EIGHTH INTERNATIONAL CONGRESS
ON ACOUSTICS, LONDON 1974

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FOREWORD

This 8th International Congress on Acoustics is sponsored by the International Union of Pure and Applied Physics of Unesco (IUPAP) and jointly organised by the Institute of Acoustics and the Institute of Physics under the auspices of the International Commission on Acoustics of IUPAP.

The technical programme of the Congress attracted over 700 invited and contributed papers. The progress papers are in a separate volume entitled Invited Papers while the contributed papers are published in two volumes, which are page numbered consecutively as for a single volume. For this Congress a change has been made in the format of presentation of these contributions, necessitated by the rapid increase in the number of papers being submitted to the International Congresses on Acoustics. One page only was allocated to each author who was also restricted to the presentation of one paper only.

The contributed papers are published as submitted. They are grouped under the major headings corresponding to the Technical sessions.

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LASER-BEAM PROBING OF ULTRASONIC FIELDS IN TRANSPARENT CRYSTALS

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INTRODUCTION

Optical beam probing techniques have proven useful in the investigation of ultrasonic fields in transparent crystals. As shown in our previous publications (1 & 2) these techniques allowed visualization of ultrasonic beam patterns and led to the successful application of a simple method to account for losses due to diffraction beam spreading of quasi-longitudinal and quasi-transverse acoustic modes in crystals.

EXPERIMENTAL AND THEORETICAL ANALYSIS

In the present work optical and ultrasonic techniques were utilized in the study of ultrasonic wave reflection from stress-free boundaries in quartz and sodium chloride. Since calculations based on infinite plane wave propagation indicate that the nature of the reflected and mode converted waves at an interface depend on the crystallographic orientation of the reflecting surface, these modes were identified and their intensities measured for a variety of reflecting surfaces. Since the direction of propagation of these reflected waves varies with the angle of incidence, the extent of the losses due to beam spreading also varies accordingly. Diffraction corrections were applied to each of these acoustic modes and the partition of energy was evaluated among the modes participating in the reflection process. The significance of our findings to the use of ultrasonic methods for industrial non-destructive testing is discussed.

REFERENCES

Es sind die Ergebnisse unserer Messungen der Diffraktion monochromatischer Neutronen \((1,05 - 1,8 \AA)\) in den Einfallssebenen \([111]\) und \([333]\) der schwingenden und nichtschwingenden Ge- und Si-Monokristalle angeführt, die in Abhängigkeit von der mit Hilfe eines Magnetostriktionsresonators erfolgten Erregung auf der Frequenz von 43 kHz realisiert wurden. Die erzielten Meßergebnisse wiesen nicht nur auf eine bedeutende Intensitätsverstärkung der Diffraktionsneutronen, sondern auch auf die Zeitmodulation des Neutronenbündels hin. (1), (2).


In Abb. 1 sind einige der in Abhängigkeit von der Schwingungsmplitude erzielten Meßergebnisse für \(R\) in verschiedenen Einfallssebenen und bei verschiedenen Wellenlängen angeführt:

- \(1,05 \AA - 1 [111]\)
- \(2 [333]\)
- \(1,4 \AA - 3 [111]\)
- \(4 [333]\)
- \(1,8 \AA - 5 [111]\)
- \(6 [333]\)

Abb. 1: Abhängigkeit von \(R\) von der Amplitude des in Schwingungen erregten Untersuchungsmusters.

Wandlerspannung: \(0 \div 1 V\), f = 43 kHz

In Übereinstimmung mit der Theorie wächst \(R\) für die Neutronen von längeren Wellenlängen an. Mit Hilfe des Zeitanalyzers wurden dann die von einem schwingenden Monokristall reflektierten Neutronen gemessen, wobei sich erwiesen hat, daß die Intensität der zurückgestrahlten Neutronen die periodische Zeitfunktion ist. (Zeitkanalbreite: 0,5 μs).

Literatur:
(2) Chalupa B., Taraba O. et al., Diffrakcija neutronov na korebljuščich-sja rešetkach monokryst. Ge a Si, Sammelband der Beiträge vom Symposium über die Neutronenspektroskopie, UdSSR, Dubno 1970 (Praha 1972)
INTRODUCTION

From light intensity measurements of second diffracted orders (left and right) the evaluation was done (1) of the complex coupling coefficient between fundamental and second harmonic ASW. Presently, dispersion effects are taken into consideration (2) and difference is evidenced which exists between light reflected and light transmitted experiments.

