined amount of sound absorption and the overall STL will be better than if the ordinary block alone was used. Many walls are laid up in this combination fashion.

The slotted block are manufactured on standard block machines using special molds, so they can be made in quantity production. A number of blocks are cast at one time on these machines, which reduces the production cost.

The slotted faces give an overall appearance which has been received favorably by many architects. Although the blocks were originally designed for use in transformer enclosures and mechanical equipment rooms, their use has spread to rooms such as auditoriums, classrooms, churches, music practice rooms, gymnasiums, etc.

This new slotted block now makes it possible to build walls of concrete masonry having good STL and predetermined sound absorption qualities relatively unaffected by painting the surface.
Sound Absorption Characteristics of an Acoustic Ceramic Tile

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Masayasu Ukai
Associate Professor of Nagoya Institute of Technology

Introduction

Most of the sound absorbing materials available for use at present are rather weak from resistivity against fire and water by their nature, and therefore are not usable in place access to high temperature or of high humidity. The principal investigator and his colleagues have produced a porous acoustic tile by way of experiment from ceramic materials to study its acoustic properties and its related properties for building materials. Further, with a view to assuring its effectiveness in practical use the tile was applied in actual construction work for controlling reverberation and noise effect.

General properties of Acoustic Ceramic Tile

The acoustic ceramic tile is made of ceramic materials which are at first crushed into granule to be mixed with heat-proof binders and then moulded for firing at 1200~1300°C. The tile is a kind of porous material having continued inner space at a volume rate of 30~40 %, the apparent specific gravity of which is rated at 1.15 g/cm³. When used together with heat-proof mortar cement, the tile is able to withstand for high heat of 1000°C. It is washable with water and free from freezing owing to its continued air cell composition.
Sound Absorption Characteristics of an Acoustic Ceramic Tile

Test Piece

The following test tiles (30 x 30 x 2, in cm) were produced for analysing their properties:

a. Moulded (00)

b. Applied with glaze by random spraying (HO).

c. Applied with glaze on the face excepting 1~5 mould face of 2 cm wide stripes (10), (20), (50).

The notation of test piece show the condition of surface and reversed side of the tile. The first symbol show the condition of surface and second symbol show the reversed side.

For example,

Symbol 0; show the moulded surface non-glazed.

Symbol 1~5; show the number of stripes which is non-glazed.

<table>
<thead>
<tr>
<th>N</th>
<th>p</th>
<th>N</th>
<th>p</th>
<th>N</th>
<th>p</th>
<th>random spraying</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>6.7%</td>
<td>2</td>
<td>13.3%</td>
<td>5</td>
<td>33.3%</td>
<td>100%</td>
</tr>
</tbody>
</table>

*N=number of stripes  
P=perforation degree %

Fig. 1

Table 1

![Fig. 1](image)

![Fig. 2](image)

![Fig. 3](image)
Sound Absorption Characteristics of an Acoustic Ceramic Tile

Symbol $H$; show the surface condition of random spraying.
And continued number show the air space in cm (Table 1).
These tiles are turned over for rather analysis (OH), (01), (02), (05). (Table 1, Fig. 1)

Test Results on Absorption Coefficient of Reverberation Chamber Method

In Figs. 2-7 are shown the results on absorption coefficient of reverberation chamber method based on ISO standard. Absorption coefficient of each test piece influenced by the changes in the respective air space is shown in Fig. 2-7, where a tendency is observed to show that the peak of absorption coefficient moves to low frequency range as the air space is increased, but to high frequency range as the perforation degree (or the surface area where glaze are not applied) is increased. Accordingly, it may be safely believed that by adjusting the extent of

--- E-107 ---
Sound Absorption Characteristics of an Acoustic Ceramic Tile

glazed area on surface the tile shall have the same effects as obtained by changing the perforation degree of perforated panel, and further that by virtue of its porosity it will exhibit acoustic properties similar to the one made of perforated panel in combination with fibrous porous materials. It is also expected to absorb sounds in high frequency range by facing its glazed surface backward.

The matters on control of sound absorbing properties are still left open for further study, where relationship between condition of granule of crushed ceramic materials that are made into this tile and the acoustic property should be clarified.

Matters on Practical Use of Acoustic Ceramic Tile

With a view to ascertaining the effect of the tile in practical use, the principal investigator the tile have been set on the wall in bath-room and small machine room, obtaining satisfactory results.

This tile having such properties as fire-proof and water-proof, is expected to be used widely not only as a general acoustic material, but also for noise control equipments of outdoor as well as industrial noise control material.

The principal investigator and his colleagues will endeavour to improve its absorption properties and matters to make easy the execution of work.
Frequency Considerations in the Subjective Assessment of Sound Insulation

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Division of Building Research
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Introduction

Modern building technology, turning increasingly to the use of light materials and prefabricated elements, has solved many of the attendant problems such as strength, durability and fire resistance, but there remains the major problem of adequate sound insulation. It is therefore becoming increasingly important to establish a reliable way of rating the sound insulation value of a wall or floor construction.

There are, of course, many regulations governing sound insulation, but an examination of their origins reveals that they are not based on much concrete acoustical evidence. The most realistic approach was to identify by social surveys certain constructions where the sound insulation was deemed adequate by the majority of the occupants. Then recommendations were devised that reflected the characteristics of those particular constructions. Several parallel developments of this sort, in Europe and in North America, culminated in a proposal for an international standard for the rating of sound insulation between dwelling units. Essentially the same scheme is already in use in several European countries and in North America, where it bears the designation "Sound Transmission Class" (STC).

This rating system tries to compress into a single number all the characteristics that may affect the ability of a wall to provide acoustical separation between two occupancies in a building. The process by which the present requirements have developed was a rough, empirical one, but the method that finally evolved is itself very precise. It begins with measurements of the sound transmission loss of the
Frequency Considerations in the Subjective Assessment

wall or floor, for a series of octave or third-octave frequency bands. The results are plotted as a graph of transmission loss (TL) against frequency, and compared to a standard contour (the STC contour) which may be thought of as an ideal TL curve that gives appropriate weight to the TL at each frequency. The comparison is made by adjusting the vertical position of the reference contour on the graph so that a limited portion of the TL curve falls below the contour. The permissible extent of this "deficiency" of the TL curve is carefully specified, as will be noted later.

It would be fortunate if this precise determination were backed by an equally precise understanding of the acoustics and psycho-acoustics involved, but this is not so. A number of people, including ourselves, have looked at the problem, but all have invoked serious assumptions, especially about the subjective side of the process. The present paper reports some systematic experimental studies of these subjective aspects.

Procedure

In any sound insulation problem there are three primary factors: - (1) the character of the intruding noise; (2) the character of the "ambient" noise (i.e., the noise that the occupants find an acceptable part of their environment); and (3) the occupant's assessment of the annoyance caused by the noise. It is the third of these factors that is the principal object of this study.

In brief, each of some 30 subjects was installed in a controlled environment, with a certain level of ambient sound, and presented alternately with two intruding noises (actually, the same noise transmitted through two frequency-weighting networks). He was instructed to make an adjustment in the level of one intruding noise until he considered the two sounds to be equally annoying. The level adjustments then provided a basis for comparing the effects of the two different networks.

An anechoic room was selected as the test environment. The ambient and intrusive noises reached the subject by way of two loudspeakers, placed one over the other and about 3 metres in front of him.

The ambient noise was random noise modified to have a spectrum corresponding in frequency and level to an NC-25 curve. The intruding noise began with a tape loop of a typical signal such as speech, and passed through either of the two frequency-weighting networks, which could be set to simulate, for example, the transmission loss of a wall. The level of intruding noise was set so that it was slightly perceptible over the ambient noise. Thus for speech a level appropriate for a 5
percent articulation index was aimed at. The subject could switch manually from one network to the other, or it could be done automatically at intervals of 2 or 3 seconds, as he made the necessary level adjustment in one channel.

To deal with the semantic problem posed by the word "annoyance," the observer was asked to imagine himself, for example, sitting in his living room reading a book or playing chess, and to gauge the annoyance of the intruding noise in such a situation.

Results

The exact form of the results depends on the spectra of intruding and ambient noise as well as on the subjective reactions. Their general form will be illustrated by the results for one particular set of conditions: intruding male speech against an ambient noise spectrum corresponding to NC-25.

All subjective assessments in this group were made with one weighting network adjusted to simulate the STC reference contour. The second channel varied in shape with the particular experimental objective.

(a) Variation of annoyance with frequency - A question of major interest is the relative importance of various frequency bands of the intruding noise. Accordingly, the observers were asked to compare octave and third-octave bands of the speech signal with the reference signal transmitted through the STC network. The results for one of these cases (third-octave bands) are shown in Fig. 1, where relative levels giving equal annoyance are plotted against frequency. The vertical range shown for each point corresponds to the standard deviation of the mean.

For comparison an STC contour is superimposed at approximately the position of best fit. The shape of the STC contour constitutes, in the normal rating process, an assumed variation of annoyance with frequency; it is therefore of interest to compare the STC assumption with actual subjective judgments.

The agreement is astonishingly close, and it seems necessary to emphasize
that these are the results for only one kind of noise (male speech) and for one spectrum of ambient noise. At least, however, it may be concluded that the shape of the STC contour is reasonably correct.

(b) Variation of annoyance with narrow-band signal level - The second element in the normal rating process is the method of comparing the shapes of an actual TL curve and the STC contour. The basic procedure simply is to allow an average deficiency of 2 dB regardless of how it is distributed with frequency; but there is an overriding limit of 8 dB for any one frequency band. The latter requirement usually comes into play only when there are narrow dips (e.g., coincidence dips) in the TL curve. An experiment was set up to simulate this situation.

A smooth TL curve, closely resembling the STC contour, was set up in the second channel, together with an auxiliary network that permitted the introduction of a dip either an octave or third-octave wide and variable in depth from 0 to 20 dB relative to the smooth curve. These simulated TL curves were then compared with the STC contour in the reference channel.

The results indicate that the change in subjective rating as a dip increases in depth is rather small. This is consistent with the average-deficiency part of the STC rating procedure, but not with the 8 dB maximum-deficiency limit. It appears that, for intruding speech, even a deep dip in a TL curve should be assessed approximately in terms of the total area of the deficiency, and the 8 dB limit for an individual band is inappropriate.
Introduction

This paper has to do with the required airborne sound insulation between dwellings. In an earlier paper [1] we stated that:

- when being in one's own dwelling, it must be possible to concentrate on mental work, without being disturbed by neighbour's radio

- when being in one's own dwelling, it must also be possible to adjust one's radio, gramophone or tape recorder (special hi-fi sets are not included) to the level one prefers for good listening conditions, without being afraid to disturb the neighbours.

As explained in that paper, we found that for these conditions the airborne sound insulation between adjacent dwellings should be 6 to 9 dB better than that of a solid brick wall of about 400 kg/m² (normal flanking conditions), assuming parallel shifting of the insulation curve.

It appeared interesting to check whether the levels chosen by the subjects for radio programs are surpassed by the levels preferred for television programs.

It is often said that as a rule the television is played on a higher level than the radio. This might be true when comparing the television level with the general radio level, as radio music is often used as an acoustic background, whereas the television sound is usually turned on only if one is watching the program. Our investigation on the required airborne sound insulation between dwellings, however, was not based on the general radio level, but on the level for attentively listening. We wanted to know whether this level is surpassed by

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1) Publication no. 303 of the Research Institute for Public Health Engineering TNO (TNO = Netherlands Organization for Applied Scientific Research)
television levels. If this would be the case, the value aimed at for the air-borne sound insulation between dwellings should be even higher than the one mentioned above.

Procedure

A television set was placed in a relatively quiet room in the laboratory. The dimensions of this room are: 4.00 m wide, 4.00 m long and 2.65 m high. In order to achieve a living room atmosphere the interior was furnished with a carpet on the floor, a curtain on one of the walls, some easy chairs and a small table.

Of 20 subjects, one at a time was shown a television program of about 40 minutes. This program had been recorded on a video recorder and consisted of 5 parts:

1. News 9 minutes, followed by an interval of 2 minutes
2. Film 13 " " " " " 1 minute
3. Cabaret 3 " " " " " 1 "
4. Light music orchestra 7 " " " " " 1/2 "
5. Show 11 "

The video recorder was installed outside the experimental room. The chair of the subject was about 3 meters away from the television set (53 cm screen).

The 20 subjects, between 16 and 65 years of age, were employees of our Institute, but not working in the Department for Sound and Light. They all had sufficient t.v. experience. The subjects got a written instruction. They were asked to adjust sound level, brightness and contrast at will. They were told (as during the radio experiment) that they should not bother about disturbing people outside the room, as it was not likely that anybody could hear them. Also it was said that although it was an experiment, no special accuracy in the adjustment was asked for. Subjects were not informed about the background of the experiment.

Before the beginning of each part of the program the sound level was turned low (from outside the experimental room), so that the subjects had to adjust each part of the program separately.

Results

The sound of the recorded programs was analysed with our automatic analyser for fluctuating sound [2]. For each of 8 octave bands the level was determined which is not surpassed during 90% of the time. Of course this level depends on the volume chosen by the subject. As we want to protect more than just 50% of the people, we did not consider the mean of the chosen volume adjustments, but the mean plus one times the standard deviation, i.e. the volume adjustment not surpassed by 84% of the subjects.

This level which was not surpassed during 90% of the time by 84% of the subjects, we call L_{90}. This L_{90} is determined for each of the 5 fragments of the
program. The Table gives the mean over the 5 fragments, $\bar{L}_{90}$, and the corresponding standard deviation $s(L_{90})$. In the Table are also included the levels for attentively listening to a radio program, found during an earlier investigation.

The analyser dates from before the ISO-Recommendation for preferred frequencies. This is the reason of the non-standard midfrequencies.

<table>
<thead>
<tr>
<th>Octave band</th>
<th>mid-frequency H(\text{Hz})</th>
<th>100</th>
<th>200</th>
<th>400</th>
<th>800</th>
<th>1600</th>
<th>3200</th>
<th>6400</th>
</tr>
</thead>
<tbody>
<tr>
<td>Television</td>
<td>$\bar{L}_{90}$ in dB</td>
<td>55</td>
<td>58</td>
<td>59</td>
<td>53</td>
<td>49</td>
<td>50</td>
<td>44</td>
</tr>
<tr>
<td></td>
<td>$s(L_{90})$ in dB</td>
<td>5.4</td>
<td>4.3</td>
<td>3.7</td>
<td>3.1</td>
<td>3.5</td>
<td>4.8</td>
<td>4.5</td>
</tr>
<tr>
<td>Radio</td>
<td>$\bar{L}_{90}$ in dB</td>
<td>61</td>
<td>69</td>
<td>77</td>
<td>75</td>
<td>71</td>
<td>66</td>
<td>56</td>
</tr>
<tr>
<td></td>
<td>$s(L_{90})$ in dB</td>
<td>8.2</td>
<td>5.4</td>
<td>1.9</td>
<td>4.0</td>
<td>3.7</td>
<td>6.2</td>
<td>6.1</td>
</tr>
</tbody>
</table>

Discussion

It is clear that the television levels are strikingly lower than the radio levels. This would lead to the conclusion that the extra sound insulation of 6 to 9 dB above that of a wall of 400 kg/m², which is advisable according to the radio experiment, would be also sufficient to protect people against their neighbour's television.

Let's now consider a number of possible objections.

The most serious objection perhaps is that each test subject was alone in the testroom. According to the original design at least two people would have been in the testroom. It is reasonable to suppose that in that case a somewhat higher level would have been chosen. Due to a coincidence of circumstances the experiment in fact was not carried out along these lines. It seems unlikely, however, that the effect of more people in the room would totally cancel the level differences shown in the Table, or even reverse them.

An objection closely related to the one just mentioned, is that the background level in the t.v. room was only about 26 dB(A), whereas the background level in the radio room was about 45 dB(A). This is an appreciable difference, comparable indeed with the differences found between the radio- and t.v.- levels. As, however, the levels actually chosen for radio and for t.v. are so much higher than the corresponding background levels, I doubt very much whether equal background levels would have led to equal radio- and t.v.- levels, although some influence is likely.

Another possible objection could be that after the experiment it turned out that about 60% of the subjects said to be sure or to think that the experiment had to do with sound annoyance; resp. with the establishment of airborne sound.
insulation requirements. These ideas could well have influenced the volume adjustment, but it is not at all obvious that this would systematically result in the choice of a relative low level.

Still another objection might be, that the sound level for each part of the program started at zero: as a result the subject always adjusted the level in the direction from low to high.

One subject said that for classical music he would probably have chosen a higher volume.

Each of the factors mentioned might influence the result. Nevertheless the differences between the levels preferred for t.v. and those for radio are such that we decided not to try to find out the magnitude of each of these influences. I think that there is no convincing material to believe that t.v. levels as a rule are higher than the levels preferred for good listening conditions to radio programs, so that the advisable airborne sound insulation based on the latter will probably also give a sufficient protection against the sound of neighbour's television.

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(Die Stimme der Memnon - Kolosse)

D. Schorr

Auftraggeber: Georg-Magnus-Gesellschaft
Düsseldorf
Einleitung:
In der Nähe der Stadt Luxor befinden sich auf der westseite des Nils 2 ca. 18 m hohe Sitzstatuen, bekannt als die sog. Memnon-Kolosse. (Fig. 1) Diese wurden von Pharao Amenophis III (1400 - 1362 v. Chr.) erbaut und flankierten ursprünglich den Eingang eines Amen-Tempels, der aber nicht mehr erhalten ist. Die Kolosse bestehen aus würzit-Monolithen und stehen auf einem Sandsteinsockel. Ihre ursprüngliche Höhe betrug 21,6 m mit einem Gewicht von ~ 1000 t. Der nördliche Koloss wurde in den Steinbrüchen nahe Kairo gebrochen und auf dem Nil nach Luxor transportiert. Nach Strabo (1) wurde der nördliche der beiden Kolosse durch ein Erdbeben, wahrscheinlich durch das bekannte Ebeben 27 v. Chr. zerstört und brach oberhalb der Gürtellinie ab, seit dieser Zeit konnte man jeweils nach Sonnenaufgang geheimnisvolle Laute an den zerstörten Koloss wahrnehmen. Da inzwischen die Geschichte dieser Statuen in Vergessenheit geraten war, dachte dies die Sage als Quell von König Memnon an seine Mutter Cose.


Von seiten der Pythologen sind verschiedene Deutungsversuche bekannt geworden: "Da die Erwärmung am Morgen, ergraben von der Bruchstelle, die ursprünglichen Teile ausgespalten, ab, die Ton von Hörleuten." (2) "Daß diese Laute durch den Wechsel von Temperatur und Luftfeuchtigkeit in einer Bruchspalte der Figur zustande kommen" (3) "Daß sich die lose aufeinander gefügten Steine infolge des raschen Wechsels der Temperatur und Luftfeuchtigkeit bei Sonnenaufgang ausdehnen, zu Arbeiten beginnen und leicht trennen". (4)


Die "Stimme des Memnon"


In Fig. 2 sind Beobachtungszeiten über dem Kalenderdatum derjenigen Inschriften abgetragen, deren Daten vollständig bekannt sind. Die
Zeitangabe bezieht sich auf die Zeit nach Sonnenaufgang. Zunächst fällt auf, daß fürs Sommerhalbjahr nur wenige Inschriften existieren. Da aber die Kolosse im Überschwemmungsgebiet des Nils liegen, wären in dieser Zeit nicht zuletzt auch der Hitze wegen die Kolosse sicher weniger ansässig. Auf der andern Seite gibt es auch im Dezember und Januar keine Eintragungen. Um einen weiter gehenden Zusammenhang mit der Sonne herauszufinden, sind in Fig. 3 a – c die Zeit des Sonnen- aufgangs in ÄTZ, die Sonnenhöhe und die Sonnenrichtung abgebildet. Leider ist aber die Zahl der Beobachtungen zu klein um außer der Tat- sache, daß der Toneffekt 1 – 2 Stunden nach Sonnenaufgang auftrat, eine differenziertere Korrelation herauslesen zu können.

**Toneffekte.**

Im folgenden sollen verschiedene Toneffekte zusammengestellt werden, die zur Erklärung des Memnon-Phänomens herangezogen werden können.

1. Windeffekte
   1.1 Helmholtz-Effekt: Anblasen eines Hohlraumes.
   1.2 Aeolis-Effekt: Prinzip der Aeolis-Harfe.
   1.3 Schneiden - Ton: Anblasen einer Schneidkante.
2. Thermomechanische Toneffekte.
   2.1 Reihungstöne: Tonentstehung bei Verschleißung durch Wärmedehnung.
   2.1 "Zinn - Geschrei"- Effekt: Tonentstehung bei mechanischer Ver- formung infolge Wärmedehnung.
   2.3 Bruchlaute: Gerauschentstehung bei Bruchbildung infolge thermi- scher Spannungen.
3. Thermische Effekte (6)
   3.1 Trevorian-Effekt: Wackler-Schwingungen durch verschiedene heiße Körper.
   3.2 Erhitzungstöne: Erhitzung eines resonanzfähigen Hohlraumes.
   (Pinsud)
   3.3 Rijke-Effekt: Partielle Erhitzung eines zylindrischen Hohl- raumes –. (Netztöne)
   3.4 Blasinstrument - Ton: Tonentstehung durch Ausblasung eines Luft- stromes (oder Jampf) aus einem abgeschlossenen, erhitzten Volumen.


Im unteren Teil der Kolosse befinden sich eine Reihe von durchgehenden Rissen, wobei die Bruchflächen bis zu 20 cm voneinander entfernt
Die Sound of the Colossi of Memnon.


Meinem verehrten Lehrer, Herrn Prof. Spandöck (25.11.66) danke ich für die Unterstützung und väterliche Betreuung der Untersuchung. Diese Arbeit soll seinem Andenken gewidmet sein.

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**Fig. 2**
Beobachtungszeiten

**Fig. 3a**
Sonnenaufgang

**Fig. 3b**
Sonnenhöhe
Sound Radiation of Metal Panels with and without Beams

Die Schallabstrahlung von Blechwänden mit und ohne Rippen

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The sound transmission loss of two aluminium panels has been tested as they are used in superficial structures of ships (see Fig. 1). The beams welded on the panels run parallel in one direction and in a distance of 40 cm. On one of the two panels, additional stiffeners are fixed in the same distance, perpendicular to the beams, and in single parts. Besides the panels with beams, an aluminium panel of the same size and thickness (4 mm) but without beams has been tested. The panels had the dimensions 2.7 m x 3.4 m. They were screwed on an angle iron frame (75 mm x 100 mm x 7 mm) cemented into the corresponding opening between two test rooms. The measurements were carried through according to ISO Recommendation R 140. The test sound was white noise of third octave bandwidth, radiated by a loudspeaker.

Fig. 2 shows the sound transmission loss of the three panels versus frequency. While the curves coincide rather well between 200 and 500 Hz, differences are stated up to 10 dB between the panel without beams and the cross-wise stiffened panel in the range from 1000 Hz to the coincidence cut off frequency 3200 Hz, in spite of a specific mass of 15 kg/m² for the stiffened panel as against 11 kg/m² for the bare panel. The sound transmission loss values of the simply ribbed panel lie between the two other curves.