EXPERIMENTAL

In Figs. 1 and 2 light intensity measurements are reported vs. length of interaction, relative to light reflected and light transmitted experiments performed at 64 MHz on LiNbO$_3$ substrates.

Difference is due to the effect of the elasto-optical interaction that in the latter case is responsible both for a different efficiency and a phase shift in the emerging light wavefronts (3).

REFERENCES

EFFETS ÉLECTRIQUES CONSECUITIFS À L'ÉVOLUTION RAPIDE D'UNE BULLE DE GAZ OU DE VAPEUR DANS UN LIQUEIDE

Chincholle L   Université de Paris, Fontenay-Aux-Roses,
Goby F   France

Une bulle de gaz ou de vapeur, soumise à un gradient de pression important acquis, lors de son implosion, une grande vitesse de translation (1).

Si l'on admet la présence de charges électriques à l'intérieur ou à l'extérieur de la bulle, il devrait être possible de détecter des courants.

Dans le cas des liquides polaires, cette charge apparente peut provenir (en application de la théorie de la double couche de Helmholtz) de la déformation du nuage électronique entourant la bulle (2). Pour les liquides diélectriques il peut y avoir électrisation par frottement.

Afin de mesurer ces courants nous avons utilisé une sonde plongeant dans un liquide soumis à un flux d'ultrasons. L'étude a été faite dans diverses conditions expérimentales (influence du liquide, du gaz dissous, de la pression, de la température).

Parmi les résultats obtenus nous signalerons :

1. des variations du potentiel en fonction de la puissance ultrasonore.
2. un phénomène de relaxation électrique suivant qu'un est en cavitation de gaz ou en cavitation de vapeur.
3. un changement de signe du courant suivant la position de la sonde.

Références :
(2) Degrois : Revue Acoustique n° 2 1969.
(3) Frenkel : Zurnal Fiz-Khimii USSR n° 3 1940.
EIGHTH INTERNATIONAL CONGRESS
ON ACOUSTICS, LONDON 1974

OPTICAL OBSERVATION OF ACOUSTIC WAVES IN SMECTIC LIQUID CRYSTALS

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Bertolotti M
Scudieri F  Sette D  Śliwiński A

INTRODUCTION

The optical changes induced by an acoustic field in p-cyano benzilidene-p-octyl-oxy-aniline (CB00A) a liquid crystal which shows its smectic - A phase between 73°C and 84°C, are described.

EXPERIMENTAL

The sample is prepared in the planar texture between two optical glasses, one of which is driven by a piezo-electric transducer, at frequencies lower than 1 MHz. The optical examination is made with the arrangement shown in Fig.1. In the conoscopic vision between crossed polarizers, when no acoustic field is applied, very sharp crosses are visible, which disappear when the field is applied. The relaxation of the material when the field is switched off is very long. The spectrum of the scattered light is also observed with an heterodyne technique. Experimental set-up

The results are discussed with reference to a mechanical instability (undulation of layers) introduced by negative pressure across the layers, and proposed in refs. (1) and (2).

(*) on leave from Institute of Physics University of Gdansk, Poland

REFERENCES

(1) H.Delaye, R.Ribotta and G.Durand-Phys.Letters 44A, 139 (1973)

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DIFFERENT SOURCES OF FORMALDEHYDE FORMATION IN THE WATER SOLUTION IN THE FIELD OF THE ULTRASONIC WAVES

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INTRODUCTION

Our task was to study different sources and mechanisms of formaldehyde formation in water solutions in the field of ultrasonic waves.

EXPERIMENTAL

The sonication of water solutions was done in hermetically sealed glass and metallic vessels. We used piezoguartz crystals with frequency 560, 750 and 850 kHz. Intensity 3-10 w/cm².

Fig. 1 represents the curves of formaldehyde formation, the water being saturated by carbon monoxide. Reaction yield increased the water being saturated with carbon monoxide and hydrogen (1). Formaldehyde is also formed in the solution of methane in the water in the absence of other gases, the yield being 2.9·10⁻⁵ molecules per 100 eV (2). Formation of formaldehyde takes place in the water solutions of compound organic substances (acid, aminoacids, carbohydrates, proteins). The yield from 0.6 to 20·10⁻⁵ molecules per 100 eV increases in the presence of argon (but not helium and oxygen) (3) at a pressure not more than 1-3 atmospheres.