To explain this behaviour of the panels more in detail it is useful to
Sound Radiation of Metal Panels

Fig. 1 Panels with beams;
- a. crosswise stiffened
- b. simply ribbed

Fig. 2 Sound transmission loss;
- 1. measured with unribbed panel,
- 2. with crosswise and 3. with simply stiffened panel; a, b. calculated after Heck's theory.

display the sound transmission loss R of the panel as the product of the inverses of the response ε and the radiation factor σ:

\[
R = -10 \log \varepsilon - 10 \log \sigma = 10 \log \frac{\frac{p_1^2}{\varrho v^2 (2\pi f)^2}}{p_2^2} + 10 \log \frac{\frac{v^2 (2\pi f)^2}{\varrho_0 c V^2}}{T_2 S \frac{0.16}{V_2}} \cdot \frac{T_2 S}{0.16 \cdot V_2}
\]  (1)

\(\frac{p_1}{p_2}\) = space averaged squared sound pressure in the loud and the quiet room respectively; \(\varrho v^2\) = mean squared velocity amplitude of the flexural waves of the panel; \(\varrho, c\) = density and sound velocity of the air; \(T_2, V_2\) = reverberation time and volume of the quiet (receiving) room; \(S\) = area of the panel without ribs. The velocity amplitude of the panels has been measured by an accelerometer (Bruel & Kjaer) of 2 g weight. From \(v\) and \(p_2\), simultaneously measured, \(\sigma\) has been calculated and displayed in Fig. 3, both for panels with and without beams. The values result as the averages of 15 points randomly distributed over the panels. At the panels with ribs, additional values were taken on the flanks of the beams. The average of all points does not deviate from that of the panel points more than 2 dB.

With the usual test set up for measuring sound transmission loss, the vibrations of the panel are excited by a random sound field. That means, forced
vibrations as well as free vibrations arise. In order to get only free vibrations, which meet theory better, the panels were excited also by an electrodynamic shaker (Philips PR 9270), and the mean sound pressure level in the receiving room was measured in the same way as above. The resulting radiation factors are also plotted in Fig. 3. Little difference is noted for the two kinds of excitation at the panels with beams in comparison with the bare panel where the difference is e.g. about 6 dB at 1000 Hz. From the fact that the radiation factors of the panel with and without beams at random sound field excitation do not differ essentially, can be taken that the response of the panels with beams is higher than that of the unribbed panel. That follows from equ. (1). 10 log $\xi$ is plotted versus frequency in Fig. 4.

The theory, as it is given by several authors (Smith, Lyon, Maidanik,
Sound Radiation of Metal Panels

Heckl, Josse, Lamure), shows the coherence between response and radiation. Heckl deduces it by reciprocity considerations. If the here measured values of the two radiation factors for the panel without beams are inserted into equ. 35 of [1], the sound transmission loss calculated in this manner agrees well with the directly measured one (see Fig. 2). The crosswise stiffened panel shows good agreement too, except a small range near coincidence. In Fig. 3 also a calculated curve of the radiation factor, deduced by Maidanik [2] for point excitation, is plotted. It runs some decibels below the experimental curve.

Further comparison of measured and calculated values was made concerning the response curves in Fig. 4. Equ. (8) in [3] and equ. (34) in [1] were used and as far as necessary the measured $\sigma$-values inserted. At the panel with crosswise beams, the dimensions of a subpanel were used for the term perimeter divided by area. The damping factor was taken from decay constant measurements with shaker excitation. It has values of about $5 \cdot 10^{-3}$ at the panels with beams and about $10^{-2}$ at the unribbed panels. The calculated response values are somewhat lower than the measured ones at the unribbed panel, and fairly coincide at the ribbed panels. The differences may be explained by the fact that theory has been developed for simply supported panels whereas the measurements were made with clamped panels as far as it concerns the boundary conditions of the total surface edge.


Measurements of the vibration Correlation
between Both Surfaces of a Multi-layer Panel.

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Introduction

Multi-layer panels vary in their airborne sound insulation characteristics since the differences of compositions and materials among them result in a variety of their sound insulation mechanism.

Also from the practical point of view, for example, concerning the studies on effective ways of their applications to practical uses or the improvement of their sound insulation properties etc., it is needed to analyse the mechanism of sound insulation properties for actual products of panel structures.

However, it is not always easy to make out the sound insulation mechanism of multi-layer panels only through their constructions and the frequency characteristics of their transmission loss curves.

We have used empirical methods to bring out the meaning of the sound insulation characteristics of panels comparing the results of the transmission loss measurement with that of an additional measurement putting porous material or liquid layer on the test panel. As shown in Fig. 1 we can distinguish the causes of minima on the transmission loss curve by this method.

Following such methods we have tried another kind of method in regard to the relation between the vibrations of both panel surfaces.
Vibration Correlation between Both Sides of a Panel

Method

The cross correlation of the vibrations between both panel surfaces is measured under the same conditions of panel setting and incident sound field as the usual transmission loss measurements.

Light vibration pickups (Acceleration-type, 1.5 gr.) are fixed with double tacked tape on the just opposite surface points of each side of a test panel. Output signals from them are led into a correlator which is similar to the one used for our studies on the space correlation of reverberant sound fields.

Examples of Application

Two ideal cases are easily expected for the relation between panel vibration properties and the results of correlation measurements, i.e. if both surfaces of a panel vibrate together as a thin simple plate the value of the cross correlation R must be +1, and if a panel forms a resonant system which can be simulated to a so-called Mass-Spring-Mass simple mechanical resonant system, R may keep +1 up to the resonant frequency and at that frequency it may change +1 to -1.

Such exceptions are verified as shown in Fig. 2 - Fig. 4 and by following ones it is proved that this method can clarify the differences of vibration properties among multi-layer panels and is available to be applied to the analysis of the sound insulation properties of them.

By the cases of Fig. 5 - Fig. 8 the results of this measurement suggest us the actual state of panel vibration for which we may feel difficulties if another way of observation is used.

In the remaining examples interesting characteristics of cross correlation are shown. We need more studies to read into these results.

Besides those, concerning about the vibration, say furthermore, the sound insulation properties of panel structures, we hope that we shall be able to receive yet more informations from the results of this method.

To conclude the report we should like to express our sincere appreciation to Mr. H. OGAWA for his cooperation to our works.

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Vibration Correlation between Both Sides of a Panel

The liquid Method

The Correlation Method

Construction and Note

Fig. 1
- 4 cm
- water layer ◦ 1
- vynil sheet
- test panel; same
- as the example of Fig. 3.

When water layer is added on a test panel following changes are observed for sound insulation properties:

The "resonance" dip shifts to lower Hz frequency, the "coincidence" dip vanishes.

Fig. 2 Plywood core, 12 mm thick, plywood 6 mm pasted on both sides.

In the measured frequency range both surfaces vibrate together like a thin simple plate. the dip is caused by the coincidence effect.

Fig. 3 Moulded styrol foam core 45 mm, gypsum boards 12 mm on both sides.

The frequency for the sudden change of R occurs to the "resonance" dip on the TL curve. But in the higher frequency the expectation from such a simple simulation goes different from the actual state.

Fig. 4 Paper role core 48 mm, steel sheets 0.8 mm on both sides.

Except in the highest frequency region the core material is considered to bind both surfaces rigidly. The broad dip is to be related to the coincidence effect.

Fig. 5 Particle board 40 mm.

Shaven wood is pressed in layers parallel to the surface, the texture is not uniform and it becomes gradually finer toward the surfaces. In this case the dip on its TL curve is related to the change of R, in fact, this panel seems to have sound insulation properties of a resonant system.
Vibration Correlation between Both Sides of a Panel

Fig. 6 Two leaves of plywood 6 mm nailed on both sides of wooden frame 25 mm.

The results of the correlation measurements for three points on the test panel differ each other in the frequency lower than that of the well-known resonance. A simplified model of vibrating panel cannot be applied there.

Fig. 7 Plywood core 12 mm, plywood 6 mm nailed (distance 30 cm) on both sides.

Sometimes nailed cover layers are effective for the improvement of the "coincidence" dip of a core panel. This panel seems to be regarded as a damped resonant system. Since the condition of the inner surface contact is not everywhere same, the resonant frequency depends on positions.

Fig. 8 Foam styrol core 25 mm, asbestos cement sheets 3 mm on both sides.

The large variety of the resonant frequencies is remarkable. This panel seems to be regarded as a multi resonant system. The broad width of the dip may come from that.

Fig. 9 Paper role core 30 mm, gypsum board 9 mm on both sides of core, and steel sheet surfaces 0.8 mm.

The dip of this TL curve is to be related to that of the highest frequency range in Fig. 4, and is caused by a certain kind of the resonance. But the results show a clear difference in vibration properties from a simple mechanical model or the case in Fig. 3. Such properties of R are found similarly in the last three examples while they are different each other in their sound insulation characteristics and outside views of the vibration phenomenon.

Fig. 10 Corrugated asbestos cement board core 34 mm, asbestos cement sheets 4 mm on both sides.

The dip is caused by the resonance of sectional sheet vibration and the change of R corresponds to it. The change of R is +1 Hz to 0 and in the higher frequency R keeps 0.

Fig. 11 Wood-shaving board (Portland cement binder) core 25 mm, asbestos cement sheets 3 mm on both sides.

Because of the harsh texture of the core material and the pasting manners in the producing process the contact of the core and the surface plates is not always rigid and the structural coupling path between both surfaces may be complex.
Zur Schalldämmung leichter zweischaliger Trennwände

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Die Schalldämmung leichter zweischaliger Trennwände ist theoretisch nur schwer erfaßbar, und selbst durch Vergleich ist das mittlere Schalldämm-Maß R einer neuen Konstruktion im voraus nur selten auf ± 2 dB genau anzugeben. Diese Unsicherheit ist bedingt durch
1. konstruktive Details der Trennwand (Art und Verteilung der Schalbrücken u.ä.),
2. durch die Güte des Aufbaues (Dichtigkeit der Randanschlüsse und Puren),
3. den Einfluß der Nebenwegübertragung in einer speziellen Einbausituation.

Bei homogenen - das sind vorwiegend schwere Konstruktionen - spielen 1. und 2. keine Rolle, während man 3. durch einheitliche Meßverfahren auszuschalten versucht /1/, wozu auch die Prüfung in Räumen mit festgelegter maximaler Schalldämmung (Laboratorien mit "bauüblichen" Nebenwegen) gehört.

In den letzten Jahren haben sich bestimmte Techniken und Konstruktionsprinzipien herausgebildet, die eine hohe Schalldämmung für leichte zweischalige Trennwände garantieren. Deshalb wurde versucht, das mittlere Schalldämm-Maß und das Luftschallschutzmaß nach DIN 4109 /2/ derartiger Wände allein durch den Konstruktionstyp zu kennzeichnen. Angaben über Gewichte sowie Schalenabstand wurden nur zur Festlegung des Gültigkeitsbereiches eines derartigen Katalogs verwendet. Dabei wurde von folgenden Prinzipien ausgegangen:

1. Es müssen so viele Typen unterschieden werden, daß die Unsicherheit im mittleren Schalldämm-Maß pro Typ weniger als ± 3 dB beträgt.
2. Leichte Trennwände sind Wände mit einem Flächengewicht m² < 150 kg/m² und einer besseren Schalldämmung als die gleichschwerer einschaliger Wände.

Es erweist sich als zweckmäßig, nach folgenden Wandgruppen zu unterscheiden:

1. Einschalige Grundkonstruktionen
Zur Schalldämmung leichter zweischaliger Trennwände

1.1 Biegeweiche Platten (d < 20 mm und m' < 20 kg/m²) Material: Gips, Gipskarton, Holzspan, Stahlblech, Glas
1.2 Biegesteife Platten (d > 35 mm und m' > 35 kg/m²) Holzspan, Stahldraht, Glas

2. Zweischalige Grundkonstruktionen, Platten nach 1.1, wie folgt verbunden
2.2 ganzzählig, steif, s' > 15 kPa/cm²
2.2 ganzzählig, weichfedernd, s' < 10 kPa/cm²
2.3 durch einzelne steife Stiele, Stielabstand bzw. dicke: a > 50 cm, d > 40 mm
2.4 durch einzelne weichfedernde Streifen, a > 40 cm
2.5 durch federnde Metallstreifen +)

3. Grundkonstruktionen mit Vorsatzschale
3.1 befestigte Vorsatzschale, z. B. an 2.1 bzw. 2.5
3.2 freistehende Vorsatzschale, z. B. vor 2.5
3.3 befestigte Vorsatzschale, an Platten nach 1.2
3.4 freistehende Vorsatzschale, vor Platten nach 1.2

4. Zweischalige Konstruktionen (Kopplung der Wandschalungen nur über bedämpften Luftzwischenraum)
4.1 Platten nach 1.1 mit Stützkonstruktionen
4.2 Platten nach 1.2, freistehend errichtet
4.3 Grundkonstruktionen nach 2.1, freistehend errichtet

Bei dieser systematischen Auswertung ergaben sich Hinweise für die praktische Konstruktion von leichten Wänden hoher Schalldämmung:
1. Die Schalldämmung von Wänden aus steifen, schweren Wandschalungen läßt sich im Mittel durch Randisolierstreifen um 3 dB verbessern. Dieselbe Maßnahme führt bei steifen einschaligen Wänden zu einer Verschlechterung der Dämmung.
2. Bei frequenzabhängig bewerteten Mittelwerten (LSM) ist es sinnvoll, durch Verstärkung der Wandschalungen das Gewicht einer Wand durch dickere Schalen zu erhöhen und einen gewissen Koinzidenzefizienz hinzunehmen.
3. Jede zweischalige Trennwand nach Typ 4 besitzt mehr als 45 dB mittlere Schalldämmung, wenn die Plattendicke d > 10 mm und das Gewicht einer Wandschale m' > 12 kg/m² betragen.
4. Mit leichten Wänden (m' < 100 kg/m²) lassen sich in einem Prüfstand mit genormter Nebenwegübertragung (Rₙₐₓ = 58 dB bzw. LSMₘₐₓ = + 6 dB) erwartungsgemäß nicht mehr als 54 dB Schalldämmung (LSM = + 3 dB) erzielen. Das bedeutet, die Übertragung des Luftschalles erfolgt etwa zu gleichen Teilen über die flankierenden Bauteile und über die Trennwand. Für die Entwicklung besserer Trennwände kommen also nur noch Prüfräume ohne Nebenwegübertragung infrage, da sonst keine Differenzierung der Wandkonstruktionen mehr möglich ist. Unter baupraktischen Bedingungen, insbesondere bei leichten flankierenden Bauteilen, durchlaufenden Estrichen und Untertecken u. ä., wird dann jedoch die Schalldämmung geringer sein als im Labor.

Gerade aber dieser letzte Umstand ist für die Praxis von großer Bedeutung, da die flankierenden Bauteile vielfach gleich oder mindestens ähnlich ausgeführt sind, wie die infrage stehende Trennwand.

Durch einen Einbau der Prüfwand entsprechend den unten skizzierten Bedingungen erhöhen sich die Schalldämmungswerte in der Praxis in der Regel wesentlich.

+ Sonderkonstruktionen, bei denen aufgrund der Querschnittsform die Steifigkeit besonders gering ist.

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scheint es möglich, auch durch Messungen im Labor, d. h. unter kontrollierbaren Einbaubedingungen, charakteristische Grenzwerte für den Bereich der Schalldämmung einer gegebenen Bauart anzugeben.

Zweischalige Trennwand aus Gipskarton-Wabenelementen
Flächengewicht ca. 45 kg/m²
Gesamtdicke 140 mm

Prüfschall: Terzrauschen/gleit. Meuliton
Empfangsfilter: Terzfilter/Oktavfilter

Literatur:
/1/ DIN 52 210: Messungen zur Bestimmung des Luft- und Trittschallschutzes
/2/ DIN 4109: Schallschutz im Hochbau
Zur Schalldämmung leichter zweischaliger Trennwände

KONSTRUKTIONSTYP

N = Anzahl der geprüften Wandkonstruktionen

KONSTRUKTIONSTYP:

N = Anzahl der geprüften Wandkonstruktionen
Sound Insulation of Stiff Lightweight Partitions

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1. Introduction

The demand for partition walls having very low surface weights and a maximum of sound insulation has led to many "single leaf" constructions which keep the limiting frequency

$$\omega_l = \frac{c}{2\sqrt{B}}$$

(1)

c = velocity of sound in air
m = mass per unit area
B = bending stiffness

as high as possible or, at least, high enough to ensure the validity of the mass law within the frequency range of practical interest.

Another way to get high sound insulation with relatively low surface weight is to keep the limiting frequency as low as possible, which means using lightweight structures with high bending stiffnesses. It has already been shown by CREMER [1] in 1942 that this is theoretically possible. He evaluated the increase of Sound Transmission Loss with frequency in the coincidence range to 9 dB/oct. compared with the 6 dB/oct. slope of the mass law. Thus it should be even possible to exceed the mass law from a certain frequency upwards.

The theoretical and practical implications of this statement especially with regard to sandwich constructions are discussed in this paper.

2. Transmission Loss of Homogeneous and Sandwich Plates

2.1 General calculations

A sandwich plate with very thin face layers as depicted on the next page is acted on by the sound pressures \( p_1 \) and \( p_2 \) due to an impinging and a radiated plane wave respectively. With the transverse velocity \( v \) equal for all layers the "insertion impedance"

$$T = \frac{p_1 - p_2}{v}$$

as function of angular frequency \( \omega \) and of the angle of incidence \( \gamma \) can be obtained
from the equation of motion, giving

\[ T = \frac{\omega^2 n m}{1 + \frac{\omega^2}{c^2} \sin^2 \gamma \left( \frac{B_1 + B_2 + B_3}{m} - \frac{c^2}{\sin^2 \gamma} \cdot \frac{B_2 + (h_2 - n) H_1 + n H_3}{S_2} \right)} \]

\[ , \quad (2) \]

with the bending stiffness components

\[ B_1 = \frac{E_1 h_1}{1 - \mu^2} \left( \frac{h_1^2}{12} + \frac{(h_2 - n + h_2)^2}{2} \right) , \]

\[ B_2 = \frac{E_2 h_2}{1 - \mu^2} \left[ \frac{h_2^2}{12} + \left( n - \frac{h_2}{2} \right)^2 \right] , \]

\[ B_3 = \frac{E_3 h_3}{1 - \mu^2} \left[ \frac{h_3^2}{12} + \left( n + \frac{h_2}{2} \right)^2 \right] , \]

the longitudinal stiffnesses

\[ H_1 = \frac{E_1 h_1}{1 - \mu^2} (h_2 - n) , \]

\[ H_2 = \frac{E_2 h_2}{1 - \mu^2} n , \]

the shear stiffness of the core

\[ S_2 = G_2 h_2 , \]

and the total surface mass of the plate

\[ m = \rho_1 h_1 + \rho_2 h_2 + \rho_3 h_3 \]

The position of the neutral plane \( n \) is given by

\[ n = \frac{h_2}{2} \left( 2 + \frac{h_2}{h_2} + \frac{E_2 h_2}{E_1 h_1} + \frac{E_3 h_3}{E_1 h_1} \right) \left( 1 + \frac{E_2 h_2}{E_1 h_1} + \frac{E_3 h_3}{E_1 h_1} \right)^{-1} \]

All rotational inertia terms have been neglected in eq. (2). The Sound Transmission Loss TL as an average value over all angles of incidence can now be calculated from the insertion impedance \[ [1] \]:

\[ TL = -10 \log \left[ \int_0^1 \left| \frac{T \cdot (1 - \sin \gamma^2)}{2 Z} \right|^2 d(\sin \gamma) \right] \]

\[ , \quad (3) \]

\( Z \) being the characteristic wave impedance in air.

### 2.2 Transmission Loss of homogeneous plates

Eq. (2) also applies to the case of a homogeneous plate when setting \( E_1 h_1 , \)

\( E_2 h_2 , \phi_1 h_1 \) and \( \phi_3 h_3 = 0 \) and multiplying the shear stiffness with the shear coefficient \( K = \pi^2/12 \) to allow for the nonuniform shear distribution across the thickness of the homogeneous plate \[ [2] \], so that

\[ S_2 = S = \pi^2/12 \cdot A \cdot h \]

The internal damping of the plate is accounted for by a complex modulus of elasticity

\[ E_2 = E(1 + j \delta) \]

Substituting eq. (2) into eq. (3) and integrating from \( \gamma = 0^\circ \) to \( \gamma = 85^\circ \) results in the curves of Fig. 1, which also shows the mass law line as obtained from eq. (3) with \( T = j \omega m \), as well as the results of measurements on a 2 cm thick model partition with \( \delta = 0.05 \). It can be seen from Fig. 1 that at high frequencies the slope of the TL-frequency curve again tends towards 6 dB/oct. due to the shear effect, but that the mass law can certainly be exceeded at frequencies above the limiting frequency, depending on the damping factor \( \delta \).

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2.3 Transmission Loss of a sandwich construction

A sandwich construction which combines a low limiting frequency and a high damping factor is known as a "spaced damping layer". The damping properties of such a plate as sketched in Fig. 2 have been described in [3]. Its Sound Transmission Loss can be calculated from eq. (3) and eq. (2) with

$$E_3 h_2 = 0$$

and

$$E_3 = E_3 (1 - j d_3)$$

Fig. 2 shows the calculated TL-frequency response for different values of $G_2$, the shear modulus of the core. These curves are again compared with the mass law and with experimental results from a 2 cm thick plate with $G_2 = 3.2 \cdot 10^9$ dyn. cm$^{-2}$.

Discrepancies between the calculations and the measurements at low frequencies are mainly due to boundary effects in the finite test panel (1 m$^2$) but also due to the frequency dependence of the damping factor $d_3$. Deviations from the calculated curve at high frequencies are caused by the neglect of the finite face layer thicknesses which would add a 6th-order term to the governing differential equation of eq. (2) [4].

In contrast to the homogeneous plate the TL-frequency curves of the sandwich plate tend to level off at high frequencies where the shear stiffness terms in eq. (2) become more and more important and where, in addition, the effective damping factor of the plate decreases since shear deformation in the undamped core layer becomes dominant.

However, if the shear modulus is high enough the mass law can be exceeded at much lower frequencies than in Fig. 1 for about the same damping factor.
3. "Transition Frequency" as Design Criterion

The "transition frequency" \( \omega_t \) where the TL-curve crosses the mass law line is obtained by setting

\[
TL = 20 \log \frac{\omega_m}{2Z} - 6 \, \text{dB} \quad (4)
\]

as given by the mass law, equal to the Transmission Loss in the coincidence range. If shear effects are neglected CREMER's equation can be used for the homogeneous plate [1]:

\[
TL = 10 \log \frac{\omega_m}{2Z} + 10 \log \left( \frac{\omega_s}{\omega_g} \right) - 3 \, \text{dB} \quad (5)
\]

yielding, together with eq. (4),

\[
\omega_t \approx \frac{\omega_g}{2d} \quad (6)
\]

By making use of eq. (2) with the relevant values for \( m \) and \( B \) the transition frequency becomes

\[
\omega_t = 1.7 \frac{c^2}{ph_c} \cdot \frac{c_1}{\sqrt{E/\rho}} \quad \text{(longitudinal wave velocity)} \quad (7)
\]

Eq. (7) only holds good for comparatively low frequencies:

\[
\omega \approx \frac{452^2}{mB} \approx 2 \cdot \frac{c_1}{n} \quad \text{as obtained from eq. (2).}
\]

A similarly straightforward relation for \( \omega_t \) of the sandwich plate of Fig. 2 can be derived if shear effects are also neglected, i.e. if in eq. (2)

\[
\omega^2 \cdot m \cdot [(h_x-n)h_x + n h_3] \ll 452^2 \quad (8)
\]

Then again eq. (5) describes the Transmission Loss in the coincidence range with

\[
\omega_s = \sqrt{\frac{m}{B_x + B_3}} \quad (9)
\]

For most practical sandwich constructions the position of the neutral plane is \( n \approx h_x \) and the damping factor \( d \approx d_3 \) [3]. The transition frequency then becomes

\[
\omega_t \approx 0.5 \frac{c^2}{d_3 h_x} \frac{c_1}{c_{13}} \sqrt{1 + \frac{h_x}{h_2} \cdot \frac{h_2}{h_3} \cdot \frac{h_3}{h_3}} \quad (9)
\]

confined to the frequency range

\[
\omega \ll 2G_2 \sqrt{\frac{1}{E_3 h_3 m}} \quad \text{as obtained from eq. (8).}
\]

In order to get high sound insulation at low frequencies \( \omega_t \) should be as low as possible. Thus, according to eq. (6), the plate should have a low limiting frequency, or high stiffness/weight ratio, and a high damping factor. Here sandwich constructions of Fig. 2 seem to be more suitable than homogeneous plates as can be seen by comparing eq. (7) with eq. (9) and Fig. 1 with Fig. 2. However, the sandwich plate requires an exceptionally high shear modulus of the core.