PRINCIPAL RESULTS

Formaldehyde is formed both by decomposition of compound substances and by oxidation or reduction of monocarbonic molecules - methane and carbon monoxide. The water is also involved in the chemical reactions. The rate of these reactions depends on the nature of the gas present and the static pressure.

REFERENCES

THE VELOCITY OF SOUND IN METASTABLE LIQUIDS

Apfel R
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Most reported measurements of properties of the liquid state have been confined to the pressure-temperature states for which the given liquid is stable. If sufficient care is taken, however, one can maintain a liquid in a metastable state (that is, superheated, supercooled, or under tension) while a given liquid property is measured.

As an example, consider water. At atmospheric pressure water has been supercooled from 0 C to -41 C (1) and possibly even below, and has been superheated from 100 C to 279 C (2) and possibly even higher. At room temperature water has been subjected to acoustic pressures of over 200 atmospheres (3) and static tensile stresses of 277 atmospheres (4). In these enormous metastable ranges very few of water's properties have been measured.

We have, therefore, set out to develop techniques of measuring properties of metastable liquids (5). Currently, we are adapting standard pulse and standing wave techniques to the measurement of the speed of sound in metastable liquids.

Our main problem is avoiding the nucleation of the stable phase from the metastable phase. In superheat experiments the cell that holds the liquid must be of one seamless material (e.g. glass or fused quartz) to avoid bubble nucleation sites. Also, prior to the experiment the liquid is subjected to hydraulic pressurization (500-1000 atmospheres) for a short period of time to assure that all solid surfaces in contact with the liquid sample are thoroughly wetted and that all microscopic pockets of gas are forced into solution.

Measurements of the sound speed at atmospheric pressure are then performed as the temperature is increased until the sample explosively vaporizes. A similar procedure can be used for measurements of the sound speed in a supercooled liquid, until the sample solidifies.

In our first experiments we used an optical technique (6) for visualizing acoustic standing waves and measuring the sound speed in water. In the frequency range 0.5 to 1.5 MHz and at atmospheric pressure the sound speed was measured to 140 C and shows an expected monotonic decrease with temperature. The main purpose of these initial measurements was to demonstrate that the liquid could indeed be maintained in its metastable state for a sufficient time (often over an hour) to carry out our experiments. These initial results have encouraged us to pursue our studies of the metastable state further.

[Work supported by U.S. Office of Naval Research.]

REFERENCES

EIGHTH INTERNATIONAL CONGRESS ON ACOUSTICS, LONDON 1974

ULTRASONIC LIGHT DIFFRACTION AND THE 3RD HARMONIC OF LIGHT GENERATION IN LIQUIDS

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Some calculations of the intensity distribution in ultrasonic light diffraction for the case of the intense laser beam generating a 3rd harmonic in liquids have been already reported (1,2) and lastly extended (3). Further attempts have been made to prove theoretical predictions.

In our experiments the special ultrasonic-optical cell has been developed with parallel movable walls (windows) allowing to adjust the interaction distance $l$ (Fig. 1) for the proper matching in refractive indices of the fundamental and harmonic of light beam what is closely connected with the coherence length ($2,4$):

$$l_s = \frac{\lambda \omega}{6(n_{3\omega} - n_\omega)}$$

Different organic optically nonlinear liquids with the proper dependence of absorption Fig. 1. Scheme of the ultrasonic-optical cell examined.

In this report some experimental results are compared with the theoretical formula for ultrasonically diffracted intense laser beam derived by J. Józefowska (3).

REFERENCES

ULTRASONIC ABSORPTION IN UNASSOCIATED LIQUIDS

Kor S K
Allahabad University, India

INTRODUCTION

Ultrasound absorption studies have been made in some unassociated liquids as a function of temperature and pressure. The results are encouraging.

EXPERIMENTAL

The temperature dependence of ultrasound absorption measurements have been made using a pulse technique at 30 MHz.