The author is very much indebted to Prof. Dr.-Ing. L. CREMER who initiated this investigation and who supervised the experimental work at the Institut für Technische Akustik of the Technische Universität Berlin.

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Composed Air-Born Sound Isolation of Partitions under Suspended Ceilings.

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When a partition of transmission loss $R_w$ separates two spaces with a suspended ceiling common to both of them, as indicated in the adjoined figure, the actually measured transmission loss of this partition, $R$, is seen to be always smaller than $R_w$. This is due to the peculiar flanking transmission via suspended ceilings from emitter to receiver rooms.

If we suppose that it is only worth while considering the propagation of sound in one sense: emitter-receiver via partition and suspended ceilings, we will derive a $\Delta R$ so that

$$R = R_w - \Delta R$$  (1)

Considering the plenum above the suspended ceiling as a two rooms configuration of common surface $S$ which transmission loss is null, the following set of equations can be written:

$$R = L_e - L_r + 10 \log S_w / A$$
$$R_c = L_e - L_1 + 10 \log S_1 / A_1$$
$$O = L_1 - L_2 + 10 \log S / A_2$$
$$R_c = L_2 - L_{rc} + 10 \log S_2 / A$$
$$R_w = L_e - L_{rw} + 10 \log S_w / A$$  (2)
Composed Sound Isolation of Partitions under Suspended Ceilings.

If we bear in mind that the sound pressure level in the receiver room, $L_r$, is built up by the contributions of the sound pressure levels $L_{rc}$ and $L_{rw}$ originating from the transmissions viae ceilings and wall (partition), we can write

$$L_r = L_{rc} + 10 \log \left( \frac{1}{\frac{L_{rc} - L_{rw}}{10}} + 1 \right) \quad (3)$$

Operating with the system of equations (1), (2) and (3) we finally obtain:

$$\Delta R = 10 \log \left[ 1 + \frac{R_w - 2R_c}{10} \right] \quad (4)$$

This formula can be considered as a generalization of the fruitful warning formula given in DIN 4109 for evaluating the diminution of transmission loss of a partition when inserting in it an element of lower transmission loss. Equation (4) is represented beneath.

![Diagram](image)

Evaluating points falling out of represented range can be read out by considering the translating character of the parameter curves.

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Composed Sound Isolation of Partitions under Suspended Ceilings.

$\Delta R$ is seen to be symmetrical with respect to the sense of the measurement. Influence of the different parameters on $\Delta R$ can be easily observed in equation (4) and in the family of curves. If the total absorption within the two rooms plenum $A_1$ and $A_2$ is almost built up by the rear absorption of the suspended ceiling, we can write the term $A_1 A_2 S_w / S_1 S_2 S$ in the simplified form: $(S_w / S) \cdot \Omega_S^2$ (5)

Experimental results in a model scaled 1:5 and comparison with predicted values of $R$. The model is a 2 cm thick plywood box with one coat of synthetic car paint in the inner surfaces (see figure of page1). The length of the left room was 81 cm and that of the right room 115 cm. The common width 96 cm and the common height 50 cm. The height of the plenum was varied between 3 cm and 1.5 cm. Results of figures a, b, c, e, and g are for interchanged emitter-receiver functions of left and right rooms. Half dodecaedron Isophon loudspeakers permanently left in both rooms radiated alternatively 1/3 octave filtered white noise. A 1/2 " B&K microphone was alternatively introduced in closely fitting Teflon rings at the corners. Reverberation times were measured by means of 1/4 recording/playback and proper filtering.

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Figure 4 represents the R values of the materials used as partition and suspended ceilings, measured in the transmission rooms facilities of the CIF for doors and windows. There is a very good agreement with Prof.'s Cremer theory for random incidence but with a slope of 5dB/Oct.

1) White draft paper, 244 g/m² (suspended ceiling in a, b, and c)
2) Baquelite plate 1.5 mm thick, 2230 g/m² (partition in a, b, c, e and g)
3) Polyester foam blanket 5 mm thick, 85 g/m² (suspended ceiling in e and g)

In figure f the Sabine absorption coefficients of draft paper and painted plywood are after Brebeck's measurements, although for rigid backing of the first one. The polyester foam $\alpha_s$ is a rough interpolation after Brebeck's measurements and equally valid for rigid backing. In our application one must expect higher $\alpha_s$'s in the lower frequency range. The $\alpha_s$ of SAP2 acoustic tiles is after CIF measurements. SAP2 was nailed to the plenum ceiling to increase $A_1$ and $A_2$ and the results are given in fig. c. In the results of fig. b, SAP2 covered plenum side walls only. The $\alpha_s$'s are calculated values with formula (4) in pretty good agreement with measurements, despite considering the plenum two reverberant rooms. Formula (4) can be a warning formula for architects.

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-E-140---
On the Sound Transmission Loss of Sandwich Panel

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Introduction

Sandwich panels have their section of the simplest composition after single and cavity panels, and are basically consist of three layers which are two surface layers and a inserted layer.

There are many kind of sandwich panels with regard to their materials, and from the acoustical point of view, at least three kind of sandwich panels have to be distinguished from each other by inserted layers, they are called in this report as rigid, elastic and resistive type.

The sound insulation mechanism of rigid type sandwich panel will be analyzed just like to a single panel, because its inserted layer has some extent of rigidity and whole panel counteract to sound wave as a single layer. Elastic type's will be analyzed as similar to a cavity panel, because its inserted layer has some extent of elasticity and its surface layers are able to oscilate respectively under the restriction of elastic force of inserted layer.

The inserted material of resistive type has, as sound absorbing porous materials, small rigidity and elasticity itself, but has some extent of resistance for travelling sound wave.

In this report, to compare sound insulation properties of three
kind of sandwich panel with each other, by the electrical simulation, formulæ to compute their sound transmission loss (TL) for normal incident sound wave will be derived.

Simulation of sandwich panels and derivation of formulæ to compute TL of them.

When the specific acoustic impedance of a panel for sound insulation is known as $Z$, TL of it for normal incident sound wave can be computed by the following equation:

$$ TL=10 \log_{10} \left(1+\frac{Z_{\text{inc}}}{Z_{\text{ref}}}ight)^{-1} \quad (1) $$

Where $Z_{\text{inc}}$ stands for the characteristic impedance of air and $\frac{Z_{\text{inc}}}{Z_{\text{ref}}}$ means complex reflection ratio of sound pressure for the sound incident side surface of the panel. And if the simulation for a panel solved analytically, the solution would be useful to practical design for sound insulative panels.

1) Rigid type sandwich panel

The simulation for a single panel to mechanical(M), acoustical(A), and electrical(E) circuit is well-known as in Fig. 1, and it will be able to apply for rigid type sandwich panel. The specific acoustic impedance for Fig 1 is

$$ z=j\omega m+\rho c \quad (2) $$

Where $m$ for M is the mass per unit area, $\rho c$ for R is the characteristic impedance for air and $\omega$ is angle frequency.

Then, substituting formula (2) into formula (1), well-known mass law is derived as follows:

$$ TL=10 \log_{10} \left[1+\left(\frac{\omega m}{2\rho c}\right)^2\right] \quad (3) $$

(But for practical use, mass law for random incidence have to be used and the influence of coincidence may not be neglected)

2) Elastic type sandwich panel

By the solution of A. London for cavity panel, when the angle of incidence of sound wave is $\theta$,

$$ TL=10 \log_{10} \left[1+\left(\frac{\omega m}{2\rho c}\right)^2 \left|\cos (kd \cos \theta) - \left(\frac{\omega m}{2\rho c}\right) \cos \theta \sin (kd \sin \theta)\right|^2\right] \quad (4) $$

where $k$ is wave number and $d$ is the thickness of air layer.
When \( \epsilon \to 0 \) and \( \epsilon \) is sufficiently small, formula (4) will be abbreviated as follows:

\[
TL = 10 \log_{10} \left[ 1 + \left( \frac{\omega M}{R} \right)^2 \right] + 10 \log_{10} \left[ 1 - \left( \frac{\omega R}{2\epsilon} \right)^2 \right] \quad (5)
\]

where \( \omega_c = \sqrt{\frac{2\epsilon R}{md}} \quad (6) \)

On the contrary, the simulation in Fig. 2 for elastic sandwich panel derives the similar result as formula (5).

For the electrical circuit in Fig. 2

\[
z = j\omega M + \frac{1}{j\omega M + R + j\omega C} \quad (7)
\]

And when \( M = M_0 = M \),

Substituting formula (7) to formula (1) following relation is obtained.

\[
TL = 10 \log_{10} \left[ \frac{1 + \omega^2(CR^2 + (1 - \omega^2CM)^2)}{4(1 - \omega^2CM)^2} \right] = 10 \log_{10} \left\{ \frac{1}{2} \right\} \quad (8)
\]

Then, when \( \omega CR^2 \ll 1 \) and \( \frac{CR}{M} \ll 1 \) formula (8) is simplified as follows,

\[
TL = 10 \log_{10} \left[ \frac{(1 - \omega^2CM)^2}{4(1 - \omega^2CM)^2} \right] = 10 \log_{10} \left[ \frac{1}{2} \right] \quad (9)
\]

Substituting \( m \) to \( M, \epsilon / \epsilon C \) to \( C \), formula (10) agree with formula (6).

For the case of elastic sandwich panel, \( d/E \) is used for \( C \) which means the reciprocal of spring constant when \( E \) is the young modulus of inserted material.

In formula (9), the second term show the influence of elastic layer, while the first term means mass law for the whole panel construction. (But for practical use, \( 10 \log_{10} \left[ \frac{(1 - \epsilon^2 + 4\epsilon^2C)^2}{1 + \epsilon^2C^2} \right] \) have to be used instead of the second term, when \( \epsilon = \frac{m}{\omega C} \) and \( \omega_c = \sqrt{\frac{2\epsilon E}{md}} \) )

3) Resistive type sandwich panel

A example of the simplest simulation for resistive type sandwich panel is shown in Fig. 3, though the specific impedance for it is
complicated as follows

\[ z = (R + r) - \omega (C + M + C_M) + \omega R (C + M + C_M + \omega C + M + M_r + j \omega (M + M_r) + C_R - \omega^2 M_M (C + C_M) - \omega^2 C + C_M + M_r) \frac{M + M_r}{M + M_r + C_R - \omega C + C_M + M_r} \]  

\[ = (R + r) - \omega^2 (C + M + C_M) - \omega R (C + M + C_M + \omega C + M + M_r + j \omega (M + M_r) + C_R - \omega^2 M_M (C + C_M) - \omega^2 C + C_M + M_r) \]  

Then, when \( M = M_r = M, C = C_M = C, r = r_K = \omega C R e \) and \( \frac{C_R}{M} \ll 1 \), substituting formula (11) to formula (1), following relation is obtained.

\[ TL = 10 \log_{10} \left( \frac{1 + \frac{1}{2} \left( 1 - \omega^2 CM \right)^2 + \frac{1}{2} \left( 1 - \omega^2 CM \right)^2 + \left( \frac{2 \omega M}{R} \right) \left( 1 - \omega^2 CM \right)^2 + \left( 1 - \omega^2 CM \right)^2 }{4 \left( 1 - \omega^2 CM \right)^2} \right) \]  

After some abbreviation and substituting \( m \) to \( M \), \( \omega \) to \( R \) and \( d/\rho \) to \( C \), it becomes to

\[ TL = 10 \log_{10} \left( \frac{1}{R} \left( \frac{\omega M}{R} \right)^2 + \left( \frac{1}{2} \right) \left( 1 - \omega^2 CM \right)^2 + \left( \frac{2 \omega M}{R} \right) \left( 1 - \omega^2 CM \right)^2 + \left( 1 - \omega^2 CM \right)^2 \right) \]  

\[ + 10 \log_{10} \left( \frac{1}{R} \left( \frac{\omega M}{R} \right)^2 + \left( \frac{1}{2} \right) \left( 1 - \omega^2 CM \right)^2 + \left( \frac{2 \omega M}{R} \right) \left( 1 - \omega^2 CM \right)^2 + \left( 1 - \omega^2 CM \right)^2 \right) \]  

In formula (13), the influence of inserted resistive material is shown only by the third term of it, because the other terms are nearly equal to formula (5) for cavity panel. (For practical use, the value of \( k \) in formula (13) would be obtained by the measurement of flow resistance for the inserted porous material.)

Note

The contents of this report had been reported to Transactions of the Architectural Institute of Japan No. 128 Oct. 1966 in Japanese.

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The calculation of sound insulation in a building

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Introduction
Most countries have regulations on minimum field transmission loss between rooms and dwellings in residential and other buildings.

Field transmission loss $R'$ generally differs from the transmission loss $R$, measured in laboratory on the partition. In the laboratory all indirect transmission paths are negligible, in the field they may sometimes transmit more sound than the direct path.

If all attenuations on the indirect paths, except energy reflections at the joints between walls and floors, are neglected, $R'$ can be calculated, regarding all important indirect transmission, on the basis of the constructional drawings. The calculation will in many cases give $R'$ with sufficient accuracy. In any case the calculation will point out weak points.

Transmission paths
The figures 1 and 2 show some transmission paths between two rooms with a partition wall with area $S_0$ and transmission loss (in laboratory) $R_0$. To the direct transmission $P_0$ through this wall come transmissions $P_n$ by indirect paths.

In the most simple case of indirect transmission this is settled by
1) $R_n$ of the wall (or floor) in the source room, forced to bending wave vibrations,
2) the reduction $J$ of the velocity amplitude at the joint between the wall in the source room and the emitting wall with free bending waves in the receiving room, and
3) the radiation reduction $E$ of the emitting wall with the area $S_n$.

In the case illustrated by figures 1 and 2 there are 12 indirect paths with one joint reduction. To these comes an infinite number of paths with two or more joint reductions. Paths with more than one joint reduction can, however, generally be neglected. One path with two
The calculation of sound insulation in a building.

Joint reductions is shown in figure 2.

Derivation of the formula for calculation of $R'$
The ISO R 140 definition of transmission loss is:
\[ R = L_1 - L_2 + 10 \log (S/A) \quad \text{dB} \quad (1) \]

$L_1$ = average pressure level in the source room
$L_2$ = average pressure level in the receiving room
$S$ = partition area
$A$ = absorption in the receiving room

For laboratory measurements $R$ is primarily defined as
\[ R = 10 \log \left( \frac{P_1}{P_2} \right) \quad \text{dB} \quad (2) \]

$P_1$ = incident power on the partition
$P_2$ = power transmitted through the partition to the receiving room

(2) can also be written:
\[ R = 10 \log \left( \frac{P_1}{P_t} \right) \quad \text{dB} \quad (3) \]

$P_1$ = average pressure in the source room
$P_t$ = """" at partition in the receiving room

We have

\[ \begin{align*}
P_1^2 &= I_1 \operatorname{Re}(Z_1) \\
P_2^2 &= I_2 \operatorname{Re}(Z_2) \\
P_t^2 &= I_t \operatorname{Re}(Z_2) \\
\end{align*} \]

\[ P_o = \frac{S_I}{S} = \frac{S_I}{S} \frac{\operatorname{Re}(Z_1)}{\operatorname{Re}(Z_2)} 10^{-\frac{R}{10}} \quad \text{W} \quad (4) \]

$P_o$ = power transmitted direct through the partition

\[ Z = \text{specific acoustic impedance in the room} \]
\[ P_n = S_n v_n^" \quad \text{Re}(Z_2) s_n \quad \text{W} \quad (5) \]

$P_n$ = power transmission by path $n$
$S_n$ = radiation area of path $n$ in the receiving room
$s_n$ = relative radiation defined by Gösele \[1\]
of the radiation area $S_n$
$v_n^"$ = average velocity amplitude of radiation area ($S_n$)

Incident power on the partition
\[ P_1 = I_1 \cdot S \quad \text{W} \quad \text{and} \quad (6) \]

Total transmitted power
\[ P_2 = P_o + \sum P_n = I_2 A \quad \text{W} \quad (7) \]

We introduce

\[ \text{radiation reduction} \]
\[ E = -10 \log s \quad \text{dB} \quad \text{and} \quad (8) \]

\[ \text{joint reduction} \]
\[ J = 20 \log \left( \frac{v_n^"}{v_n^"} \right) \quad \text{dB} \quad (9) \]

$v_n^"$ = average velocity of the wall (or floor) of path $n$ forced to vibrations in the source room
$v_n^"$ is determined by the transmission loss $R_n$ of the forced wall

\[ R_n = 10 \log \frac{\operatorname{Re}(Z_1)}{v_n^" \operatorname{Re}(Z_2)} \quad \text{dB} \quad (10) \]

With the relationships above and doing the assumption $Z_1 \approx Z_2$ we get:
The calculation of sound insulation in a building

\[
R' = -10 \log \frac{1}{10} R_p^{10} + \sum_n \frac{S_n}{10 R_n + J_n + E_n} \quad \text{dB} \tag{11}
\]

Summary

If we know the laboratory transmission loss of the different walls and floors in a building, their relative radiation and the velocity reduction at the joints between walls and floors, we can calculate the field transmission loss of any partition between two rooms. The method could easily be developed for calculating the field transmission loss between rooms not bordering each other.

The method can give sufficient accuracy for the practise if the following requirements are satisfied:

1) The boundary conditions for the walls and floors must not differ too much from those in the laboratory arrangement. This may for example be the case in certain element constructional systems.

2) The internal losses in the walls and floors must not be too high. (Damping factor \( \eta < 5\% \))

Joint reduction has to be determined in laboratory models or in existing buildings and relative radiation to be measured in laboratory as well as transmission loss. For many purposes, however, \( R, E \) and \( J \) can be calculated with sufficient accuracy by wellknown theories of Cremer, London, Gösele, Heckl, Watters and others. \([2 - 7]\)

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The calculation of sound insulation in a building.

**FIG. 1**

Sound transmission directly and indirectly by the source forcing the partition wall.

**FIG. 2**

Sound transmission indirectly by the source forcing side walls and floors.

Bibliography, cont.


Energy Density Formulas in Transmission-Loss Measurement

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Introduction

Several formulas\(^1\sim^4\) are used in laboratory measurement of transmission-loss, where the source room is reverberant and the receiving room is dead, or in field testing, where the condition is similar to the above rooms. In this paper, a generalization of these formulas is given for the direct component of energy density from the test panel, and a simple method is developed to obtain the transmission-loss.

Assumptions

The theory in this paper involves following two assumptions;

(A) The acoustic power \(W\) (watt/m\(^2\)) radiated from unit area of a surface source is uniform over the whole surface.

(B) The fractional acoustic power \(J(\Theta)\) radiated from unit area in unit solid angle at the specific angle \(\Theta\) with the normal has a directivity of the nth power of \(\cos\Theta\);

\[ J(\Theta) = J_0 \cos^n \Theta \] (watt/m\(^2\cdot sr\))

where, \(J_0\) is the acoustic power per unit area and solid angle in the normal direction to the surface.

The relation between \(W\) and \(J_0\) can be found by integrating \(J(\Theta)/R^2\) on the hemisphere with the radius \(R\) around the radiating surface.

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\[ W = \int_0^{\pi} \frac{J(\theta)}{R^2} 2\pi R (\sin \theta) R d\theta = \frac{2\pi J_0}{n+1} \quad \text{[watt/m]} \quad \text{(2)} \]

Fig. 1 shows the directivities of Eq. (1) as the power \( W \) is constant, and \( n=\infty \) corresponds with plane wave radiated from the source.

Fig. 1 Directivities of source

Fig. 2

Energy Density Formulas

The sound energy density \( D_P \) [watt·sec/m²] at a point \( P \) by the surface source is obtained by following integrations (Fig. 2):

\[ D_P = \frac{1}{c} \left( \int_A \frac{J(\theta)}{s^2} dA - \int_A \frac{J_0 \cos^2 \theta}{s^2} dA \right) = \frac{(n+1)W}{2\pi c} \int_A \frac{\cos \theta}{s^2} dA \quad \text{(3)} \]

where, \( c \) is the velocity of sound, and \( s \) is the distance from a surface element \( dA \) on the source \( A \) to the point \( P \).

For a circular source, whose radius is unit, an axial energy density is

\[ D_C(\ell) = \frac{(n+1)W}{c} \int_0^{\alpha} \cos^{n-1} \theta \sin \theta d\theta \quad \text{(4)} \]

where, \( \alpha = \tan^{-1}(1/\ell) \), and \( \ell \) is the axial distance of receiving point \( P \) (Fig. 3). Then, for \( n=0 \),

\[ D_C(\ell) = \frac{W}{c} \log_e \left| \frac{\ell^2 + 1}{\ell} \right| \quad \text{(5)} \]

for \( n > 0 \)

\[ D_C(\ell) = \frac{W(n+1)}{cn} \left\{ 1 - \left( \frac{\ell}{\sqrt{\ell^2 + 1}} \right)^n \right\} \quad \text{(6)} \]

for \( n=\infty \),

\[ D_C(\ell) = W / c \quad \text{(7)} \]

For a rectangular source, whose size is \( axb \), an axial energy density is

\[ D_R(\ell) = \frac{2(n+1)W}{\pi c} \int_0^{b/2} \int_0^{a/2} \frac{\ell^n}{(x^2 + y^2 + \ell^2)^{(n+1)/2}} \, dx \, dy \quad \text{(8)} \]

where, \( x \) and \( y \) are coordinates of the rectangular source (Fig. 4).

Then, for \( n=0 \),

\[ D_R(\ell) = \frac{2W}{\pi c} \int_0^{b/2} \frac{1}{\sqrt{y^2 + \ell^2}} \tan^{-1} \left( \frac{a}{2\sqrt{y^2 + \ell^2}} \right) dy \quad \text{(9)} \]
Energy Density Formulas in TL Measurement

for \( n = 1 \),
\[
D_R(\ell) = \frac{4W}{\pi c} \tan^{-1} \frac{ab}{2L/\sqrt{a^2 + b^2 + 4L^2}}
\]  
(10)

When one side length \( a \) is infinite, Eq. (8) becomes as follows.