THEORETICAL ANALYSIS

The mechanisms responsible for the contribution to the total ultrasound absorption are shear viscosity, compressional or structural and vibrational relaxation

\[
\left(\frac{\alpha}{T^2}\right)_{\text{Total}} = \left(\frac{\alpha}{T^2}\right)_{\text{Shear}} + \left(\frac{\alpha}{T^2}\right)_{\text{Comp}} + \left(\frac{\alpha}{T^2}\right)_{\text{Vib}} \tag{1}
\]

\[
= \frac{2\pi^2}{c^2} \left(\frac{4}{3} \eta_s + \eta_v\right) \tag{2}
\]

\(\eta_v\) is the volume viscosity which accounts for all the excess absorption over that of the classical value.

Considering the existence of holes in liquid lattices (1), the expressions for the pressure dependence of shear and compressional viscosities are

\[
\frac{\eta_{sp}}{\eta_{so}} = \exp \left[ \frac{pV_p}{kT} \right] \tag{3}
\]

\[
\frac{\eta_{cp}}{\eta_{co}} = \exp \left[ \frac{(1 - f)pv_e}{kT} \right] \tag{4}
\]

Using equations (1) to (4) the various contributions to the total ultrasound absorption as a function of pressure are evaluated. With some modifications the temperature dependence of various contributions to the total ultrasound absorption are evaluated.

The molecular diameter evaluated from the above studies are as expected, showing the validity of the calculations.

REFERENCES

EIGHTH INTERNATIONAL CONGRESS
ON ACOUSTICS, LONDON 1974

ACOUSTIC TECHNIQUES FOR STUDYING THE VISCOSITY OF
FLUIDS IN THE CRITICAL REGION

Carome E F
Fisch M R

John Carroll University, Cleveland, Ohio, USA

INTRODUCTION

We are continuing efforts to determine the shear viscosity $\eta_s$ of
simple fluids in the vicinity of liquid-vapor critical points using
acoustic resonator techniques (1-2). We have used a tuning fork made
from a flat $x$-cut quartz plate 0.1 cm thick, 1.27 cm wide, and 5.08 cm
long, and are able to examine directly its displacement versus frequen-
cy response. Typically the fundamental resonance frequency of such
an element is 2 kHz and the half power bandwidth $\delta f$ ranges from .04 Hz
in vacuum to 60 Hz in highly viscous fluids.

INITIAL RESULTS

We have determined the variations of the damping of such resonators,
submerged in simple fluids contained in a horizontal flat cavity of
0.3 cm total vertical height, as the fixed mass of fluid trapped in
the cavity is brought isochorically into the critical region.

Measurements have been made along various isochores of xenon,
carbon dioxide, and sulfur hexafluoride in the gaseous phase to .003°C
of the coexistence curve at densities in the range 0.9 to 1.1 $\rho_f$.
Within 0.3°C of the coexistence curve we have observed substantial
increases in the damping, and have concluded that $\eta_s$ may increase by
a factor of ten in the critical region. Since this differs substan-
tially from the results of other workers (3-4) we are conducting a
series of experiments to investigate further the effects we have noted.

CURRENT WORK

We are repeating our measurements using forks and cavities of differ-
ent configuration, and also are attempting to determine if finite
amplitude effects might be present. In addition we are using a win-
dowed cell and are optically probing the fluid to determine if macro-
scopical flow processes may occur. Finally, we are conducting experi-
ments using an electromagnetically driven wire resonator.

We anticipate that the results of these present studies will shed
some light on the possible existence of a shear viscosity anomaly in
the critical region of simple fluids. We plan to report more fully on
our work at the time of delivery of this paper.

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(1) E. F. Carome and R. P. Moeller, Proc., IEEE Ultrasonics Symposium,
Boston (1972) 109; (2) W. P. Mason, Physical Acoustics and the
Properties of Solids (Van Nostrand, N.Y. 1958) 107; (3) J. Kestin et
(1971) 27, 1182.
ULTRASONIC VELOCITY IN ALCOHOL MIXTURES

Ștețiu C.  Babeș Bolyai University, Cluj, Romania

INTRODUCTION

Temperature and concentration dependence of ultrasonic velocity in mixtures of some primary alcohols were measured in order to evaluate the heat of mixing.

In previous papers (1) studies were made of the pure primary alcohols and their mixtures over the full concentration range and within the temperature range 0-90°C.

EXPERIMENTAL

The ultrasonic interferometer in which is mounted an x-cut quartz cristal driven at 1 MHz has been described (2). The following mixtures of two alcohols were investigated: methanol in ethanol, l-propanol, l-butanol, l-pentanol, l-heptanol, l-octanol and l-decanol. Measurements of sound velocities were made within the temperature range 0-90°C. Densities were determined within the range 15-90°C by the pycnometer method. Viscosities were measured within 0-90°C using an Ostwald method.