For \( n = 0 \),
\[
D_S(\ell) = \frac{W}{c} \log_b \left[ \frac{b}{2L} + \sqrt{\frac{b^2}{4L^2} + 1} \right]
\]  
(11)

for \( n = 1 \),
\[
D_S(\ell) = \frac{4W}{\pi c} \tan^{-1} \frac{b}{2L}
\]  
(12)

Sound Pressure Level Formulas

The sound pressure levels along the central axes of these sources are given by

\[
SPL_2(\ell) = 10\log_{10} \frac{W}{120 + R(\ell)} \quad \text{[dB]}
\]  
(13)

where, \( R(\ell) = 10\log_{10} \left| D_p W/c \right| \)

and \( D_p \) is \( D_C(\ell), D_R(\ell) \)
or \( D_S(\ell) \) in the equations from Eq. (4) to
Eq. (12). Calculation results of \( R(\ell) \) are
presented for the circular sources in Fig. 3
and the rectangular sources in Fig. 4 respec-
tively. The axial distance in Fig. 4 is
normalized by the value \( ab/2(a+b) \).

It is interesting to see that the levels of \( R(\ell) \) for every \( n \)-
value (\( 0, n(\infty) \)) are within
Energy Density Formulas in TL Measurement

±1 decibel at the points, where the axial distance is equal to a half of radius for the circular source in Fig. 3 or normalized axial distance \( l \cdot 2(a+b)/ab \) is equal to unit for the several rectangular sources, in which, \( a/b = 1, 2 \) or \( 3 \), in Fig. 4.

Measurements of Axial Sound Pressure Levels

In a coupled-room which has reberberant and anechoic rooms, the measurements of the axial sound pressure levels were made for a 1.76x0.88(mxm) vinyl-cholide board (6mm thickness).

The results in Fig. 5 show the properties of a reduction curves from the actual surface source.

Conclusions

In transmission-loss measurement, the power \( W \) radiated from a test panel is equal to \( D_1 c \tau /4 \), where \( D_1 \) is average energy density in the source room, and \( \tau \) is transmission coefficient of the panel. Then, using Eq. (13),

\[
TL = SPL_1 - SPL_2(l) - 6 + R(l) \quad \text{[dB]} \quad \text{---------(14)}
\]

where, \( SPL_1 \) is the average sound pressure level in the source room.

When \( SPL_2(l) \) is measured at the point where the axial distance is equal to \( ab/2(a+b) \), the term \( R(l) \) may be neglected in Eq. (13) and (14), and then the transmission-loss is easily estimated without measurements of the sound pressure level distributions in the dead receiving room.

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Fig. 5 Axial SPL Reductions in anechoic room.
Model investigation on orthotropic partitions

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Introduction

The most common floor partition in multi-storey houses is the isotropic, homogeneous concrete partition. Isotropic means a plate where surface density and stiffness is unchanged and independent of now (the way) a square specimen is cut out. The acoustical properties of such constructions are rather well-known and can also be calculated with acceptable accuracy.

As a rule the dimensions of the partition give a low coincidence frequency. Within the frequency range 100-3150 Hz the transmission loss of the partition then can be raised by increasing mass and bending stiffness.

As a measure of sound insulation is used index $I_a$ in the following text (due to ISO Recommendation no 880, May 1965, "Rating of sound insulation for dwellings"). $I_a$ states the sound insulation quality for "normal" sounds.

Orthotropic floor partitions

From constructional and economical reasons - particularly by large spans - orthotropic plates are chosen, which can be given larger stiffness and sustaining capability than the isotropic plate with the same surface density. Common constructions are hollow plates, homogeneous plates with stiffening beams in one direction or in two orthogonal directions.

The knowledge of sound insulation by orthotropic plates is limited and methods are lacking for calculating or estimating the insulation by plates with various shapes.
Model investigation on orthotropic partitions,

Here, the results from preparatory studies of orthotropic plates are to be reported.

Model measurements

In the formulas for calculating the acoustical properties of a homogeneous plate a change in dimension (thickness) has the same influence as a corresponding change in frequency.

One condition is, however, that the other factors in the formulas (dynamic elasticity modulus, density and internal damping factor of the material) are not changed.

Thus, the same insulation index $I_a$ is obtained when measuring a concrete plate, thickness $h$, within the normalized frequency range 100-3150 Hz as when measuring a concrete plate, thickness 0.2 h in the range 500-16000 Hz.

In the present tests the scale 1:5 has been used and investigations were made on:
1. homogeneous plates with stiffening beams in one direction and two orthogonal directions resp. and also on isotropic plates as references. The plates were cast in gypsum.
2. hollow plates with cylindrical channels in different dimensions and an isotropic plate as a reference. The plates were made in concrete.

Measuring arrangements

The measurements were carried out in two model measuring chambers, volume 2.0 m$^3$ (1.5 x 1.3 x 1.0 m$^3$) and size of test openings 0.6 x 0.6 m$^2$.

When measuring, the plates under test has been suspended between the two test openings and the only contact with the chambers was formed by soft rubber sealings.

Influence from material constants etc

The coincidence frequency for the tested objects falls below 2000 Hz (corresponding to 400 Hz in full scale). Theoretically changes in material constants involves the following change in sound insulation index $I_a$:

- elasticity modulus $E$ : $I_a = I_{a1} - I_{a2} = 5 \log \frac{E_1}{E_2}$
- density $\rho$ : $I_a = 15 \log \frac{\rho_1}{\rho_2}$
- damping factor $\delta$ : $I_a = 10 \log \frac{\delta_1}{\delta_2}$
- thickness of plate $h$ : $I_a = 30 \log \frac{h_1}{h_2}$ (isotropic plates)
Model investigation on orthotropic partitions.

Due to these formulas it is possible to consider known changes in material data and plate thickness.

**Scaling the measuring results**

If divergences in material data between model and full scale plates are known the results from model tests can be scaled to full scale constructions according to the given formulas.

However, the diffusion of the transmitted signal is influencing the results. To avoid the uncertainty in diffusion when scaling, measurements have been made also on solid reference plates the results of which are compared to full scale tests in laboratory. The modified plates are then directly comparable to the reference plates.

**Results**

In the figure on page 4 sound insulation results from full scale laboratory tests are shown as a function of plate thickness $h$. These results are found to agree very well with the relation

$$I_a = 30 \log h + \text{constant}$$

The results from the model tests are given in a similar way as a function of thickness $h_{\text{equiv}}$ ($h_{\text{equiv}}$ = thickness of isotropical plate with the same surface density as the specimen).

The measured values for the reference plates in gypsum and concrete resp. connects as can be seen to the curve.

The remaining results show increased practical sound insulation for all the orthotropic plates as compared to the isotropic with the same surface density.

The results indicate that it is possible to attain floor partitions with high sound insulation even with a rather low total density if the stiffness is increased in one direction or two orthogonal directions. It should be possible to find the optimum conditions in this respect.
Model investigation on orthotropic partitions

![Graph showing sound attenuation with variable partition thicknesses.](image)

**Orthotropic Plate with Beams**

[@(a, b) Gypsum Model]

**Orthotropic Hollow Plate**

[@(a, b) Concrete Model]

**Isotropic Plate**

[@ Gypsum Model
@ Concrete Model
@ Concrete Full Scale]
Transmission loss of double door constructions.

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The problem of doors with a high T.L. is often raised in the field of acoustics. The applications of these doors are several. The most common problem is to reach sufficient secrecy between two rooms of which one for example can be a session-room of directors and the other one a passage or a secretary room. Another application is to keep a low sound-level in a room bordering on another and more noisy room.
The latter problem is the one - which for our part - raised the question of sound-proof doors.

As acoustic consultants to the new broadcast and television studios in Gothenburg, Sweden, we had to decide which door construction should be chosen, and the choice was between heavy specially constructed single doors, as commonly used in these vicinities - and two lightweight doors with distance and absorbent lining between.

Some of the most important problems concerning the building structure in studio buildings are:
1) The sound insulation
2) The installation of electric cables passing from one room to another.
3) The installation of ventilation ducts and tubing from one room unity to another.
As regards the sound insulation the mostly used structures are founded on the
Transmission loss of double door constructions.

Use of heavy wall- single or double arrangements. However we found that using this solution we would have great problems solving the installations of electricity and ventilation. We therefore preferred to use light double stud walls with a distance between the outer sheets of gypsum = 50 - 100 cm (1,5 - 3 feet). Doing this we gained.

1) A lightweight construction with decreased demands of the structure (load).
2) Cavities for all sorts of installations.

When this choice was made, the decision of door constructions was not difficult. Using double walls we had obtained a natural acoustic sluice between the inner and the outer wall, and we now had to find some acceptable door with a sound insulation as a principle close to or equal to that of the walls (double sheets of gypsum).

In respond to the economic demands of the project we started with the best standard-door in the Swedish market and in laboratory test we got a mean T.L. = 26 dB. (This door as to earlier lab-tests holds 34 dB. It is provided with a good rubber-sealing, and used mostly in dwellings between the staircase-wall and apartments). It is clearly to be read from the curve that the low T.L. is the consequence of leakages (we know from experience that most doors when mounted in buildings are badly fitted) and we therefore sealed the door with putty on both sides and got a mean T.L. = 33 dB. According to our calculations we regarded this T.L. of a single door as sufficient initial value and the work determining the influence of distance and absorbent lining between two such doors could begin.

In the test opening of the laboratory was built a 25 cm (10") brick-wall in which a 34 dB standard door was placed. At a distance from this wall another door was arranged and between these doors we built a passage, the walls and roof consisting of elements (two single 1/2" gypsum sheets with 2" between). All connections were tightened with mineral-wool, and flanking transmission was screened. The following test were planned:
A) Distance 15 cm (6") no absorption on walls in sluice
B) " 15 cm (6") absorption on walls in sluice
Transmission loss of double door constructions.

C) Distance 25 cm (10") absorption on walls in sluice

D) " 100 cm (40")  " -  " -  " -

E) " 200 cm (80") " -  " -  " -

The absorption consisted of 5 cm (2") thick mineral-wool bats (70 kg/m³ = 4.3 lbs/ft³) mounted on the walls.

The first two measurements (A and B) indicated some interesting results.

1) As the single-door measurements showed that the tightening was of great importance, it was not surprising that the difference between A tightened and A untightened was 4-5 dB and that the difference occurred within the high frequencies.

2) More important was that the absorbent lining (B untightened) practically gave the same increase as tightening (A tightened).

3) Though using absorbent lining - the tightening was of importance.

The difference between B tightened and untightened still was 4-5 dB.

Naturally the question was raised whether an absorbent lining - thick enough - could compensate the decrease of T.L. by untighteness.

Already the next measurements (C) showed a smaller difference between tightened and untightened doors.

The T.L. (average 100 - 3150 Hz) untightened was 49 dB. We regarded this as a remarkable result as the total construction depth is no more than 375 mm (15").

The next measurement (D) was carried out with a distance between the two doors = 100 cm (40") and we got a T.L. = 53 dB both for the tightened and the untightened doors.

Measurement E with a distance = 200 cm (80") gave a T.L. = 56 dB for the untightened doors and 57 dB for the tightened ones, a difference which probably can be ascribed to the accuracy of measurement.

With these results (all including doors with absorbent lining between)

B) Distance 15 cm T.L. = 45 dB

C) "  25 cm T.L. = 49 dB

D) "  100 cm T.L. = 53 dB

E) "  200 cm T.L. = 56 dB
Transmission loss of double door constructions.

we got an indication that the increase of insulation in relation to the distance between the doors will be greater with small distances, probably due to the fact that the absorbent linings have not fully compensated the sound leakage. With greater distances the increase is judged to be 2 dB for every doubling the distance.

To settle the distance where the full effect of absorption is reached we made one measurement with the distance 50 cm (20").

We got a T.L. = 51 dB and the curves of tightened an untightened doors were equal. The value 51 dB fits well in the serial between T.L. = 49 dB (distance = 25 cm) and T.L. 53 dB (distance = 100 cm).

At the same time we did establish that the limit where the absorption compensates the untightness lies somewhere between 25 cm and 50 cm, (10-20").

Finally we made a measurement with two doors with a slightly higher initial value (37 dB) at a distance of 50 cm (20") but this did not raise the T.L. (51 dB). The T.L. can probably be raised by using thicker mineral-wool and also by covering the inside area of the doors. However we regarded the attained values as sufficient for our propose, and the constructions are now carried out in the new Gothenburg Radio & Television center, which is to be completed at the end of 1969.
Spatial Properties of Reverberant Sound Fields

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Introduction

Earlier studies have been made of the statistical properties of the sound field set up in a reverberant enclosure by a sound source fixed in space and radiating a steady, one-frequency signal into an elastic fluid. It was shown theoretically, and confirmed experimentally by random sampling at well separated points in the field, that the spatial distribution of the mean squared pressure follows a gamma distribution.

The mean squared pressure is an important parameter as it is proportional to the energy density of the field. Such measurements in reverberant spaces may be made when it is desired to measure the power output of a transducer or source. In general the signal radiated by a source contains more than one frequency component, but this case is of practical interest and its analysis is helpful in the study of more complicated fields.

The sound field considered here is termed reverberant, by which is meant that at each measuring point there are many incoming wave trains, uniformly but randomly distributed in phase and direction of incidence. Such a field exists in an enclosure when its average absorption coefficient is less than 0.10 and its smallest dimension is at least 4 times the sound wavelength.

Thus the energy is not distributed uniformly throughout the space, and to
Spatial Properties of Reverberant Sound Fields

estimate the mean energy density of the field it is necessary to make several measurements at different points. Other work\(^2\) has indicated that it is advantageous to make each point at least \(l = \lambda/2\) distant from its neighboring points, so that the values measured at the points are all statistically independent.

The gamma distribution used here has a cumulative distribution function

\[
F(x) = M^m \left[ (m-1)! \right]^{-1} \int_0^x \frac{M^{-m}}{x^{m-1}} dx
\]

where \(X\) is an integer which determines the variance of the distribution, and equals the number of measuring points used.

We suppose that measurements of the mean squared pressure are to be made in the field, to determine the mean energy density and hence the power output of the source. An important question arising is: how accurate is the mean of a given number \(N\) of measured values? Here we answer this question, with particular reference to the case where only a limited region of space is available for the measurements, which imposes a limit on the number of statistically independent measuring points available. We consider three cases, where the points must lie a) along a straight line, b) in a circle on a plane, and c) in a spherical volume.

a. Line. Since each point must be at least the distance \(l\) from its neighbors, the best arrangement is to have them uniformly spaced along the straight line. Then the number of points \(N_1\) within a distance \(nl\) of a center point is

\[
N_1 = 1 + 2n
\]

where \(n\) is an integer.

b. Plane. Here the densest packing of points, subject to the constraint that no two points are closer than a distance \(l\), is obtained when the points lie at the vertices of equilateral triangles of length \(l\), arranged in a hexagonal lattice in 2 dimensions. Taking a measuring point at the origin, the number of points lying within a circle of radius \(nl\) is found, by counting, to be

\[
N_2 = 1 + 3(n^2 + n)
\]

where \(n\) is an integer. Eq. 3 shows that a circle of radius 3\(l\) contains 37 points.

c. Sphere. In this case the problem is closely related to that of the densest packing of rigid spheres. The densest packing of 4 points puts them at the vertices
Spatial Properties of Reverberant Sound Fields

of a tetrahedron of edge \( l \); with 4 spheres of diameter \( l \), their centers lie at the vertices of a similar tetrahedron. The sphere problem has a well known solution.\(^3\). As tetrahedral elements will not fill compactly a 3-dimensional space, the densest arrangement contains octahedra as well. Owing to symmetry there exist 2 different densest-packed arrangements, each having the same density, i.e. number of spheres per unit volume. They are called 'hexagonal' and 'face-centered cubic', and are well known to crystallographers.

The asymptotic expression for the number of points within a spherical volume of radius \( nl \) is

\[
N_3 = 4 \sqrt{2} \pi \left( \frac{n}{3} \right)^{3/2}, \quad n \gg 1
\]

Eqs. 2 and 3 are exact for integral values of \( n \), while Eq. 4 is approximate. Graphs of these equations are shown in Fig.1, where for simplicity the functions are plotted as smooth curves, for continuous values of \( n \). The true plots would consist of stepped lines, the value of the number of points \( N \) increasing discontinuously in steps of one as \( nl \) increased. The curves shown are useful for intercomparison, and in showing the trend of \( N \) in each case as \( nl \) increases.

Accuracy of Mean

Now that we know the number of statistically independent sampling points available within a distance \( nl \) from an origin in 1, 2 or 3 dimensions, we can calculate the probability that the mean of the measured values lies within a prescribed range of the true value, i.e. the true mean that would be reached in the limit if the number of sampling points was increased to infinity.

Such probabilities are calculable from the values of the gamma distribution for the appropriate values of \( N \). For example, Fig. 2 gives a graph of the percentage probability that the mean of the measured values lies within \( \pm 1 \) db of the true mean, versus \( n \), where \( nl \) is the radial distance of the boundary defining the space available for sampling, in 1, 2 and 3 dimensions.

Thus, for example, if a spherical region of radius \( 2l = \lambda \) can be sampled, Eq. 4 shows that approx. 47 points are available, and Fig. 2 shows that the corresponding probability of the mean of these 47 values lying within \( \pm 1 \) db of

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the true value is approx. 88%. In fact the asymptotic approximation of Eq. 4 is
rather conservative for such a small value as \( n = 2 \), and the true number of points
is 57, corresponding to a higher probability of approx. 91.5%. However, Fig. 2 is
useful in showing the trends of the functions.

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The sound energy density distribution in a room

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Introduction

The sound field in a room can be divided into two distinct parts, the direct sound field and the reverberant sound field. The intensity of the direct sound decreases with the square of the distance to the source and thus the level of the direct sound decreased with 6 dB for doubling the distance. The energy density of the reverberant sound is according to the theory, thought to be constant throughout the room.

That this simple representation is only an approximation and that the reverberant sound level decreases too with the distance to the source is well-known. The shape of the room, especially when the room is not a "simple" one and has to be considered as consisting of different parts who behave as coupled rooms are known to be responsible for spatial differences in sound energy density (1).

But even in "simple" rooms of rectangular shape the reverberant sound energy density proves not to be constant throughout the room.

The steady state sound energy density

Lübke and Gober (2) found that the sound level in large and low rooms decreases with the doubling of the distance to the source with an amount that is proportional the average sound absorption coefficient of the rooms.
Our results are in general in good agreement with the measurements of Lübke and Gober. We come, however, to a somewhat different interpretation.

As can be seen from fig. 1, the decrease of steady state sound energy density with the distance to the source is in essence an exponential one. The data reported in fig. 1 were measured in three long, almost bare corridors (piers of Schiphol airtterminal), (section 3.5 x 17 m²).

At 140 m from the source in the middle pier (M) some structures, who diffused the sound, caused a leveling of and farther on a steeper decrease of the sound level with the distance to the source.

This demonstrates the importance of diffusion. Receding from the source the sound becomes less and less diffuse, until at last only sound waves with a direction of propagation parallel to the ceiling are left. The sound, partly reflected by the structures, becomes then again more diffuse and the initially steeper decrease is partly restored.

The height of a room has a great influence on extent of the decrease with distance, if we have the same acoustical properties for ceiling and floor, the decrease with distance is proportional to the inverse of the height.

In large rooms even if the sound absorption in it was low and, therefore, the reverberation time long, we found that after a certain distance, wherein one could hold that the sound level decreased proportional to the doubling of the distance, the decrease became again exponential (fig. 2c).

In small rooms (e.g. to about 2000 m³) the decrease depends only on the reverberation time as shown in fig. 2a and 2b for rooms with a reverberation of 3.5 and 2.2 sec.

Out of these and similar figures we could compose fig. 3.

The decaying sound field

After the sound source has been stopped, the sound field decays in "simple" rooms in such a way that in general the differences level almost
The Sound Energy Density Distribution in a Room

out in a very short time, being in most cases of the order of magnitude of the time needed for sound waves to travel some times the mean free path.

There remain, however, even in a "simple" room, some differences in the mean sound energy density at different places. The density is in general higher in the middle of the room than near to the walls and especially near to sound absorbing surfaces.

Although these differences are in general small, they indicate that the Sabine reverberation time formula is to be preferred to the Eyring formula, as in the latter case the energy density of the decaying field at a certain moment has to be higher near to the walls and again especially near to the sound absorbing surfaces than at a greater distance to those surfaces e.g. in the middle of the room.


![Graph](image)

fig. 1. The sound level in dependance of the distance to the source in three long bare corridors (piers of Schiphol airterminal).
The Sound Energy Density Distribution in a Room

Fig. 2. The relative sound level \((\text{SPL-PWL} + 10 \log \frac{V}{24})\) in dependence on the relative distance (the distance divided by the critical distance) to the source.

Fig. 3. As Fig. 2. Summary of the results in "small" rooms.

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Mode Spacing Statistics in 200 m$^3$ Reverberation Chambers

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Introduction

In the design of a rectangular reverberation chamber, it is desirable to optimize the spacing between normal modes, even when a rotating vane is used as a diffusing element. Room dimension ratios that lead to suitable mode spacings have been studied by several authors$^{1,2,3}$ In this paper, some of the results obtained from computation of mode spacing statistics on a high-speed digital computer are presented. The frequencies of the normal modes are given by

$$f_{mn} = \frac{c}{2} \sqrt{\left(\frac{1}{x}\right)^2 + \left(\frac{m}{y}\right)^2 + \left(\frac{n}{z}\right)^2}$$  \hspace{1cm} (1)

where $c$ is the speed of sound, $x$, $y$, and $z$ are integers, and $x$, $y$, and $z$ are the room dimensions. The "ideal" mode spacing at a frequency $f$ is given by$^1$

$$\delta = 1 \left(\frac{4\pi V f^2}{c^3} + \frac{Sf}{2c} + \frac{L}{8c}\right)$$  \hspace{1cm} (2)

where $V$ and $S$ are the room volume and surface area, respectively. This "ideal" spacing and the actual spacing ($\delta$) between two adjacent normal mode frequencies in a given frequency band may be combined in a number of ways to determine how closely the room approaches an ideal room. Bolt$^2$ has defined the frequency spacing index as

$$\phi = \frac{1}{(f_b - f_a)} \sum_{a}^{b} \left(\frac{\delta^2}{\delta}\right)$$  \hspace{1cm} (3)

where $f_b$ and $f_a$ are the upper and lower frequency-band limits, respectively. Sepmeyer$^3$ has used the rms spacing index

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Mode Spacing Statistics in 200 m$^3$ Reverberation Chambers

\[ I = \left( \frac{1}{N} \sum_{n=0}^{N-1} \left( \frac{\delta}{\delta_n} \right)^{2} \right)^{1/2} \]  \hspace{1cm} (4)

where \( N \) is the number of modes in each band, and the ratio \( \delta/\delta_n \) is computed for each pair of modes. Sepmeyer has also considered an angular spacing index, but it will not be used in this paper.

The purpose of this paper is: (a) to verify Sepmeyer's results, (b) to evaluate \( I^2 \) in a frequency range studied by Bolt, and to compare \( I^2 \) with an evaluation of \( \Psi \) in the same range, and (c) to present new results for spacing statistics for a 200 m$^3$ reverberation chamber, in third-octave bands having ISO-recommended center frequencies.

Comparison with Previous Results

The results obtained by Sepmeyer are for normalized half-octave bands with room dimensions normalized with respect to length. Computations made in this study verified his results. Results obtained by Bolt have been recomputed. The contours obtained are qualitatively similar to those obtained by Bolt, but are much more irregular, and have "optimum" regions \( (\psi \approx 1) \) which were not given by Bolt. Sepmeyer has suggested that the index \( I^2 \) is more appropriate as a measure of spacing statistics. The index \( I^2 \) has been plotted in Figures 1 and 2 for a 200 m$^3$ room in the frequency range 30-90 Hz. This corresponds approximately to the normalized frequency range \( 0.5 \leq \psi \leq 1.5 \) used by Bolt. These contours are also much more irregular than those obtained by Bolt, and an "optimum" region, shown in Figure 2, was not shown on Bolt's original contours.

RMS Spacing Index for a 200 m$^3$ Room

To supplement Sepmeyer's results, computations have been made for the mean-spacing index (INDX AV)*, rms-spacing index (RMS), mean mode spacing (AV), rms mode spacing (RMS), and maximum mode spacing (MAX) for rooms having dimension ratios in the squares shown in Figure 3. (*The capital letter abbreviations in parentheses are used in Table I.) For convenience, several room dimension ratios suggested previously have been plotted in the figure. The 100 Hz third-octave band is generally most critical, and contours of equal-I for this band are shown in Figures 4 and 5. Optimum regions for this band are not identical to those that have been used previously; however, there are clearly "good" and "bad" regions in the plane. Additional contours for a constant value of \( \chi \) (as opposed to constant-volume contours) are similar, but not identical.
Mode Spacing Statistics in 200 m$^3$ Reverberation Chambers

Design of a Reverberation Chamber

To design a reverberation chamber, one must find suitable $p$ and $q$ values by comparing contours in various bands. In one 200 m$^3$ room that was designed, the height was limited, and the dimension ratios $p = 0.68$ and $q = 0.42$ were chosen. To aid in the detailed inspection of the mode structure for any given room, the mode spacings and spacing statistics for approximately 5000 rooms (including all of those in the rectangles of Figure 3) were tabulated using an increment of 0.0025 in $p$ and $q$. The room volume was kept constant ($V = 200$ m$^3$). An example of the results is shown in Table I. When construction of this room is complete, the mode structure will be studied experimentally and compared with the computations presented in this paper.

| Table I. Frequency spacing statistics, normal modes, and room dimensions for a 200 m$^3$ room having $p = 0.68$ and $q = 0.42$ |
|---|---|---|---|---|---|---|
| VOL= 200.00 | X= 8.88 | Y= 6.04 | Z= 3.73 | P= 0.680 | Q= 0.420 | SER= 273 |
| 4 MODES IN 50 HZ THIRD OCTAVE BAND |
| 46.12 | 48.08 | 50.02 | 54.20 |

| 8 IN BND | 63 | INDX AV= 0.949 | RMS= 1.192 | SP AV= 2.121 | RMS= 2.600 | MAX= 4.49 |
| 56.97 | 57.56 | 58.11 | 60.17 | 60.23 | 64.71 | 66.62 | 68.89 |

| 10 IN BND | 80 | INDX AV= 1.228 | RMS= 1.362 | SP AV= 1.934 | RMS= 2.182 | MAX= 4.40 |
| 73.29 | 74.18 | 75.81 | 77.48 | 79.46 | 81.37 | 82.55 | 82.90 | 85.45 | 87.62 |

| 23 IN BND | 100 | INDX AV= 0.955 | RMS= 1.172 | SP AV= 1.031 | RMS= 1.234 | MAX= 2.54 |
| 90.16 | 92.23 | 93.53 | 93.82 | 94.24 | 94.55 | 96.17 | 96.53 | 96.84 | 97.10 | 98.45 | 99.01 | 100.04 | 100.95 | 103.34 | 104.01 | 104.54 | 106.65 | 107.26 | 108.41 | 109.01 | 110.12 | 111.98 |

| 38 IN BND | 125 | INDX AV= 1.061 | RMS= 1.397 | SP AV= 0.776 | RMS= 1.032 | MAX= 3.44 |
| 112.36 | 112.67 | 113.16 | 113.93 | 115.12 | 115.34 | 115.57 | 116.21 | 119.65 | 120.34 | 120.45 | 121.45 | 122.91 | 123.00 | 123.78 | 124.22 | 124.43 | 125.03 | 125.73 | 127.22 | 127.90 | 128.23 | 128.87 | 129.15 | 129.42 | 131.57 | 133.25 | 133.74 | 135.58 | 135.96 | 136.74 | 137.14 | 137.40 | 137.78 | 138.35 | 138.51 | 138.54 | 139.70 |

Conclusions

As part of a theoretical study of the normal-mode spacing statistics in a reverberation chamber, the rms-spacing indices computed by Sepmeyer have been verified. At low frequencies (30-90 Hz), contours of the mean-square spacing index are qualitatively similar to those obtained by Bolt, but are much more irregular. The mode spacing statistics in the 100 Hz third-octave band are most critical for a 200 m$^3$ chamber room; however, all bands should be considered in the design of a room. Spacing statistics for approximately 5000 rooms have been tabulated, and a 200 m$^3$ reverberation chamber having dimension ratios $p = 0.68$, $q = 0.42$ has been designed.

—E-171—
Mode Spacing Statistics in 200 m$^3$ Reverberation Chambers

References


Measurements on a Single Helium Filled Balloon

L. P. Delssaso and Vern O. Knudsen

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University of California, Los Angeles

Introduction

This paper and the following one are a progress report of the continued investigation of room acoustics we began five years ago in the Acoustical Laboratory of the then new Physics Building of the University of California, Los Angeles. The two companion papers deal with the diffusion of sound by large (meteorological) balloons. The first paper reports our findings in an experimental investigation, in the anechoic room, of the reflection, transmission, refraction and diffraction of sound by a single helium filled balloon having a diameter of about 40 inches. The second paper reports the results of our investigation of the diffusion of sound in the reverberation room by 20 of the large helium filled balloons.

Many schemes and devices have been proposed and utilized during the past half century for the purpose of obtaining diffuse sound fields in rooms, especially reverberation rooms. A completely diffuse field is a prime requirement underlying (1) the application of the formulas of geometrical acoustics for the determination of sound absorption coefficients of acoustical materials, and (2) the use of these coefficients for calculating the reverberation times of all other rooms. Among the schemes and devices that have been used in reverberation rooms are skewed (non-rectangular) walls; convex walls and ceiling splays; polycylindrical columns; spherical, cylindrical, and other protrusions of the walls; suspended diffusers;
such as cubes, spheres, or convex surfaces, formed by bending sheets of plastic material or metal into cylindrical surfaces, having different dimensions; and rotating panels or vanes of various shapes and sizes. For our old (1929) reverberation room at UCLA, after tests of several of the above mentioned diffusers, we installed a motor driven paddle, about 12' x 12' in the rectangular room (19' x 20' x 16'). The paddle, which was rotated about 30 rpm, was made up of nine smaller plane vanes oriented at random angles. We have not yet introduced permanent diffusers in the new (1963) reverberation room, (a) primarily because we have been using the room to simulate the many existing rectangular rooms in which one "wall" - the floor or the ceiling is absorptive, and the other five "walls" are reflective, but (b) because we wanted to investigate the relative merits of various schemes for attaining the required conditions of diffusion before making any permanent installation of diffusers. Our present objective is to determine the merits of helium balloons compared with other types of diffusers.

Measurements on a Single Helium Filled Balloon

We report in this paper our measurements made in an anechoic chamber on a single helium balloon. Both transmission and reflection measurements were made. A spherical source consisting of a loudspeaker, to which a 40 in long, 1/2 in diameter tube had been attached, was placed 10 ft from the course of a microphone. This unit was arranged to move with constant speed along a straight line for a distance of 22 ft. The RMS sound pressure was recorded by a B & K level recorder. The frequency of the driving oscillator was maintained to ±0.1%. For reference purposes the sound pressure was recorded as the microphone traveled the 22 ft past the source in the empty room. The record thus obtained agrees well with the calculated values assuming spherical spreading of the sound from the small source.

The helium balloon under investigation was next placed a short distance beyond the microphone as shown in Fig. 1. In this position the sound reaches the microphone by a direct path from the source and by a reflected (scattered) path from the surface of the balloon. As the microphone moves along its course the direct and reflected components will change phase and amplitude, producing the interference
Measurements on a Single Helium Filled Balloon

pattern (See Fig. 1). The two records made it possible to calculate the reflected sound for positions of the microphone along its course. For example, at the center of the course the lower curve shows a maximum value. This indicated that the direct and reflected components are in phase. The path length of the reflection from the microphone was 1.42 ft. The wavelength for the 4000 Hz record was 0.282 ft. The echo length was 5λ. The 3 dB increase at this point between the reference and lower curve shows that the reflection coefficient is 0.172. The calculated reflection for an air-helium surface at normal incidence is 0.182.

For the transmission measurements the helium balloon was placed as shown in Fig. 2, about midway between the source and the microphone. The record obtained in this case is more complex than the reflection one. At each end of the course the microphone receives (1) the direct wave from the source, (2) a reflected component from the outer surface of the balloon, and (3) a refracted component through the balloon resulting from the difference in velocities in air and helium.

As the microphone approaches the "shadow" of the balloon, the component of the sound which travels through the air is reduced and, in combination with that portion which travels through the helium, produces the interference pattern shown in the lower curve of Fig. 2. Note that the general effect of the balloon has been to reduce the sound pressure substantially while the microphone is in its "shadow", and to increase the sound pressure a small amount when the microphone is at the ends of its run. These gross effects are accounted for largely by Snell's law of refraction. The interference effects are more complex at higher frequencies, and are very sensitive to the orientation and placement of the balloon.

For the purpose of displaying these effects, a motion picture has been made of pulsed sound at 4000 Hz for a significant part of the microphone motion. This film will be shown at the meeting.

The following paper will describe some diffusive effects of the helium balloons in a reverberation room.
Measurements on a Single Helium Filled Balloon

FIGURE 1
SOUND REFLECTED FROM A HELIUM BALLOON

FIGURE 2
SOUND TRANSMITTED THROUGH A HELIUM BALLOON
Diffusion of Sound by Helium Filled Balloons

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Introduction

The preceding paper demonstrated that helium filled balloons are effective diffusers of sound because they scatter incident sound by reflection and also transmitted sound by refraction. The task of computing the diffusive action of as many as 20 randomly located balloons in our reverberation room would be formidable, even for a theoretical physicist with a modern computer. We therefore report the following experimental findings:

The effects of 4 air filled and of 4 helium filled balloons (about 40-inch diameter) on the rates of decay of pistol shots: Decay curves were obtained with a B & K level recorder, with a 1/3 octave band pass filter, having mid frequencies at 125, 250, 500, 1000, 2000 and 4000 Hz. The effects of the air filled balloons were negligible; the helium filled ones increased the decay rates, improved the linearity of decay, and reduced fluctuations during decay. Decay curves also were obtained for 10 and 14 balloons. Subsequently, detailed work was done with 20 of these helium balloons. All data obtained throughout the investigation have been corrected for the effects of humidity and temperature, and of the rate of collapse of the balloons.

Effects of 20 Helium Filled Balloons: The method chosen to exhibit the diffusive properties of the balloons, because of its practical value, consisted of an
Diffusion of Sound by Helium Filled Balloons

Examination of their effects on the decay curves when different sized rectangles of 2" Fiberglas (1.5 lbs/cu.ft) were placed on the floor of the room, for different elevations of the microphone that responds to the reverberant sound (see Fig. 1).

If the sound field were completely diffuse, according to geometrical acoustics, the decay curves would be linear, and the added absorption of the Fiberglas would be proportional to its area, independent of its shape or position, and of the positions of the source and receiver. Even for a diffuse sound field, the edge effect is obviously less when the absorptive specimen is in a corner of the room. We investigated the effect of moving the 8' x 9' specimen of Fiberglas different distances from one corner of the room. Although the effect was not large, it was in quantitative agreement with effects reported by Esche\(^1\) and by Meyer and Kuttruff.\(^2\)

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\(^1\) Esche
\(^2\) Meyer and Kuttruff

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1. Esche
2. Meyer and Kuttruff
We obtained decay curves for the six 1/3 octave bands for each of the five specimens of Fiberglas, with and without the 40-inch balloons, for each of the five microphone elevations. With no Fiberglas on the floor, the 20 balloons increased the decay rates appreciably at low frequencies. At the 125 Hz band, the initial decay rate was 6.1 dB/sec with no balloons, and 9.4 dB/sec with the 20 balloons. The balloons also diminished the fluctuations in the decay curves. The calculated "surface absorption coefficients" of the balloons diminished progressively from .05 at 125 Hz to .01 at 2000 Hz (see Fig. 2).

The undashed curves of Fig. 2 give the absorption coefficients (calculated by the standard reverberation room method) of the five specimens of Fiberglas, with the 20 balloons in the room. The decay rates used for calculating the absorption coefficients were the average values of the initial decay rates measured at the five microphone elevations. There is a progressive diminution in the coefficients with increase of surface up to 261 sq ft. For the three larger areas there is relatively good agreement for the coefficients as a function of frequency. It is reasonable to believe that the values obtained for these three large surfaces are typical of the absorption coefficients that would be obtained in rectangular rooms having good diffusion with 50-100% of the floor or ceiling treated with a similar absorptive material. The dashed lines in Fig. 2 give the calculated coefficients for three typical areas, with no balloons in the room.

In a previous investigation\(^3\) we found that when the floor was completely covered with a 2" blanket of mineral wool (1.4 lbs cu/ft), with no diffusion in the room and with the pistol in the upper part of the room, the decay rate of the 250 Hz band was as slow as 6.2 dB/sec, whereas in the lower part it was as high as 39.3 dB/sec. In contrast, with the entire floor covered with the 2" Fiberglas, but with the 20 balloons in the room, and with the pistol shot in the lower part of the room, the decay rate at the 250 Hz band was 39.2 dB/sec in the upper part of the room and 37.3 dB/sec in the lower part.

The data in the following table are selected from a total of 420 decay curves to show how the initial decay rates (dB/sec) depended upon the elevation of the
Diffusion of Sound by Helium Filled Balloons

microphone, with 20 or no balloons in the room, for three different areas of Fiber-glas on the floor. The average of the decay rates at 2\' and 6\' above the floor are listed to characterize the reverberation in the low part of the room; the average at 18\' and 22\' in the high part. The numbers in parentheses give the dB range for which the average initial decay was approximately linear. The approximate effective reverberation times are 60 divided by the listed decay rates.

<table>
<thead>
<tr>
<th>No. of Balloons</th>
<th>Microphone</th>
<th>Decay rates(dB/sec) for 560 sq ft</th>
<th>Decay rates(dB/sec) for 174 sq ft</th>
<th>Decay rates(dB/sec) for 72 sq ft</th>
</tr>
</thead>
<tbody>
<tr>
<td>20</td>
<td>High</td>
<td>250Hz 1000Hz 4000Hz</td>
<td>250Hz 1000Hz 4000Hz</td>
<td>250Hz 1000Hz 4000Hz</td>
</tr>
<tr>
<td>20</td>
<td>Low</td>
<td>37(30) 55(33) 72(38)</td>
<td>23(27) 24(23) 34(49)</td>
<td>18(26) 15(38) 25(41)</td>
</tr>
<tr>
<td>0</td>
<td>High</td>
<td>23(10) 20(18) 36(18)</td>
<td>17(22) 16(28) 31(31)</td>
<td>13(29) 13(39) 26(49)</td>
</tr>
<tr>
<td>0</td>
<td>Low</td>
<td>36(13) 21(13) 38(20)</td>
<td>19(23) 17(30) 31(36)</td>
<td>14(22) 13(37) 26(45)</td>
</tr>
</tbody>
</table>

From the examination of the entire series of decay curves with the 20 balloons in the room, it was apparent that they furnished sufficient diffusion to justify the use of standard reverberation time formulas for determining practical values of coefficients of sound absorption of most acoustical materials, for frequencies above 125 Hz, provided the reverberation room is about as large as 19\' x 30\' x 24\', and the specimen covers about one-half of the floor. We are currently investigating other diffusive agents; the results will reveal, we believe, whether the diffusive effects of helium filled balloons have more than academic interest.

Footnotes
1. V. Esche, Experimentelle Untersuchungen zum Freiwerden einer Absorberkannte beim Kanteneffekt, Hochfrequenztechnik und Elektroakustik, 76, 4, 133-141 (1967); describes how absorption of absorptive material depends on its separation from walls and on its division into strips, as a function of wavelength.
Several physical conditions at measurements
in reverberation rooms

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ISO Recommendation [1] requires an omnidirectional sound incidence for
sound absorption coefficient measurements in reverberation rooms. It
is recommended to use diffusers hung over the room at different an-
gles, so that their projection on walls would cover equal percentage
of their surfaces. The measured sample has to cover a single area of
10-12m². In order to verify the correctness of these regulations, the
angular collision frequency characteristic has been introduced by
means of mathematical modelling. Its significance was studied for two
reverberation rooms without diffusers. The influence of sample size,
its position, sound source position and room shape is demonstrated.

Mathematical modelling method.
The principle of this method lies in dividing the sound energy flow,
radiated from the source into n elementary flows and in following the
elementary flows' intensity up to 60-dB-decrease. This enables us to
choose an arbitrary area on the room surface and to register there
the incident sound energy. Registering simultaneously both the inten-
sity and time of incidence, we are able to establish the local rever-
beration course and time. Following collision frequency and angles of
incidence of the energy flows, we may assess the angular collision
frequency characteristic of the incident sound energy.
Angular collision frequency $\mu$, was defined as a function of angle of incidence $\Psi$;

$$\mu(\Psi) = \frac{1}{S} \int m(\Psi) \cdot dS$$  \hspace{1cm} (1)

where

$$m(\Psi) = \frac{N(\Psi)}{\bar{N}}$$  \hspace{1cm} (2)

$N(\Psi)$ is number of collisions from an interval $\Psi < \Psi < \Psi + d\Psi$ and $\bar{N}$ represents the mean number of collisions over the hemisphere $\Sigma$ (Fig.1);

$$\bar{N} = \frac{1}{S} \int N(\Psi) \cdot d\Sigma$$  \hspace{1cm} (3)

Thus the angular collision frequency (1) represents for any $\Psi < 0, \pi/2$ an average value of $m(\Psi)$, taken over the registration area $\Sigma$ which usually was the surface of the sample. To the introduction of angular collision frequency led the well known fact, that in most materials

$$\alpha = \alpha(\Psi) \quad \text{but} \quad \alpha \neq \alpha(\phi)$$  \hspace{1cm} (4)

Due to measuring method used, the value of $\alpha$, obtained from measurements in a reverberation room is a function of $\mu(\Psi)$. This represents the influence of sound energy propagation mechanism in enclosures:

$$\alpha = \frac{1}{\mu} \int \alpha(\Psi) \cdot d\mu$$  \hspace{1cm} (5)

The proofs can be seen in author's works [2, 3, 4].
Several physical conditions at measurements

Fig. 3.

Conditions at measurements in reverb. rooms.

By means of mathematical modelling, two reverberation rooms were examined: DK1 as an example corresponding to ISO Recommendation and BBC reverberation room described in literature, differing from ISO Recommendation in small volume and inconvenient linear dimensions.

Fig. 2 shows the influence of sound source position. A sample of surface area 1m² was placed in DK1, sound source in positions Q_a (left upper corner), Q_b (3m above floor center). The angular collision freq. characteristics a and b show a remarkable difference caused by change in sound source position. Compared, the semicircle $\mu_{\text{dif}}$ represents the angular collision freq. characteristic of a true diffuse field.

Fig. 3 shows the influence of sample position. As in Fig. 2, the same sample was placed in DK1, positions a, b, c. Sound source was in position Q. Characteristics a, b, c show clearly that moving the sample by 3m from a to b or merely by 1.4m from a to c causes significant changes in physical conditions - comparison with conditions $\mu_{\text{dif}}$ available.

Fig. 4 shows the influence of sample size and room shape. The changes in
Several physical conditions at measurements

characteristics, both due to different sound source positions (Fig. 4a, b) and room shape (Fig. 4c) were reduced to minimum when sample in size 12m² was used.

Conclusions.
Physical conditions at α-measurements in reverberation rooms were investigated by introducing the angular collision frequency characteristic, which reflects the distribution of angles under which sound energy reaches the sample. These physical conditions have been found the more influenced by sample position, sound source position and room shape, the smaller the sample was; when the recommended size 12m² had been used, the changes proved to be minimal. In any case, the characteristics differ substantially from diffuse field conditions. A correction can be reached by using diffuse, but they may not be arranged randomly. A further work of our Institute will concern this problem.

CONTRIBUTION A L'ETUDE DU COEFFICIENT D'ABSORPTION ACOUSTIQUE DES MATÉRIAUX
PAR LE PROCÉDE DE LA SALLE RÉVERBERANTE

Monsieur VAU, Ingénieur à la Compagnie de Saint-Gobain
Monsieur LEHRMANN, Professeur à la Faculté des Sciences et Directeur de
l'Institut Universitaire de Technologie du HAB.

Introduction

Lors d'une communication présentée au 5ème Congrès International de LISBON (1), nous avons décrit un appareil destiné à mesurer automatiquement les temps de réverbération (réverbéromètre automatique). Il était possible, avec cet appareil, de mesurer avec précision les temps de réverbération propres à un espace clos réfléchissant, à condition d'effectuer un nombre suffisant de mesures. Grâce à son utilisation, nous avons pu mettre en évidence certains aspects importants, notamment le caractère gaussien de la distribution des temps de réverbération effectués en bruit blanc.

Nous avons ensuite, depuis cette époque, divers travaux complémentaires dont nous nous proposons d'en préciser les principaux aspects.

Étude approfondie de la répartition sonore dans un espace clos réfléchissant

Une salle réverbérante n'est pas un élément passif (2). Elle possède notamment des modes de résonance propre qui font que, lorsque l'un d'entre eux coïncide exactement avec la fréquence d'excitation de la source, la quasi-totalité de l'énergie présente dans la salle se retrouve uniquement sur cette fréquence.

La salle entre en résonance (tout comme un tuyau) et on voit apparaître un régime stationnaire, avec surfaces nodales et ventrales. L'admittance des parois de cette salle a une part prépondérante sur la nature du réseau stationnaire qui prend appui sur elles (3 et 4).
À l'aide d'un appareillage approprié, on peut déterminer la présence de ces modes propres. Sur les courbes de réponse de la salle effectuées en fréquence pure, il est possible d'observer une succession de "crêtes" et "creux" susceptibles de représenter des maxima et des minima de pression. C'est ce que "voit" tout du moins le microphone placé à tel ou tel endroit de l'espace et qui capte ainsi les particularités des réseaux d'ondes stationnaires qui s'établissent dans son voisinage (5).

Quand on règle la fréquence de mesure sur un de ces maxima ou minima et que l'on coupe l'excitation sonore, l'énergie localisée à cet endroit de la salle disparaît progressivement, ce qui ne fait que traduire le phénomène de réverbération de la salle à la fréquence considérée. La seule particularité, cependant, tient dans le fait que tous les enregistrements graphiques de temps de réverbération effectués en un point où existe un maximum se ressemblent. Même observation, d'ailleurs, pour ceux qui sont effectués en un point où existe un minimum.

Toutefois, sur les enregistrements effectués sur la plupart des minima, on observe, immédiatement après la coupure du son, une brusque remontée du niveau sonore (5 à 15 dB) avant la décroissance de celui-ci. Aux points où existent ces minima de pression, les ondes sonores sont plus ou moins en opposition de phase et, au moment de la coupure du son, l'onde réfléchie apparaît et sa présence se manifeste alors brusquement par cette remontée du niveau sonore.