DISCUSSION

All the substances were hydrogen-bonded and all were miscible with their partners at all concentrations and all temperatures.

Kudriavtsev (3) developed the following expression, based in part on thermodynamic reasoning, for the velocity v of sound in a non-ideal binary mixture

\[ v^2 = c \frac{M_1}{M} v_1^2 + (1-c) \frac{M_2}{M} v_2^2 + nmXH \]  

(1)

where \(v_1\) and \(v_2\) are the velocities of sound in pure components, \(M\) is the average molecular weight of the mixture \(M_1\) and \(M_2\) the molecular weights of the components, \(c\) the mole fraction of the first component, \(n\) and \(m\) constants, \(X = C_p/C_v\), and \(H\) is the heat of mixing.

Using eqn. (1) values of \(H\) were calculated and compared with the values obtained from other methods.

REFERENCES

CONTRIBUTIONS STRUCTURELLES SUR LA VITESSE DES ULTRASONS DANS LES LIQUIDES ORGANIQUES NON-IDEAUX

Ausländer D, Babeș-Bolyai University, Cluj, Romania
Macovei I

INTRODUCTION

Les interactions moléculaires dans la série des alcools par l'intermédiaire des liaisons d'hydrogène, respectivement dans les solutions à componante non-polaire furent investiguées au moyen de plusieurs méthodes expérimentales basées sur diverses représentations structurales (1).

PARTIE EXPERIMENTALE

Les effets des modifications structurelles sous l'action de l'agitation thermique furent étudiées à l'aide de mesurations de la vitesse de propagation des ultrasons à diverses densités et températures, afin de déterminer la compressibilité adiabatique.

PARTIE THEORIQUE

Dans un domaine de température assez éloigné des valeurs critiques, en négligeant l'énergie cinétique des molécules, la dépendance de la vitesse des ultrasons de la racine carée de l'énergie met en évidence une sensibilité de la compressibilité adiabatique plus élevée que celle de la vitesse, en ce qui concerne les déviations par rapport à la linéarité caractéristique aux liquides idéaux.

En considérant les valeurs obtenues en tant que sommes des termes vibrationnels et structuraux, les variations des gradients dv/dT et dβ/dT permettent d'estimer les fractions -OH libres dans des conditions d'équilibre thermique.

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EINLEITUNG

Strukturrelaxation in Flüssigkeiten mit starken molekular-
ren Wechselwirkungen führt zu einer Ultraschallabsorption,
die erheblich über dem aus Schallgeschwindigkeit, Dichte und
Schubviskosität berechneten klassischen Wert liegt.

EXPERIMENTELLES

Die Temperaturabhängigkeit der Ultraschallabsorption in
Methanol, flüssigem Ammoniak und Wasser wurde mit Hilfe
des Debye-Sears- und des Resonatorverfahrens im Frequenz-
bereich 1 - 100 MHz gemessen.

THEORETISCHE ANALYSE

Die Strukturabsorption kann formal durch eine Volumviskosi-
tät $n_v$ in die hydrodynamischen Gleichungen eingeführt wer-
den. Dann ist

$$\frac{2a_s}{f^2} = \frac{4\pi^2 n_v}{\rho_0}$$

wobei

$$n_v(\omega) = \frac{1}{9VvT} \sum_{a,b} \int_0^\infty dt \ e^{-i\omega t} \langle j^{aa}(0) j^{bb}(t) \rangle$$

und

$$j^{aa} = \frac{d}{dt} \sum_j p_j r_j^a - \left[ pV + \left( \frac{\partial p}{\partial E} \right)_V (E-\langle E \rangle) + \left( \frac{\partial p}{\partial N} \right)_V (N-\langle N \rangle) \right]$$

Hierbei sind $p_j$ und $r_j$ die Impuls- und Ortskoordinaten des
jten Moleküls. $j^{aa}$ stellt den Impulsfluß bei einer isotro-
en Kompression dar. Der Mittelwert $\langle ... \rangle$ bezieht sich auf
ein großkanonisches Ensemble im thermischen Gleichgewicht.

Die Hamiltonfunktion des Systems kann zumindest for-
mal in "Translations"-Moden und in "Struktur"-Moden sepa-
rriert werden, wobei die Wechselwirkung zwischen beiden rela-
тив gering ist. Damit ergibt sich ein Ausdruck für die
Strukturabsorption, der die Strukturwärme enthält. Mit Hil-
fe des n-Zustandsmodells lassen sich die experimentellen
Daten für Methanol und flüssiges Ammoniak mit $n = 2$ und
für Wasser mit $n = 3$ deuten.

Durch die Annahme weiterer Zustände läßt sich auch
der Einfluß einfacher Ionen auf die Struktur der Flüssig-
keiten quantitativ deuten.
LA DÉTERMINATION ULTRASONIQUE DES NOMBRES DE SOLVATATION
DANS QUELQUES SOLUTIONS ALCOOLIQUES

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INTRODUCTION

La puissante compression à laquelle sont exposés les molécules de la couche de solvatation, sous l'effet du champ électrique de l'ion, modifie d'une manière correspondante la compressibilité de la solution respective; la connaissance de ce paramètre offre ainsi la possibilité de calculer le nombre d'hydratation.

PARTIE EXPERIMENTALE

On a effectué des mesurages de la vitesse de propagation de l'ultrason et de densité, dans des solutions alcooliques de certains halogènes des métaux alcalins et alcalino-terreux, les valeurs étant utilisées à calculer la
\[ \beta_{ad} = \frac{1}{\rho v^2} \]
dans certains intervalles de concentrations.

PARTIE THEORIQUE

Le nombre de solvatation a été calculé à l'aide de la relation:

\[ z = \left( \frac{\rho - \rho_0 (\beta/\beta_0)}{c_0} \right) - c \frac{M}{M_0} \]

En dehors de la dépendance de z de la température et de la concentration, l'on constate des effets différenciés des ions dans le sens de l'organisation, respectivement de la désorganisation structurelle (1); dans le premier sens ce sont les petits ions qui agissent, dans le deuxième sens les grands ions. La valeur du nombre de solvatation résulte de l'équilibre des interactions spécifiques des anions et cations, à de grandes concentrations prédominant le type ion - ion.

BIBLIOGRAPHIE

Einfrierungseffekt in Konzentrationsgefäßen


Einfrierungseffekt in homogenen Flüssigkeiten


Die Experimente werden so ausgeführt, dass man Wärmegänge in einer durch akustische Strömungen aufgebauten Wärmequelle erzeugt. Da die akustischen Strömungen in ursächlichem Zusammenhang mit dem Schallstrahlungsdruck stehen, lässt sich der Effekt sowohl vor der Schallquelle Q wie vor dem Reflektor R beobachten. Es lässt sich nun durch Verwendung einer höher viskosen Flüssigkeit (Polystyrol-Lösung) erreichen, dass nach dem Ausschalten der Schallquelle in der Mittelachse eines etwa 4 cm breiten Ultraschallbündels eine schmale nur etwa 0,4 cm breite und 10 cm lange Säule einer kurzzeitig eingefrorenen Schallwelle sichtbar wird. Obwohl der Durchmesser dieser Säule nur zwei- bis dreimal so gross wie die Wellenlänge ist, tritt kein Beugungsphänomen auf (siehe Skizze)

(2)W.Schaaffs und L.Haun, Acustica (1968) 20 548 ; (1972) 27 166 ; (1973) 28 303; (3)W.Schaaffs, Acustica 28 171
LA COMPRESSIBILITÉ MOLALE APARENTE DES SOLUTIONS AQUEUSES DES HALOGENURES

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INTRODUCTION

Les propriétés molales apparentes ont l'avantage d'être déterminées par des mesurages des mêmes grandeurs caractéristiques aux solutions et aux solvants purs, permettant une comparaison directe avec une solution idéale. Les variations mises en évidence mesurent les interactions existentes dans les solutions réelles.

RESULTATS

Les valeurs expérimentales de la vitesse de l'ultrason ont permis de calculer la compressibilité molale apparente, $\Phi K_2$, pour les solutions aqueuses des fluorures et des chlorures, à différentes températures et concentrations.