Si on mesure les temps de réverbération (obtenus aux fréquences pour lesquelles on observe des maxima ou des minima de pression) à l'aide du réverbéromètre automatique, on constate que la suite des valeurs des temps, pour une fréquence et une position de microphone données, sont identiques : aucun traitement statistique ne peut être appliqué. Une seule valeur (la première ou la même par exemple) suffit alors pour déterminer exactement la valeur du temps de réverbération. On peut donc en conclure que la nature gaussienne classique des résultats obtenus en utilisant un bruit blanc semble, à priori, être due à la nature aléatoire propre de ce bruit.

Mesures comparatives des coefficients d'absorption obtenus dans six salles réverbérantes avec le même appareillage et le même matériel.
Dans nos essais, nous avons utilisé un bruit blanc filtré par tiers d'octave. Cependant
l'existence des modes propres de la salle de mesure se fait également sentir avec le bruit blanc filtré. Les courbes de décroissance obtenues avec ce type de source sont alors très semblables aux décroissances sonores les plus longues observées sur les fréquences propres situées dans le tiers d'octave utilisé. Ce résultat est surtout valable pour les fréquences basses (jusqu'à 300 Hz environ). On peut ainsi mieux comprendre les dispersions observées sur les valeurs des temps de réverbération obtenus en basse fréquence, à cause de la présence de réseaux stationnaires à larges mailles (4). En de nombreux points de l'espace, il est possible d'obtenir des valeurs de temps avec une grande précision pour chacun de ces points, mais les loisres varient d'un point à l'autre. Au-dessus de 200Hz les modes propres deviennent de plus en plus nombreux, les mailles des réseaux se resserrent : la dispersion des mesures diminue nettement. Dans les six salles réverbérantes étudiées (de volumes 50, 53, 106, 126, 169 et 210 m$^3$), nous avons d'abord procédé à l'étude des temps de réverbération ($T_o$) salles vides afin de déterminer le coefficient d'absorption $\alpha_o$ des parois. Puis, nous avons procédé à l'étude des temps de réverbération ($T_m$) de ces mêmes salles garnies d'un même matériau (fibre de verre) dont la surface ($S_m$) varie de 1 m$^2$ jusqu'à 12 m$^2$. Toutes les mesures étaient effectuées à l'aide du réverbéromètre automatique et les résultats ont été transcrits, au même moment, sur un perforateur de bande. Ces bandes perforées ont été ensuite soumises à différents tests statistiques à l'aide d'une machine IBM 360 (6). En cherchant les liens possibles qui pourraient exister entre les variations de toutes les valeurs obtenues, on s'aperçoit que tous les paramètres en jeu : fréquence, volume, surface des salles, des matériaux, sont parfaitement indépendants. Ainsi, nous avons constaté que deux séries de temps $T_o$ ayant une moyenne identique, dans deux salles différentes, n'ont aucune corrélation entre elles. Cette premiè re constatation semble signifier qu'une salle donnée ne peut fournir à priori des résultats plus signifi-
On peut tracer $T = f(S_m)$ pour toute autre fréquence. Sur quelques échelles nous avons vérifié qu'elles appartenaient bien à une famille d'hyperboliques en testant des approximations successives à l'aide d'un polynôme de la forme: $y = ax + bx^2 + \frac{c}{x^2}$. Les valeurs des constantes $a$, $b$, $c$ et $d$ changeant pour chaque fréquence et pour chaque salle, mais la forme générale de l'équation reste toujours la même.

D'autre part, nous avons constaté qu'il existait dans des salles ayant même $\alpha_0$, un accord satisfaisant entre les coefficients $\alpha_m$ à condition de disposer dans ces salles une quantité adéquate de matière. Cette quantité peut disparaître complètement en éliminant des lois de similitude acoustique (3). Ainsi entre deux salles, de même $\alpha_0$, ($10^6$ m$^{-2}$ et $150$ m$^{-2}$) nous obtenons les $\alpha_m$ suivantes:

<table>
<thead>
<tr>
<th>Fréquence (Hz)</th>
<th>200</th>
<th>2000</th>
<th>5000</th>
<th>10 000</th>
<th>15 000</th>
<th>30 000</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\alpha_m$ = 106 m$^{-2}$</td>
<td>0.05</td>
<td>0.05</td>
<td>0.01</td>
<td>0.07</td>
<td>0.07</td>
<td>0.09</td>
</tr>
<tr>
<td>$\alpha_m$ = 106 m$^{-2}$</td>
<td>0.31</td>
<td>0.63</td>
<td>0.92</td>
<td>1.12</td>
<td>1.06</td>
<td></td>
</tr>
</tbody>
</table>

Conclusions

Il serait intéressant de mettre également en évidence les rapports $\frac{A}{B} = \frac{\beta_{m=n}}{\beta_{m=0}}$ pour chaque bande de fréquence, avec une surface de matière donnée. On peut ainsi arriver à traduire très exactement cette qualité propre à un même matériau de réduire dans les mêmes proportions le temps initial $T_0$ de différentes salles de même $\alpha_0$.

Une étude plus approfondie de ce rapport devrait nous permettre de souligner l'importance pratique de ce rapport.

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Investigations into the Precision of Measurement of Sound Absorption Coefficients in a Reverberation Room

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(Secretary of the Committee for the Method of Measurements of Sound Absorption Coefficients in a Reverberation Room)

Precision measurements of the reverberant sound absorption coefficient have been drawing everlasting attentions in the history of architectural acoustics. In Europe and America, the round robin tests were carried out repeatedly to standardize the method of measurements. Results of these investigations were collected in the ISO Recommendation or ASTM Standards. However, it seems that some essential problems remain unsolved, such as diffusivity in the reverberation room, edge effect of test materials and so on.

In Japan, the 3rd round robin test was carried out by the cooperation of 13 research laboratories from 1965 to 1966. Details of reverberation rooms used for this round robin test are shown in Table I. Glass wool board (50 mm thick, 25 Kg/m³) was used for test materials.

Table I. Reverberation rooms used for the round robin tests

<table>
<thead>
<tr>
<th>Laboratories</th>
<th>Volume (m³)</th>
<th>RT of empty room (500 Hz)*</th>
</tr>
</thead>
<tbody>
<tr>
<td>1. Building Research Institute</td>
<td>248</td>
<td>8.82 s.</td>
</tr>
<tr>
<td>2. Inst. of Ind. Science, Univ. of Tokyo</td>
<td>192</td>
<td>5.62</td>
</tr>
<tr>
<td>3. Kajima Inst. of Const. Technology</td>
<td>140</td>
<td>3.98</td>
</tr>
<tr>
<td>6. Kobe Univ.</td>
<td>130</td>
<td>6.94</td>
</tr>
<tr>
<td>7. Nippon Asbestos Co. Ltd.</td>
<td>138</td>
<td>3.32</td>
</tr>
<tr>
<td>8. Nihon Univ.</td>
<td>169</td>
<td>7.90</td>
</tr>
<tr>
<td>9. Takenaka Tech. Research Lab.</td>
<td>164</td>
<td>5.26</td>
</tr>
<tr>
<td>10. NIK Tech. Research Lab.</td>
<td>157</td>
<td>4.63</td>
</tr>
<tr>
<td>11. Tokyo Inst. of Technology</td>
<td>74</td>
<td>3.41</td>
</tr>
<tr>
<td>12. Toyo Univ.</td>
<td>278</td>
<td>7.70</td>
</tr>
<tr>
<td>13. Yokohama National Univ.</td>
<td>144</td>
<td>6.48</td>
</tr>
</tbody>
</table>

* Total area of suspended diffusers was approximately equal to the 80% of the surface of the floor.
Test areas and the number of suspended diffusers were altered in nearly the same manner with the European round robin (1959)\textsuperscript{1}). Other specifications for measurements conformed to the ISO Recommendation R 354. Test results are illustrated in Fig. 1. Deviations of measured absorption coefficient were fairly large. Thus, investigations into the precision measurements of absorption coefficient have been pursued for these two years. This paper describes the results of investigation concerning the evaluation of the sound field in reverberation rooms and the method of measurements of reverberation time in rooms.

Conditions of the sound field in a reverberation room

In general, measurements of sound absorption coefficients in a reverberation room are carried out on the assumption of a diffuse field in a room. Diffusivity in a room is connected with room volume, room shape, diffusers put in the room, areas of test materials and so on. Among these factors, the room volume would have influence mainly upon the diffusivity in the low frequency region. Test materials used for this investigation have relatively small absorption coefficient in the low frequencies and it would be rather inadequate to discuss the requisite volume of reverberation room only through this investigation. Thus, the main project of this research was aimed at the diffuse conditions in the middle and high frequency region.

In European round robin and the resulting ISO Recommendation, the diffuse condition in a room was specified by the number of diffusers. However, this specification does not determine the sound field uniquely for different shape or volume of rooms. It would be desirable to introduce the index of diffusivity which can be measured directly in the room. Various methods or quantities have been proposed, such as fine structures of decay curve, directional distribution of sound energy, cross correlation in the sound field and so on.

Through preliminary investigations, directional diffusivity or ratio of horizontal to vertical mean energy during decay process was adopted for the evaluation of the sound field. So far as measurements
Precision of Measurement of Reverberant Sound Absorption Coefficient

of absorption coefficient were concerned, it was concluded that the
diffusivity would be sufficient, if the ratio of horizontal to verti-
cal mean energy during decay process was between 0.7 and 1.5 \(^2\).

From the measurements of this ratio, the sound fields of all rooms
could be regarded as the same order of diffusivity, when the total
area of suspended diffusers in each room would reach up to 80 \% of the
floor area of the room.

Measurements of reverberation time

To investigate the fluctuation
of absorption coefficient related to
the reverberation time measurements, experiments were conducted in the
following ways.

(1) The absorption coefficient of
identical test materials was mea-
ured by using the same instruments in
11 reverberation rooms. The rever-
beration time was read by only one
person from the recorded decay cur-
ves. The absorption coefficient
obtained is shown in Fig. 2.

Compared with the results of the
3rd round robin, deviations of
absorption coefficient measured in
different rooms become relatively
small, within ±10 \% in most frequencies.

(2) From the results of the above experiments, it would be reasonable
to think that the personal tendency for reading the decay curve would
play an important role.

Reference decay curves were recorded in one of the reverberation
rooms shown in Table I, with and without test materials. Copies of
these decay curves were sent to all research laboratories and were read
by 25 persons. Deviations of coefficient are fairly large and are
nearly the same as in Fig. 2. Statistical analysis showed a significant
difference among readers or decay curves.

(3) In the next step, decay curves were recorded by each level recor-
der in separate reverberation rooms and these records were read by only
one person, just as in Fig. 2. Deviations of absorption coefficient
thus obtained were fairly large compared with Fig. 2. From these
results, it was concluded that the dynamic characteristics would some-
times be connected with errors in absorption coefficients and it would
be necessary to adjust the level recorder carefully.

The 4th round robin test with revised specifications

The 4th round robin tests were conducted using the same test materials, where the specifications for measurements were supplemented by referring to the above investigations. Major articles are as follows: adjustments of writing speed and linearity of a high speed level recorder, proper setting of writing speed and paper speed, determination of adequate number of decay curves for each frequency and exclusion of unfit curves. Results of this round robin showed fairly small deviations among different laboratories (Fig.3).

Fig.4 shows the maximum deviations of absorption coefficient obtained for successive steps of this investigation. Corresponding results of the European round robin tests are also shown in this figure.

In conclusion, the accuracy of measurements is sufficient for usual purposes, if the deviation is kept within the range of this 4th round robin test. This would be expected under the application of the specification for measurements derived from this investigation.

---

1) C.W. Kostt: Acustica 10 (1960) 400
Application of Helmholtz Resonators in a Large Hall

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Technische Universität Hannover
Institut für Fernmeldetechnik

We had to take care for good roomacoustical conditions in a large hall with 1200 seats.

In Fig. 1 is shown the groundplan and a cross section of the hall. The base is 1150 m$^2$ and the volume 17500 m$^3$.

The hall is connected to two surrounding corridors of a total volume of 11500 m$^3$ by great apertures. The floor is made of marble, the ceiling is mainly made of glass. The other surfaces are covered with plaster.

For architectural reasons 12 m above the floor a ceiling consisting of a light transmitting screen of sheet metal is hung up. In this ceiling 85 areas of 1.3x1.3 m$^2$, equally spaced were provided for sound absorption. The usual roomacoustical provisions - damping great parts of the surfaces and the corridors - were not allowed by the architect. So for reverberation time regulation only 144 m$^2$ of the ceiling were available.

The acoustical disadvantageous shape was given by surrounding
Application of Helmholtz Resonators in a Large Hall

As the brick-work without the ceiling was finished the measurement of the reverberation time yielded the values shown in curve 1 of Fig. 2. When the glass ceiling was installed we measured the reverberation time shown in curve 2 of Fig. 2. From this was to calculate that the reverberation time at full occupation of the hall at high frequencies would reach the good value of nearly 1 sec without any roomacoustical provision. But it was necessary to raise quite an amount the absorption of the room at low and medium frequencies. For this purpose only resonators came into question.

Resonant frequency \( f_o \) and equivalent absorbing area \( A_o \) of a Helmholtz resonator (Fig. 3) are:

\[
f_o = \frac{c}{2\pi} \sqrt{\frac{G}{V}} \quad A_o = \frac{c^2}{2\pi f^2} \frac{4\alpha}{(1+\alpha)^2}
\]

\[c = \text{sound velocity} \quad G = \frac{\pi r^2}{1+\pi/2\cdot r} \quad \alpha = \frac{R_d}{R_{ro}}
\]

\[R_{ro} = \text{radiation resistance} \quad R_d = \text{damping resistance}
\]

The frequency response of the absorption is given by:

\[A = A_o \frac{(1+\alpha)^2}{[(f/f_o)^2 + \alpha]^2 + Q_o^2 v^2} \quad Q_o = \frac{c}{G\cdot r} \quad v = \frac{f - f_o}{f_o} = \frac{\alpha}{Q_o}
\]

\(\alpha\) may be evaluated by measuring the stream velocity in front of the resonators aperture. At these measurements we made good experiences with an open thermoelement with constant heating current, even at great dampings.

Calculations and measurements showed that it would be possible to achieve the desired frequency response of the absorption at low frequencies by 4 types of resonators.

Due to the small area available for the resonators they must be arranged close together. But because of the interference of the resona-
tors in this case the absorption values must not be added. For this reason the following measurements were made in an echoic chamber.

In a wooden plate (Fig. 4), 1,7 m² area and 19 mm thickness, were drilled 100 holes and above each hole was fixed a tin on the plate. In front of the apertures of the resonators was a 7 mm layer of pressed glass wool G.

Table 1 shows the hole diameters d, the volumes V, the corresponding resonant frequencies f₀ of the undamped resonators and the number of tins arranged on one plate.

<table>
<thead>
<tr>
<th>d [cm]</th>
<th>V² [cm²]</th>
<th>f₀ [Hz]</th>
<th>No. of tins</th>
</tr>
</thead>
<tbody>
<tr>
<td>3,0</td>
<td>2930</td>
<td>130</td>
<td>16</td>
</tr>
<tr>
<td>3,0</td>
<td>1810</td>
<td>164</td>
<td>24</td>
</tr>
<tr>
<td>2,5</td>
<td>900</td>
<td>204</td>
<td>22</td>
</tr>
<tr>
<td>3,0</td>
<td>900</td>
<td>233</td>
<td>38</td>
</tr>
</tbody>
</table>

3 of this plates were put in an echoic chamber. For the measurements we used small-band noise and a selective receiver with a bandwidth of 1/3 octave.

Fig. 5 shows the measured absorption coefficient. In extensive test series all parameters were varied. The above described construction was found to be the optimal one.

In order to extend the range of absorption to medium frequencies and to diminish it at high frequencies we fixed at h = 30 cm before the plate with the resonators a perforated plate of gypsum between pastboard with t = 9,5 mm thickness, p = 15 % area of the holes and d = 10 mm diameter of the holes damped by a 7 mm layer of pressed glass wool G (Fig. 6).

This system represents a variety of resonant absorbers with direct connected volumes. The resonant frequency of the undamped system is:

\[ f_0 = \frac{c}{2\pi} \sqrt{\frac{p}{h(t+0.8d)}} \]

With this combination we measured the absorption coefficient as shown in Fig. 7. It can be seen that the pre-built absorbing system for
the medium frequency range does scarcely influence the behavior of the low frequency absorbers.

85 of this units were put in the between-ceiling. In order to damp reflexions from the glass ceiling the upper sides of the absorbers were covered with glass wool and a perforated plate.

Fig. 8 shows the reverberation time of the hall.
Curve 1: empty hall without absorbers.
Curve 2: empty hall with absorbers.
Curve 3: finished hall with 1200 persons.

The curve in Fig. 9 is found by calculating the absorption of the described absorbing systems by the variation of the reverberation time in the hall. It can be seen that we get a bit different values for the absorption coefficient measured in the hall and in the echoic chamber as to be expected.

The influence of the coupled corridors causes a bend of the decay curve at nearly 30 dB sound pressure decay.

Because of the largeness and the disadvantageous shape of the hall we installed a system of small loudspeakers which get delayed signal voltages from a magnetic tape.

Therefore a speaker with normal voice is good to be understood at all places and the listener is under the subjective impression that the sound is coming only and direct from the speaker.
Utilization of non-linear properties of resonators
for improving acoustic conditions in rooms

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Institute of Automatic Control
Polish Academy of Sciences

1. Introduction

The classical theories of Eyring and Sabine are mainly based
upon the assumption that the sound absorption coefficient does not
depend on the value of the sound intensity. The purpose of this work
is the investigation of non-linear phenomena occurring in resonance
systems and of their influence on acoustic conditions in enclosures.

2. Non-linear properties of Helmholtz resonators

The non-linear properties of Helmholtz resonators occur due to
the dependence of the effective resistance of resonators on the par-
ticle velocity in the neck, which was widely elaborated by Ingard.

For the resonance frequency at values of particle velocity in
the neck \( v > 50 \text{ cm/sec} \) the part of radiated energy is lost on account
of swirls nearby the edge of the aperture. These turbulence losses
depend on the value of the particle velocity and on the sound inten-
sity.

The dependence of turbulence losses on the sound intensity, in-
fluences the variability of the absorption coefficient in the func-
tion of the sound intensity.

According to the results of the investigation of the compensa-
tion of the sound intensity radiated by the resonator with sound in-
tensity reflected from the surroundings the resonator \(^1\), two cases are
possible: (i) in the case of the preponderance of the radiated pres-
sure the increase of turbulence losses should cause the increase of
the absorption coefficient; (ii) in the case of preponderance of the
sound intensity reflected from the resonator surroundings, is should
involve the decrease of the absorption coefficient.
3. Results of the experimental investigation

Dependence of the absorption coefficient $\alpha$ of the resonator upon the sound intensity have been measured in the Kundt’s tube.

![Graph 1](image1.png)  
![Graph 2](image2.png)

**Fig. 1.** Dependence of the absorption coefficient on the frequency for the metal-box resonator for various sound intensity levels

**Fig. 2.** Dependence of the absorption coefficient on the frequency for resonator with fiber cover for various sound intensity levels

The resonance curves taken for various levels of the sound intensity for the resonator with volume $V = 75 \text{ cm}^3$, diameter of aperture $d = 2 \text{ cm}$ and thickness of cover $l = 2 \text{ mm}$ are represented in Fig. 1. It results that with the increase of the sound intensity level, the absorption coefficients at the resonance frequency is increasing too.

A contrary character of changes of the absorption coefficient in the function of sound intensity has been obtained for the resonator with fiber cover is represented in Fig. 2.

A distribution of nodal points of the standing wave sound intensity has been changed, as compared with the previous case (maxima have appeared of a distance $l_{\text{max}} = n \lambda/2$).

In Fig. 3 the dependence of changes of the absorption coefficient for the resonance frequency in the function of changes of the sound intensity level are charted. The parameters of metal box resonator was as follows $V = 16 \text{ cm}^3$, $d = 1.6 \text{ cm}$, $l = 0.3 \text{ cm}$. The results obtained show the increase of the phenomenon of the non-linearity of turbulence losses at the sharpened edge of the resonator aperture (b).
Utilization of non-linear properties of resonators

The non-linear properties of the resistance of losses appear also in the plate resonators with results from Fig. 3c for the plate made of plexiglass 3.7 cm thick rigidly fastened on edges in Kundt's tube at a distance of 7.5 cm from the rigid surface.

4. Influence of the non-linear properties of the resonance systems on the acoustic properties in enclosures

The differential equation, which describe the acoustical phenomena in enclosure while the \( \alpha = f(I) \) is an nonlinear are as follows

\[
N = \frac{(V/c)(dI/dt)}{[\alpha(I)S/4]} I, \quad (1)
\]

while: \( N \) - sound source power, \( V \) - volume of the enclosure, \( S \) - limiting area the enclosure, \( c \) - velocity of sound wave propagation.

In order to investigate the influence of the absorption coefficient from the sound intensity level, let us consider a simplified case, where the absorption coefficient \( \alpha \) changes by jumps and amounts to \( \alpha_0 \) for the pressure range from 0 to \( p_0 \) and \( \alpha_1 \) for the pressure range \( p > p_0 \). Let us assume that the jump of the value correspond to the time moment \( t = k\tau \), where \( \tau = 4NV/cS \).

Fig. 4. The character of the decay of sound for a jump of

In this case the dependence upon the sound intensity is non-stabilized state can be expresses in the form:

\[
I = \frac{4NV}{S} \left[ \sum_{i=0}^{k} (1 - \alpha_0)^i + \sum_{i=k}^{n} (1 - \alpha_1)^i \right].
\]

After transformation

\[
I = 4NV(S) \left\{ \frac{(1/\alpha_0)(1 - \alpha_1)^k/\alpha_0)}{[1 - \alpha_1]^k/\alpha_0) + [1 - \alpha_1]^k/\alpha_1)] - [1 - \alpha_1]^k/\alpha_1) \right\}. \quad (3)
\]

The sound intensity in steady-state can be expressed by formula

\[
I = I_0 + \Delta I. \quad (4)
\]

--- E-199 ---
Utilization of non-linear properties of resonators

The \( \Delta I < 0 \) for \( \alpha_1 > \alpha_0 \) and \( \Delta I > 0 \) for \( \alpha_1 < \alpha_0 \), where

\[
\Delta I = \frac{4NV/S}{(1 - \alpha_0)/\alpha_0 + [(1 - \alpha_1)/\alpha_1]}.
\]  

(5)

The formula for the sound decay is as follows

\[
I = \frac{4NV/S}{(I_0 + \Delta I)[(1 - \alpha_0)/\alpha_0 - (1 - \alpha_1)/\alpha_1]}.
\]  

(6)

In the logarithmic scale it denotes a refraction of the decay of sound with the moment \( t = k \tau \) (Fig. 4).

In the case of an increase of the acoustic power of the source, the threshold value of the sound intensity at which a change of the coefficient \( \alpha \) appears, will occur earlier for \( t_1 < k \tau \). This will involve an earlier appearance of the refraction of the curve \( I_b \), a greater change of sound intensity for the steady-state.

An example of experimental result of the change of the decay curve slope for two various levels of the sound intensity with difference 20 dB measured in an reverberation chamber, in which resonators were placed, are represented for the resonant frequency by curves in Fig. 5.

Fig. 5. Diagrams of the sound decay in the reverberation chamber with 30 resonators set up:
- (a) sound level 105 dB,
- (b) sound level 85 dB

It results from the above considerations that when making use of the non-linear properties of resonator, we can obtain a change of the sound dynamics in halls.

5. Conclusions

1. Due to non-linear properties of turbulence losses in Helmholtz resonators, the absorption coefficient for the resonance frequency may depend of sound intensity for the high intensity level. These changes increase in cases of sharp opening edges. The non-linear properties may appear in plate resonators too.

2. Depending on the conditions of the resonator surroundings the non-linearity of turbulence losses can cause an increase or a decrease of the absorption coefficient, as the sound intensity level increases.

3. The presence of resonance systems in rooms can be a cause of a curvilinear sound decay in the logarithmic scale (such as in the case of coupled rooms).

4. The non-linear properties of resonators may be utilized for a change of the acoustic signal dynamics in halls, which contributes to improving the intelligibility of speech sound in halls.

Reference

The new acoustics laboratory of the Catholic University at Leuven

Henri Myncke

Technical Director of the "Instituut voor lage Temperaturen en technische Fysica" – Leuven (Belgium)

The new acoustics laboratory at Leuven has a triple purpose: research, teaching and control measurements. Different measuring rooms have been constructed which are in complete agreement with the ISO-
recommendations, namely: one anechoic chamber, one reverberation room and four sound transmission rooms. The lay-out is shown in figure 1.

1. main entrance
2. conference room
3. electronic apparatus room
4. entrance for materials
5. sanitary installations
6. store-rooms
7. reverberation room
8. anechoic chamber
9-10-11-12. sound transmission rooms
13. laboratory and workshop
14. office (the four last rooms situated on the second floor are not shown).

All the measuring rooms are acoustically insulated amongst each others and against noise coming from outside, either air-borne or impact sound. This was obtained by a complete double wall construction, the inner concrete boxes being supported by series of metal springs fixed between metal plates mounted on a cork base, as shown in figure 2. The resonance frequency of the springs under load was determined experimentally by W. Timmerman and M. Lejeune (Laboratory for Testing of Materials). This was done by means of an inductive displacement transducer (see figure 3).

a. prefabricated concrete elements: 10 cm
b. bricks: 18 cm
c. air space with 5 cm mineral wool: 10 cm
d. concrete box: 30 cm
e. steel springs: 12.5 cm
f. cork: 7 cm
h. hard cement screed: 5 cm

The damping curve was registered with a X-Y recorder. The result for the reverberation room having a weight of 146 tons is shown in figure 3 and the resonance frequency which can be computed from this curve is 2.3 Hz, while in the case of the anechoic chamber of 228 tons, it is 3.7 Hz.

Figure 3
The new acoustics laboratory of the Catholic University at Leuven

The total insulation can be calculated in considering it as a combination of air-borne and impact sound insulation by the following formula: \( R = R_1 + 10 \log \frac{v_1^2}{v_2^2} \), \( R_1 \) being the transmission loss of the wall at the emitting side, \( v_1 \) and \( v_2 \) being the velocity of the walls. The following values were obtained:

<table>
<thead>
<tr>
<th>Frequency (Hz)</th>
<th>Sound insulation (dB)</th>
</tr>
</thead>
<tbody>
<tr>
<td>125</td>
<td>73</td>
</tr>
<tr>
<td>250</td>
<td>78</td>
</tr>
<tr>
<td>500</td>
<td>83</td>
</tr>
<tr>
<td>1000</td>
<td>88</td>
</tr>
<tr>
<td>2000</td>
<td>93</td>
</tr>
<tr>
<td>4000</td>
<td>96</td>
</tr>
</tbody>
</table>

All the measuring rooms are foreseen with a special door in the inner and in the outer wall, each having a transmission loss of:

<table>
<thead>
<tr>
<th>Frequency (Hz)</th>
<th>Transmission loss (dB)</th>
</tr>
</thead>
<tbody>
<tr>
<td>125</td>
<td>32</td>
</tr>
<tr>
<td>250</td>
<td>43</td>
</tr>
<tr>
<td>500</td>
<td>48</td>
</tr>
<tr>
<td>1000</td>
<td>51</td>
</tr>
<tr>
<td>2000</td>
<td>53</td>
</tr>
<tr>
<td>4000</td>
<td></td>
</tr>
</tbody>
</table>

A. The anechoic chamber

The main purpose of this room is to create free field conditions of sound for physical, electro-acoustical and power measurements. These conditions were controlled by measuring the sound pressure as a function of the distance and of the frequency. The deviations from the \( 1/r \) - law are given in figure 5 (1). The maximum theoretical deviations were calculated by means of the formula given by H. Biestel (2):

\[
A = 20 \log \left( 1 + \frac{p_2}{p_1} \right) \quad \frac{p_2}{p_1} = 7 \cdot r \cdot R \cdot S^{-1/2}
\]

\( r \): the distance between microphone and loudspeaker  
\( R \): the reflexion coefficient  
\( S \): the area of the interior surface.

Dimensions of chamber: 700 x 710 x 600 cm³  
Useful dimensions: 530 x 540 x 430 cm³  
Wall thickness: 30 cm  
Weight: 228 tons  
Critical frequency: 90 Hz  
Absorbing wedges: 5150 pieces of 80 cm height and a base of 20 x 20 cm²

---

---
B. The reverberation room

The development of the walls is given in figure 6, while the reverberation time measured with white noise and 1/3 octave filters is shown in figure 7. The limiting frequency: $f_{\text{lim}} = T00 \cdot V^{-1/3} = 121.3$ Hz.

Total volume: 192 m$^3$  
Wall thickness: 25 cm  
Weight: 146 t.

The total surface area $S$ agrees with the proposal: $S > 6.7V^{2/3}$, then we found: $(S = 207.5 \text{ m}^2) > 199 \text{ m}^2$. The other characteristics were compared with those proposed by P.G.Towson (3):

$$L_{\text{max}} < 1.9 \cdot V^{1/3} = 11 \text{ m}$$

$\sqrt[3]{2/3} = 11.5 \cdot e^{0.024 \cdot S'}$, with $S'$ the floor area.

We obtained, respectively 33 m$^2$ and 38 m$^2$.

C. The sound transmission rooms (4)

The four sound transmission rooms permits the measurement of sound transmission loss and insulation against impact sound in agreement with ISO R.140 and without flanking effect. The volume of each room is 87 m$^3$ and the weight 90 tons. The size of the vertical or horizontal samples is 16 m$^2$ or 4 m$^2$. Wall thickness: resp. 30 and 2 x 38 cm.

References: (1) Report nr.1129, Grünzweig und Hartmann
(2) H.G.Diestel "Zur Schallausbreitung in reflexionsarmen Räumen", Acustica, vol.12, 1962, p.113-118
(3) P.G.Towson "A study of the criteria for the design of reverberation chambers" - Arch. Science Rev. dec. 1966
(4) H.Myncke "Conception des nouvelles chambres de mesures acoustiques de l'Université Catholique de Louvain" Chaleur et Climats, n°383, nov.1967.
The Acoustic Laboratory of Rockwool AB, Skövde, Sweden.

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Civilingenjör
Rockwool AB, Skövde, Sweden.

Introduction

Rockwool AB, Sweden, is an industry for mineralwool products. In order to develop new products and test total constructions involving mineral wool products, a research station was built at the main factory in Skövde, Sweden. This research station includes an acoustic laboratory.

The acoustic laboratory admits all acoustic measurements of interest for a mineral wool industry and is build in accordance with ISO R354 and ISO R140. It is constructed to allow rapid measurements but economic factors were also taken into account when building. Ingemanssons Ingenjörsbyrå, Gothenburg, has been consulted for the acoustic problems. The laboratory gives possibilities for the following measurements.

Absorption factor measured both by the reverberation-chamber and the impedance-tube method.
Transmission Loss of walls and floors.
Impact-sound Transmission of floors.
Dynamic stiffness measurement of porous materials.
Insertion loss of ducts and mufflers.

The construction of the laboratory (See fig. 1)

The reverberation room is built of reinforced concrete, the floor 25 cm, the
walls and the roof 20 cm thick. Volume: 200 m³.
Dimensions: 4,65 x 5,85 x 7,35 m, parallell
walls. The room is isolated against solid born
sound from the other rooms and from the building
itself. It rests on 8 cm mineral wool (Rockwool
= 200 kg/m³). 30 diffusers are suspended from
the ceiling in order to obtain a diffuse sound-
field. The diffusers consist of laminated plastic
sheets, 3 mm thick and with the dimensions 125 x
80 cm. In the floor a concrete slab (3,85 x 2,60
x 0,10 m, area 10 m²) is placed on an electro-hydraulic elevator. The slab can be
lowered about 80 cm. Thereby we can measure the absorption with different airspaces
behind the sample without moving it. This is a very time-saving arrangement.

Towards the transmitting room there is a test opening for walls being tested,
dimensions 3,60 x 2,80 m, area 10 m². This opening can be closed by a concrete door
operated by an electric motor.

The transmitting room and the "impact-noise room" is built in one unit. The
walls of the rooms are built of brick, transmitting room 24 cm thick and impact noise
room 12 cm. It is all erected on a slab of 25 cm reinforced concrete, isolated from
the rest of the building by 8 cm mineral wool (Rockwool, = 200 kg/m³). To avoid
flanking transmission the rooms have floating floors of 5 cm concrete on 5 cm Rock-
wool ( = 150 kg/m³), and the walls and ceilings have damping panels of 2 x 13 mm
gypsum board and 13 mm porous woodfiberboard on wood studs 2" x 4" with 10 cm Rock-
wool ( = 40 kg/m³). There are two test openings between the rooms, 2,5 x 4 mm each.
In one of them is placed a concrete slab, 16 cm thick. The other can be provided with
an optional floor.

Dimensions: Transm. room 6,2 x 5,0 x 3,9 m, 124 m³, Impact Sound room 6,2 x 5,0
x 3,6 m, 107 m³.

Test measurements. The reverberation times of the three rooms are shown in tab. 1.
The Acoustic Laboratory of Rockwool AB, Skövde, Sweden

<table>
<thead>
<tr>
<th>Frekvens Hz</th>
<th>125</th>
<th>250</th>
<th>500</th>
<th>1000</th>
<th>2000</th>
<th>4000</th>
</tr>
</thead>
<tbody>
<tr>
<td>Rev. room, without diffusers</td>
<td>17</td>
<td>11</td>
<td>9</td>
<td>6,5</td>
<td>4,5</td>
<td>2,5</td>
</tr>
<tr>
<td>with diffusers</td>
<td>11</td>
<td>8</td>
<td>6</td>
<td>5</td>
<td>4</td>
<td>2,5</td>
</tr>
<tr>
<td>Transm. room</td>
<td>1,8</td>
<td>2,7</td>
<td>2,6</td>
<td>2,5</td>
<td>2,1</td>
<td>1,6</td>
</tr>
<tr>
<td>Imp. noise room</td>
<td>2,0</td>
<td>2,6</td>
<td>2,4</td>
<td>2,4</td>
<td>2,1</td>
<td>1,7</td>
</tr>
</tbody>
</table>

Tab. 1. Reverberation Time for the rooms.

During the installation of the diffusers there were some absorption measurements undertaken. The results are established in fig. 2. The absorbing material is the standard material used in second Round Robin (C.W. Kosten, Acoustica Vol. 10, 1960, p. 400 - 411). The fig. shows the abs. factor as it varies with the number of diffusing elements. It is rather obvious that the variation between 20 and 30 diffusers is small, indicating that the diffusion is good.

Fig. 3 describes the influence of different edge conditions. The accordance is rather good between the material enclosed by a reflecting wood border and the material placed on the concrete slab and this lowered so that the material is in level with the floor.

In the same figur the results from the second Round Robin are drawn; corresponding to a volume of 218 m³.

The maximum transmission loss we can measure in a laboratory is limited by the flanking transmission. This must therefore be very small. It is of great value to know the flanking transmission.

Therefore the flanking transmission has been measured by vibration measurements. (T. Kihlman, The influence of flanking transmission on insulation against airborne sound; BFR. no. 39). The results from one of these measurements are shown in fig. 4. The thick line shows the measured T.L. of the test wall consisting of 12 cm brick, 20 cm Rockwool (ρ = 40 kg/m³) and 12 cm brick.

The thin lines show the transmission loss from the different areas in the receiving room obtained from vibration measurements. The thin solid line refers to the brick-wall. These measurements indicate that it is rather impossible to measure walls with still higher T.L. because of the flanking transmission.
Fig. 2  $\alpha_s$ of 50 mm sillan ($\delta = 100 \text{ kp/m}^3$) with different number of diffusers.

Fig. 3  $\alpha_s$ of 50 mm sillan ($\delta = 100 \text{ kp/m}^3$) with different edge conditions.

Fig. 4  Transmission loss for test wall and flanking areas.
ACOUSTICS OF THE KUMASI UNIVERSITY ASSEMBLY HALL

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Introduction

The Assembly Hall of the University was designed by Max Gerlach and D.H. Gillies (of Gerlach and Gillies, Reyburn, England) and the acoustic consultant was J.H.A. Crockett, London with D.B.O'Neil as his associate. The Hall which is not yet officially opened was completed in 1966 at a cost of over quarter million pounds.

The Hall was intended to be used for University functions like convocation and congregation, musical performance, showing documentary films, public lectures, etc.

In this paper an attempt has been made to compare the designed values of reverberation times with those obtained experimentally.

DESCRIPTION OF ACOUSTIC FEATURES OF THE HALL

The length of the hall is 140 ft., its width is 90 ft., and the maximum height (from floor to ceiling) is 36 ft. The effective volume is 419,000 cu.ft. The hall has the seating capacity of 2000 audience. Along the length on both sides there are galleries: 24 and 14 clerestory windows and doors respectively which the designers advise should normally remain open, thus allowing a great deal of acoustic energy to escape which in effect lowers the reverberation time.

There is about 15 x 90 ft. balcony with audience capacity of about 200; here there are 7 rows of benches with cushioned tops. The ceiling which is covered with porous absorbers also supports concave surface reflectors. The reflectors should physically behave like a (acoustic) concave mirror, diffusing the sound field in all directions, but slightly focussing it to the central area of the floor where audience number is expected to be high with consequent high audience absorption. (1).

The reflectors are broken up, (Fig.2) presumably for smooth exponential decay of the sound energy. The rear doors are all covered with absorbing materials. Provision for sound amplification system has been made, but not yet installed.
**Acoustics of the Kumasi University Assembly Hall**

**Design Valued for Reverberation Time**

<table>
<thead>
<tr>
<th>Reverberation Time</th>
<th>Doors and windows closed</th>
<th>Reverberation Time</th>
<th>Doors and windows opened</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>125Hz</td>
<td>250Hz</td>
<td>500Hz</td>
</tr>
<tr>
<td>Empty Hall</td>
<td>2.37 sec</td>
<td>1.76 sec</td>
<td>1.4 sec</td>
</tr>
<tr>
<td>Half Full</td>
<td>1.96 sec</td>
<td>1.45 sec</td>
<td>1.09 sec</td>
</tr>
<tr>
<td>Full Hall</td>
<td>1.86 sec</td>
<td>1.38 sec</td>
<td>1.00 sec</td>
</tr>
</tbody>
</table>

**Measurement of Reverberation Time When the Hall Was Empty**

Measurement of reverberation time at frequencies centred on 125, 250, 500, 1000, 2000 and 4000 Hz were made in the Hall under the condition of doors and windows opened and closed respectively. During measurement there were in the Hall 549 metal chairs (unoccupied) of which 31 had cushioned seats. There were a piano and an electric organ on the stage. A woolen carpet was also spread on the stage covering about two-thirds of the total stage area.

Standard measuring equipments of Brüel and Kjær such as Beat Frequency Oscillator type 1013, Audio Frequency Spectrometer Type 2112, 1/2 inch omnidirectional condenser microphone Type 4133 and the Level Recorder Type 2305 were used. For the sound amplification, the output from the B.F.O. was fed to a 20 watt a.f. Amplifier (Phillips Type EL 6405) and a 10" loudspeaker mounted in a baffled cabinet. The desired 'warbled' tone was generated by using the frequency modulation device of the oscillator.

The loudspeaker was first placed at the extreme left, then at the middle and finally at the extreme right of the stage and for each position of the loudspeaker decay curves were obtained for nine selected position of the microphone. This in all gave 27 recordings for a single frequency. Each recording was also repeated for not less than 5 times. The paper and writing speed were chosen as 100 mm per sec. and 160 mm per sec. respectively for the 50 dB potentiometer setting of the level recorder.

Reverberation times were calculated using the protractor and no readings were taken from curves which were not straightlined up to 20 dB. Average reverberation times are given below in Table II.

**Average Reverberation Times in Sec.**

<table>
<thead>
<tr>
<th>Frequency in Hz</th>
<th>Doors and windows opened</th>
<th>Doors and windows closed</th>
</tr>
</thead>
<tbody>
<tr>
<td>4000</td>
<td>1.87</td>
<td>2.07</td>
</tr>
<tr>
<td>2000</td>
<td>2.08</td>
<td>2.32</td>
</tr>
<tr>
<td>1000</td>
<td>2.47</td>
<td>2.82</td>
</tr>
<tr>
<td>500</td>
<td>2.67</td>
<td>2.88</td>
</tr>
<tr>
<td>250</td>
<td>2.95</td>
<td>2.40</td>
</tr>
<tr>
<td>125</td>
<td>2.00</td>
<td>2.22</td>
</tr>
</tbody>
</table>

**Measurement of Reverberation Time of the Hall When Full**

The experiment was repeated during the University Congregation when the Hall was full, the doors and windows were all opened and a sound amplification system
was used in the hall to reinforce the sound pressure level. The microphone and the
level recorder were placed at the rear end under the balcony. The results are
given below in table III.

<table>
<thead>
<tr>
<th>FREQ. IN HZ</th>
<th>250</th>
<th>500</th>
<th>1000</th>
<th>2000</th>
</tr>
</thead>
<tbody>
<tr>
<td>Average Reverberation time in sec.</td>
<td>1.95</td>
<td>1.06</td>
<td>1.05</td>
<td>1.30</td>
</tr>
</tbody>
</table>

DISCUSSION OF RESULTS

(a) Reverberation Curves: These are drawn from tables 1, 11 and 111 and are
shown in fig.1. Figs 1(a) and 1(b) are the curves corresponding to measurements
when the hall was empty and full respectively. Figs 1(c) and (d) are the curves
drawn from designed figures when the hall is empty and full respectively. Since
the shape of the curve (a) has a completely different form than that of (b) it can
be argued that it is the absorption of the audience which determines the shape of
the reverberation curve for a large hall; shape of the hall has only a slight
influence. Similar behaviour has been observed for large halls (2) like St. Thomas
Church at Leipzig; Musikaal Town Casino, Basle, and many others.

(b) Reverberation time: The reverberation time, as measured, was 2.47 sec. at
the mid-frequencies when the hall was empty and the doors and windows were all
opened. This time reduces to 1.06 sec. when the hall is full. The designers
expected 1.17 sec. for an empty hall and 0.91 sec. for the full hall. It can now be
concluded that the designed value and the measured value of the reverberation time
agree well when the hall is full. From the shape of the freq-reverberation time
curves, one could also infer that the audience absorption was greatest at the
mid-frequencies.

(c) Sound Level in the Hall: During the congregation, a sound level metre
(which was calibrated at a standard reference sound pressure of 0.0002 dyne per
sq. cm.) was used to determine the sound pressure level. On the average this
fluctuated between 54–68 dB when measurements were made under the balcony.

Personal interview conducted after the ceremony indicated that the addresses
were audible at all positions. On another occasion, when a play was staged in the
hall with no provision of sound amplification and the hall was almost full, a large
number of audience at the balcony had to come down to the main floor because of low
sound level.

CONCLUSIONS

As to the reverberation time of the hall when full, there is close agreement
between the designed value and the measured value which are 0.91 sec. and 1.06 sec.
respectively at mid-frequencies. It should be however borne in mind that the
designers considered that 2000 seats should be upholstered. Hence if the Hall is
furnished according to the plan, these two figures would come closer. Nevertheless, with a reverberation time of 1.06 sec. when the Hall is full, the intelligibility for speech sound should be very good. According to Knudsen (3) the optimal reverberation time for speech sounds in a hall of volume 400,000 cu.ft. should be about 0.9 sec.

Since the average intensity of speech sound level for average speaker does not exceed 50 dB when the hall is full (4) whereas the level should be about 70 dB for most favourable hearing conditions, the use of a sound amplification system is a necessity whenever the hall is full.

The sound pressure levels measured along both sides of the Hall indicate that they are very nearly equal, but they are less than that at the middle of the hall. This shows that the reflectors actually behave like acoustic concave mirrors which attempt to diffuse the sound in all directions, but slightly focus it to central area of the hall.

According to few subjective opinions, the hearing conditions along the sides of the hall are not as favourable as around the central portion. This indicates that the sound pressure level is not the same everywhere in the Hall. Although 100% diffusion of sound with equal intensity is difficult to achieve in practice in any hall anywhere, the designers should have given more thought to achieve this goal.

REFERENCES


Die akustische Konstruktion des Fernseh-Zentrums in Berlin

Dietrich Schwarze
Leiter der Hauptabteilung
Technische Planung und Ausrüstung im Sender Freies Berlin

Einleitung und Übersicht


Das Fernseh-Zentrum gliedert sich in (Abb. 1)
Werkstätten mit Montagehalle und Fundus, - Produktions-Studioblock (Studio A: S = 800 m², V = 10.600 m³; Studio B und Studio C: S = 560 m², V = 7.020 m³)

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Die Akustik des Fernseh-Zentrums in Berlin

Technisches Zentrum mit zentralem Geräteraum und drei Senderegion mit Ansagestudios \( S = 80 \, m^2, V = 450 \, m^3 \)
Technik-Haus mit Synchronstudios, Schneide- und Umschnitträumen
2 Aktualitäten-Studios, Studio D + E mit \( S = 200 \, m^2, V = 1600 \, m^3 \)

Abb. 1 - Lageplan des Fernseh-Zentrums

(8) Montagehalle, Handtischlerei, Maler; (9) Maschinentischlerei;
(14) Studios A - E; (18) Regieraume, Ton, Bild, Licht;
(19) Zentraler Geräteraum und Hauptschaltraum;
(20) Magnetische Bildaufzeichnung

Studioblock und Werkstatt befinden sich an der gemeinsamen 6 m breiten Studiostraße. Unter den 3 großen Produktionsstudios sind 3 ebenso große Probenstudios angeordnet, deren Höhe jedoch nur 3,60 m beträgt. Damit können die sogenannten Kaltproben, d.h. Proben ohne elektronische Bild- und Toneinrichtungen, unter gleichen Bedingungen stattfinden, wie die Aufnahmen in den Produktionsstudios mit Technik.

Die akustischen Bedingungen

Das Fernseh-Zentrum befindet sich an einem der verkehrsreichsten
Die Akustik des Fernseh-Zentrums in Berlin


Der nächstgelegene Flugplatz Tegel ist 6,6 km entfernt. Düsenflugzeuge überfliegen das Fernseh-Zentrum in Höhen bis 800 m, Hub- schrauber in Höhen bis 100 m. Aus all diesen Störgeräuschen ergeben sich die bauakustischen Forderungen für die Studios[2].


Eine solche in Abb. 2 angegebene Luftschalldämmung von 75 dB bei 4000 Hz ist nur in einer Bauweise nach dem Prinzip "Haus in Haus" zu erreichen, wenn man gleichzeitig eine hinreichende Körperschalldämmung erzielen will.

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Die Akustik des Fernseh-Zentrums in Berlin

Die innere Schale einschließlich der Deckenkonstruktion mit abgehängener Beleuchterdecke ist ein mit Mauerwerk ausgefachter steifer Stahlrahmen, der auf Stahlfedern steht. Der Studiofußboden ist auf Längsdämmbügeln (LDB) aufgelegt. Zur Erzielung einer hinreichend kurzen Nachhallzeit, $T = 0,8 - 1$ sec, steht vor der inneren Studioschale eine Lochsteinwand ($\sigma = 39\%$) mit hinterlegten Mineraldämmplatten.

Die akustischen Messungen

Die durchgeführten Körperschallmessungen an den inneren und äußeren Wandschalen bestätigen die Berechnungen, nämlich nahezu eine konstante Pegeldifferenz im tiefen Frequenzbereich mit einem nachfolgenden Anstieg um ca. 12 dB je Oktave. In Abb. 3 sind die vergleichbaren Meßpunkte an den 3 Studios aufgetragen.

![Diagram](image)

**Abb. 3**

**Literatur**

1. Delang und Kosten, Schalltechnik 22 (1962)
Introduction

The world-renowned Karlovy Vary Spa will have a new area of cultural and cure buildings designed by the architects Věra and Vladimír Machonin, and their associates. In the final course of designing this area, I was requested by the authors for the cooperation in the design for good acoustics of the Festival Cinema-theatre, as an expert. The building site is already coming up this year.

Being assembled to one block, the buildings are situated on the bank of Teplá-river. The Festival Cinema-theatre grows up cantilever-like from one corner of the block. It has a seating capacity of 1,200 and volume of 13,000 m³.

Structure and Acoustic Design of Cinema-theatre

The loadbearing structure of the hall stands on six pillars only. Being shaped as shown in Fig. 1 it is completely built from steel rods. The projecting part of the hall is more than 20 m in length. It is evident that the projecting part of the hall must be as light as possible.

As the hall is not consequently surrounded by various silent rooms, then it has not usual effective protection against exterior noises. Therefore the high sound isolation by means of high sound
transmission loss of its walls, floor and ceiling is demanded. However, the Berger's law /1/ makes the last-mentioned demand contrary minded in comparison with the paragraph mentioned above. The coat of the hall is to be as heavy as possible, according to this law.

The structures of floor and ceiling are loadbearing ones and no additional mass was demanded for them. They are massive enough to protect the public area against both mechanical loads and exterior noises. However, walls do not bear any other load except their own. If built by usual methods they are to weight not less than 450 kp/m² from the point of view of sound isolation.

This weight makes the loading on the hall up to 5,000 kp/m in length what is already, from the point of view of statics, unacceptable. With agreement of Prof. Ferdinand Lederer, the weight of the wall was limited to less than 250 kp/m². It is evident that the wall was to be solved by some unconventional method. As the wall was considered to be one of the most interesting problems in acoustic design I have decided to pay attention only to its structure.

Method used for the Wall Design

To the next design, the theory of Wintergerst /2/ and Gösele /3 and 4/ simplified by Mouric and Schwarz /5/ was used. It was evident that in connection with the demand of statics the simple wall was unacceptable. These walls built of traditional materials as concrete or bricks behave as "rigid" ones. Their wave coincidence is used to be found under the sound isolation frequency-band, i.e. under 100 Hz. The growth of transmission loss in the dependance of frequency is 4,5 dB/octave only and for the above mentioned demand the mass of
450 kg/m² is wanted. The diagram of transmission loss of such a wall is shown in Fig. 2.

Key: 1 - concrete panel, 8 cm, 185 kg/m²; 2 - mineral-fibrous leave, 2 cm, 0.36 kg/m²; 3 - sheet-steel, 2 mm, 15.7 kg/m²; 4 - rock-felt sheet in plastic foil, 6 cm, 7.4 kg/m²

a - T.L. demanded; b - T.L. of a "rigid" wall, 450 kg/m², estimated; c - T.L. of a "compliant" wall, approx. 200 kg/m², estimated; d - T.L. of the wall, approx. 200 kg/m², tested

Fig. 2 Airborne-sound transmission loss

Dividing such a wall into two thinner leaves a system of "compliant" walls can be obtained or at least, one of them with the wave coincidence over sound isolation frequency-band, i.e. over 3150 Hz. The growth of transmission loss of it makes 6 dB/octave up to approximately 1 octave before the "critical" frequency.

In the designing I was to accept the demand of architect that the external coat of the wall would be, from aesthetical reasons, formed by concrete panels 8 cm in thickness. Concrete slab of this thickness behaves as a "rigid" wall and consequently it is necessary to apply an additional but "compliant" one. The steel sheet 2 mm in thickness was chosen as the most suitable of all the materials both from fire and economical reasons.

As the dampness of sheet-steel itself is very small I decided to apply a rock-felt sheet 7.4 kg/m² in mass, directly connected to it. The "own" frequency of the system concrete panel - air - sheet-steel
was also necessary to be put under the sound isolation frequency-band. That's why the air-break of 53 cm in thickness was made. At last the effect of standing waves in the air-break was to be paralyzed. I decided to solve this task by means of two mineral-fibrous leaves 0.36 kg/m² in mass. The final structure of the wall and the estimated course of its airborne-sound transmission loss are shown in Fig. 2. The frequency 2,500 Hz among \( f_{k1} = 200 \) Hz of concrete panel and \( f_{k2} = 5,550 \) Hz of sheet-steel was considered as "critical" one.

Preliminary designed structure of the wall was subjected under the acoustic laboratory measurements. The result was satisfactory as shown in Fig. 2. I have to add that the mass up to 250 kg/m² for sound absorption tilings was preliminated.

Obtained Results

The obtained results in these points may be summarized:

1. The mass of the wall by means of a suitable structure to 50% of any traditional wall was lowered. It is possible to realize the intention of architect without using additional pillars under the projecting part of the wall. - 2. The result of laboratory measurements corresponds to more than three times greater mass of any simple wall built from frequent building materials. - 3. The result obtained corresponds to the preliminary estimation.

It was impossible to reproduce the final form of the wall but the slides pictures of it will be projected. The equations applied must be found by the reader in the references.

Acoustical Design of Corbett Auditorium Building

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University of Cincinnati

Acoustical Requirements for the Auditorium and Building

The major facility in this building is an auditorium providing complete stage facilities for grand opera. The seating capacity of only approximately 800 (variable for different functions) is not intended to be commercially compatible with the full stage, because the facility is primarily for education. Other important uses for the facility include concerts on a 60-rank organ; large symphonic orchestra, band and chorus concerts; small choral, string, woodwind, brass, or percussion groups; piano, harpsichord, solo instrument or voice recitals; and convocations. The wide range of reverberation times needed for such varied functions defined the first acoustical problem.

It is intended that this auditorium simulate the acoustics of typical, larger acoustically satisfactory opera houses and concert halls. To accomplish this with a stagehouse nearly one-third larger in volume than the maximum listening space available was the second acoustical problem.

Directly below the main floor of the auditorium are a 150-seat recital hall and a rehearsal hall used for symphonic orchestra, band, and choral groups of 100 to 150 musicians. All three of these facilities are scheduled for simultaneous use throughout most of the day and evening. To prevent disturbing transmission between these spaces was the third acoustical problem.
Reverberation Control

An acoustically rather transparent ceiling with downward lighting and darkened surfaces above permitted acoustical use of the entire space between the roof and the auditorium floors. The effective acoustical height averages 46 feet although the effective visual height averages only 33 feet. By holding the fixed sound absorption primarily to aisle carpeting and theatre chairs with upholstered cushions and backrests, and by using a fairly rigid and well sealed orchestra shell to isolate the entire proscenium opening from the huge absorptive stagehouse, it was possible to obtain for organ a mid-range reverberation time of 2.2 seconds without audience, which lowers to 1.7 seconds for a capacity audience. This is not a cathedral condition but it enhances the organ sounds quite satisfactorily. Before the orchestra shell was installed the organ voicers were at a loss because of stagehouse absorption.

When the shell cannot be scheduled (e.g. during opera preparations) the proscenium can be partially closed by panels which slide in from the ends. Lowering
the fire curtain completes the closure satisfactorily except at low frequencies (below 250 Hz). When not used the organ opening is closed electrically by panels which complete the sidewall.

For symphony orchestra the optimum reverberation time is approximately 0.2 seconds lower. This can be obtained with the orchestra in the shell by unrolling electrically approximately 1100 Sabins of Fibreglas blanket distributed in the above-ceiling space.

Smaller ensembles or soloists perform on the orchestra lift in front of the stage curtain, sometimes with a small portable shell directly behind the musicians. The curtain subtracts the shell volume from the room and adds 600 Sabins more, lowering the time to 1.3 seconds for full audience (1.6 seconds for no audience) which is suitable for chamber music and certain instruments. For small choral groups desiring more reverberation the Fibreglas blankets can be rolled up.

Opening the curtain for opera is roughly equivalent to having the curtain closed, because the stage sets reflect some sound into the auditorium. Reverberation during a well attended opera performance is about 1.4 seconds, optimum for opera in an auditorium of 300,000 cubic feet. For Wagnerian opera longer reverberation can be obtained by raising some or all of the Fibreglas blankets.

The audible effect of the variable absorbers in the above-ceiling space is much greater than the measured change in reverberation time would suggest. A sensation of spaciousness largely disappears when the upper part of the room becomes absorptive, which is very useful in going from organ to chamber music.

The room design should be ideal for performing electronic compositions using distributed sound sources. In addition to complete freedom of invisible ceiling loudspeaker locations there are lighting slots concealed in sidewall offsets and also rear-wall sound and lighting control room windows which can be opened.

Shape and Diffusion

Room width varies from only 34 feet at the proscenium to 72 feet at the rear, allowing early reflections to follow the direct sound quite closely. The orchestra shell extends the small taper of the auditorium walls and ceiling. A high degree
of diffusion is produced by cylindrical shell panels, divergent shaping of the ceiling struts, occasional small inverse-taper breaks in sidewall trend, alternate protruding sidewall panels, and a deeply serrated contour for the balcony front and for the rear wall both below and above the balcony. The latter effectively prevent "backslash" to the stage without any highly absorptive treatment of the rear wall.

Noise Level

Noise spectra measured with air conditioning on in the three major listening spaces of the building yielded NC levels of 21 in the auditorium, 22 in the recital hall and 25 in the orchestra rehearsal hall. These very low levels of ambient noise make inaudibility of music transmission between the spaces rather difficult to achieve because masking is negligible. Nevertheless with a simultaneous bass viol recital, a 120 piece band rehearsal, and an organ concert there was no audible interference. Only when several doors in sequence are all open is transmission audible.

Sound Isolation Method

The auditorium has both inner and outer walls with small lighting, storage, organ, and stairway rooms between. The entire main floor seating area consists of separate concrete slabs isolated from the building slab by compressed Fibreglas pads. The ceiling of the recital hall below is suspended from the building slab on hangers compressing Fibreglas pads, with the intervening air space absorptively treated. The orchestra rehearsal hall ceiling has two stages of this type of isolation. Both of the lower rooms have independent concrete floors resting directly on the earth. The inner sidewalls of both rooms rest upon the independent slabs and are spaced from surrounding building walls. The edge of the ceilings is sealed to the sidewalls by Neoprene sponge strips under compression.

Acknowledgment

The architectural cooperation of Mr. Edward Schulte and his associates, and the support of Dean Jack Watson and of Mr. Ralph Corbett, the donor, are noted.
Acoustics of Powell Symphony Hall, St. Louis (1968)

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Introduction

The St. Louis Symphony Orchestra (the second oldest symphony orchestra in the U.S.A.) held its inaugural concert in its new home, Powell Symphony Hall, on January 24, 1968, under the direction of its conductor, Eleazar De Carvalho. The new concert hall was formerly a motion picture theater, the interior of which was completely rebuilt. Opened in 1925 as the St. Louis Theater, it was designed by the firm of Rapp and Rapp, Architects, who were well known in the 1920s as the architects for the Paramount Theater in New York City. The St. Louis Theater had a seating capacity of 3900. This is in contrast with the seating capacity of 2689 for Powell Symphony Hall. The Architects for the new hall were the firm of Wedemeyer, Cernik and Corrubia; and the associate architect was Ben Schlanger.

Structure and Acoustical Design

The principal acoustical problems in the design of Powell Symphony Hall were: (1) to reduce the original volume of the theater, (2) to add significant elements of diffusion to the hall, (3) to select suitable materials and furnishings so as to yield optimum reverberatory characteristics, and (4) to design and construct a shell in the stage area.
Acoustics of Powell Symphony Hall

The first two problems were solved jointly in the following way. The volume of the original theater was reduced considerably to the new value for the Hall of 780,000 cubic feet. One reduction in oubage was achieved by decreasing the depth of the seating area in the balcony; i.e., the rear wall was moved forward. In addition, arches which had served as exits to aisles around the auditorium at orchestra level were closed off. Large volumes near the splay on either side of the proscenium, which once had been locations for organ pipes in the old theater, were enclosed by heavy plaster convex surfaces which formed portions of the new wall construction in the auditorium. These convex surfaces were added to promote diffusion of sound in the hall.

Powell Symphony Hall has one balcony. It is unusually high above the orchestra floor, being separated from it by a tier of boxes which are shallow in depth. As a result there is a particularly good flow of sound in the area beneath the balcony soffit.

Over the original concrete floor was constructed an oak floor having a thickness of slightly more than \( 1\frac{1}{2} \) inches. Upholstered chairs having exposed wood backs were selected. A minimum amount of low-pile carpet was used in the aisles only. Some of the original walls were back-plastered to increase their mass. The resulting reverberation time in Powell Symphony Hall, unoccupied, in the intermediate frequency range is 2.9 seconds.

The stage area is approximately 2200 square feet. It has a wood floor similar in construction to that used in the Hall. The shell, which was built on the stage of the old theater, was constructed of wood, backed with heavy plaster, and also of heavy plaster on lath. Both its vertical walls and overhead ceiling splay are highly articulated in order to provide for increased diffusion. At the rear of the shell the width is 46 feet; at the proscenium the width is 66 feet. The height of the overhead splayed ceiling is 35 feet above the stage floor at the rear of the shell and it rises toward the proscenium at an angle of \( 17^\circ \) with respect to the horizontal.
The stage incorporates a series of four risers. These risers, on which a portion of the orchestra plays, have an oak flooring similar to the stage floor, approximately \( 1\frac{1}{2} \) inches thick. Although heavy, they are built in sections and are removable.

Presentation of this paper will be accompanied by photographs showing Powell Symphony Hall before and after its construction. Plans and sections of the Hall will be presented, together with graphs showing the occupied and unoccupied reverberation time vs. frequency characteristics.
Palaceside Building, Tokyo, which is occupied by Mainichi Newspapers Tokyo Head Office, has also a lot of rentable offices. Because of this occupancy characteristics, it was necessary for the architects to prevent the rotary press vibration from being transmitted to the office spaces.

It was then known to the architects that a conventional practice depended on the use of dry sand for vibration-proofing in this sort of cases; however, it was also known that the sand was not the most effective solution to the problem and that its properties would deteriorate with a lapse of time. The author made some research studies on the properties of sand and other alternatives such as springs and rubber cushions and, after consultation with the Mainichi's representatives, suggested the use of rubber cushions.

A unit cushion was composed of two sheets of synthetic rubber (fig.1) with a sheet of steel (fig.2) sandwiched in between. These cushions were placed under the concrete bed of the rotary presses in a manner as shown in

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Rubber Cushions minimize Vibration Transmission from Rotary Presses

fig. 4. A number of concrete blocks were placed on the concrete floor slab, and the cushions were adhered to the tops of concrete blocks so as to minimize the vibration transmission to the floor slab. Each cushion was covered with steel sheet (fig.3) so that rubber would not be directly exposed to oil which might drop from the machines above. On these protective steel sheets were laid 18 mm thick cement asbestos boards on which was poured 800 mm thick concrete bed. The concrete bed was isolated from the rest of the structure by means of 100 mm thick carbonized cork sheets, as shown in Fig.5, b).

Fig. 5, a) shows the entire plan of a concrete bed which measures 3.9 m by 94.8 m. The printing plant has two rows of such concrete beds, each weighing about 1,800 tons, and having 585 cushions. Each bed accommodates 30 rotary presses and 9 folding machines which weigh about 1,000 tons.

The natural frequency of this vibration system measured after the high speed rotary presses had been installed was 9.5 c/s approx. and the deflection of rubber cushions was 2.8 mm approx. The studies in the course of design indicated that the natural frequencies of similar rotary presses were in varying ranges of 17-20 c/s, 38-42 c/s, 120-180 c/s., etc. among which the 17-20 c/s appeared to be most prevailing. From the measurements of both the vibration of the presses in operation and the vibration-proofing effects of rubber cushions, the frequencies were known to be in a 22-25 c/s range and a 100-110 c/s range in the first zone and the second zone respectively, with 25 c/s and 100 c/s most prevailing.

Measurements were taken both when the presses were in operation and when they
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were at rest, with the measuring
instruments placed at eight locations
on the press beds and the structural
floor. From some 140 readings of
maximum amplitudes, the ratios of the
in-operation time amplitudes to the
at-rest time amplitudes have been
computed for the identical frequen-
cies and are shown in Table 1.

From the results it is known
that when the presses were in
operation, the amplitudes as measured
at the press bed (Point D) increased
to 5 to 12 times the rest-time amplit-
itudes; however, the in-operation
time amplitudes at the structural
floor (Point B) showed an increase of
only 2 to 5 times the at-rest time
amplitudes. Table 2 indicates the maximum in-operation time amplitudes recorded at

![Fig. 5 Plan and section of concrete bed and rubber cushions](image)

Table 1. Max. amplitudes of vibration of the concrete
bed for rotary press and concrete slab of
the building ($\mu = 10^{-3}$ mm)

<table>
<thead>
<tr>
<th>Direction of vibration</th>
<th>c/s</th>
<th>On the concrete bed (D)</th>
<th>On the floor (B)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>in operation $M$ ($\mu$)</td>
<td>in operation $M$ ($\mu$)</td>
</tr>
<tr>
<td>Vertical</td>
<td></td>
<td>1.63</td>
<td>0.14</td>
</tr>
<tr>
<td></td>
<td>25</td>
<td>1.21</td>
<td>0.24</td>
</tr>
<tr>
<td></td>
<td>100</td>
<td>1.67</td>
<td>0.084</td>
</tr>
<tr>
<td>Horizontal</td>
<td></td>
<td>0.84</td>
<td>10.0</td>
</tr>
<tr>
<td>(longitudinal)</td>
<td>25</td>
<td>1.67</td>
<td>0.034</td>
</tr>
<tr>
<td></td>
<td>100</td>
<td>3.01</td>
<td>0.033</td>
</tr>
</tbody>
</table>
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respective measuring points.

**Table 2. Max. in-operation time amplitudes measured at 8 different positions**

<table>
<thead>
<tr>
<th>Measuring points</th>
<th>Floor level</th>
<th>Location</th>
<th>c/s</th>
<th>Maximum amplitude (µ)</th>
<th>Direction of vibration</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>B4</td>
<td>bed</td>
<td>100</td>
<td>3.30</td>
<td>longitudinal</td>
</tr>
<tr>
<td>B</td>
<td>B4</td>
<td>floor (column base)</td>
<td>110</td>
<td>0.52</td>
<td>&quot;</td>
</tr>
<tr>
<td>C</td>
<td>B4</td>
<td>floor (&quot; )</td>
<td>100</td>
<td>0.42</td>
<td>transverse</td>
</tr>
<tr>
<td>D</td>
<td>B4</td>
<td>bed</td>
<td>100</td>
<td>3.42</td>
<td>longitudinal</td>
</tr>
<tr>
<td>E</td>
<td>B3</td>
<td>floor (column base)</td>
<td>110</td>
<td>0.31</td>
<td>&quot;</td>
</tr>
<tr>
<td>F</td>
<td>1st Flr</td>
<td>floor (on beam)</td>
<td>16</td>
<td>1.09</td>
<td>vertical</td>
</tr>
<tr>
<td>G</td>
<td>1st Flr</td>
<td>floor (center of slab)</td>
<td>15</td>
<td>0.98</td>
<td>&quot;</td>
</tr>
<tr>
<td>H</td>
<td>B4</td>
<td>floor (&quot; )</td>
<td>100</td>
<td>0.38</td>
<td>&quot;</td>
</tr>
</tbody>
</table>

As indicated, the amplitudes at the floor were only 0.3 to 1.1/µ max. while those on the press bed were 3.3 to 3.4/µ. Moreover, the amplitudes in neighborhood of 1.0/µ recorded on the floor was caused only by a low frequency vibration of 15 or 16 c/s; therefore, the vibration of presses must have been beyond human perception except for the persons who happened to stand on the press bed.
Aali Ghapoo

Ein Schloss aus dem Mittelalter im IRAN
mit raumakustischen Merkmalen

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der Bautechnik, Wasserversorgung, Mathematik und Poesie heute noch im Iran existieren. Auch werden viele maerchenhafte Geschichten erzählt, die mehr oder weniger auf seine Weisheit zurückzuführen sind.


Selbstverständlich ist dieses Märchen von der Wahrheit sehr weit entfernt, trotzdem aber im Grundgedanken verwandt. Es ist klar, dass die geschnittenen Gipsplatten, die an den Wänden in etwa 15 - 20 cm Abstand in Kassettenform angebracht sind, nichts anderes als Helmholtz-Resonatoren sind. (Abb. 3). Wir kennen sie heute gut als Tiefenabsorber, deren Grundfrequenz nach der Gleichung:

\[ f_0 = \frac{c}{2\pi} \sqrt{\frac{\pi R^2}{L + (\frac{\pi}{4}) R}} \]  
[Hz]
berechnet werden kann. Es scheint, dass dieses Zimmer dem Koenig wegen seines guten Klanges gefallen hat, weil damit aeusserst steroender Nachhall der tiefen Frequenzen beseitigt war. Da nach der Berechnungsformel fuer die Nachhallzeit:

\[ T = \frac{0.16 \cdot V}{\sum a \cdot S} \text{ [sec]} \]

ist, wird bei einem hohen Absorptions-Koeffizienten \( \alpha \) der Helmholtz-Resonatoren der Wert "T" fuer die tiefen Frequenzen klein. Auch wegen der anderen schallschluckenden Materialien, die damals in diesem Zimmer waren, Teppiche, Vorhaenge, Rucklehnen usw. kann man sich vorstellen, dass hier eine gute Akustik fuer die 5 - 6 Hofmusikanten mit ihren alt-persischen Musikinstrumenten herrschte.

Wir beabsichtigen, gelegentlich eine wissenschaftliche Untersuchung dieser Raeume durchzufuehren, um die Gestaltung mit den derzeitigen Mitteln zu erforschen und um den Nachhallverlauf, die Diffusitaet und den Deutlichkeitsgrad zu ermitteln. Hierbei soll auch eine Probeaufnahme gemacht werden. Das Ergebnis wird den Interessenten durch Veroeffentlichen, evtl. in den Kongresstagen, mitgeteilt werden.