On a mis en évidence la variation linéaire de $\Phi K_2$ avec la racine carrée de la concentration molaire, en accord avec les conclusions de la théorie Debye-Hückel. Avec l'élévation de la température la pente de ces droites décroît et la compressibilité molale apparente à dilution infinie, pour chaque solution, croît. À la même température ces valeurs sont plus grandes pour le cation de rayon plus grand, ce qu'indique un nombre plus grand des molécules d'eau attachées au chaque ion. On a calculé aussi ce nombre d'hydratation, avec la présomption que le volume molaire des molécules d'eau contenues dans la première couche d'hydratation est le même que celui des molécules de l'eau pure.1-2. Les valeurs obtenues sont caractéristiques aux ions existants en solution.

BIBLIOGRAPHIE

ULTRASONIC ABSORPTION OF ISOMERIC ALCOHOLS IN NON-POLAR SOLVENTS

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INTRODUCTION

The plot of the ultrasonic absorption $\alpha/f^2$ ($\alpha =$ absorption coefficient $f =$ frequency) of solutions of alcohols in non polar solvents versus the alcohol mole fraction $x$ goes through a maximum at $x<0.1$. The maximum has been attributed to the self association of alcohol molecules through H-bond (1). In this paper we report the absorption of a series of isomeric octanols in cyclohexane. This study was undertaken to follow the changes of the curves $\alpha/f^2$ vs $x$ brought about by an increasing steric hindrance about the hydroxylic (OH) group.

EXPERIMENTAL

The main results are shown on figure 1 where the quantity $\Delta \alpha/f^2$ represents the difference between the absorption of equimolecular solutions of a given isomeric octanol and of 2,2,3-trimethyl 3 pentanol. This last compound was used as reference because it is characterized by a curve $\alpha/f^2$ vs $x$ which is coincident with that of 2,2,4 trimethylypentane. Absorption maxima are found only for alcohols where the OH group is relatively unhindered. When two or more methyl groups are close to the OH group the curves show a levelling off or a continuous increase.

DISCUSSION

Ultrasonic absorption appears to be more sensitive than dielectric absorption (2) to H-bonding in alcohol solutions.

Attempts have been made to explain the maxima in terms of various association models for alcohol molecules. The model monomer $\leftrightarrow$ linear tetramer $\leftrightarrow$ cyclic tetramer (3) fits the results if the excess absorption is assumed to be associated with the fast step monomer $\leftrightarrow$linear tetramer. The reaction monomer $\leftrightarrow$ trimer (4) can also explain the results.

REFERENCES

ULTRASONIC METHODS FOR DETERMINING THE EQUATION OF STATE OF LIQUIDS AND AMORPHOUS SOLIDS AT HIGH TEMPERATURES

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INTRODUCTION

Apparatus is developed to obtain PVT data for liquids and amorphous solids, at temperatures up to 1000°C and pressures up to 30,000 psi, from ultrasonic velocity (1) and time-of-flight measurements (2). Equation of state data for liquid lithium is presented.

EXPERIMENTAL

Figure 1 shows the apparatus for speed of sound measurements in liquids. To cover a wide temperature range, it is necessary to use a gaseous pressure system and an internally heated pressure vessel. Measurements were made in liquid lithium and the speed of sound was found to be c(cm/sec) = 470100 - 66.7 T(°C) + 0.586 p(psi) over the range 180 - 500°C and 0 - 12000 psi. To obtain PVT data for glasses, the acoustic dilatometer, described in a previous paper (2), is redesigned for operation over a wider temperature range.

THEORETICAL ANALYSIS

The analysis of the lithium data has so far been limited to one aspect, namely the increase in pressure at constant volume, since this can be directly related to the effective ion-ion interaction. Experimental values are compared with calculations made using the perturbation theory of liquids (3) and previously published (4) values for the ion-ion interaction in lithium. Good agreement between theory and experiment is obtained, as shown in Fig. 2.

REFERENCES

In last years ultrasonic propagation has been studied extensively in the critical region of both one- and two- component fluid systems. As a consequence of critical fluctuations, a large excess absorption (bigger in simple fluids) and velocity dispersion are observed in such systems. However, recent theoretical predictions (1) show the possibility of a smaller and smaller influence of critical fluctuations as the number of components increases. In order to test such predictions, we performed some ultrasonic measurements near the plait (critical) point of the ternary system Water-Benzene-Ethanol at 25°C and 1 atm. Absorption coefficient (κ) and sound velocity (c) were measured at different frequencies both as a function of temperature at fixed (critical) composition and as a function of composition at 25°C. In both cases, on approaching the plait point, a very small increase of κ/f² was observed. In addition this quantity resulted almost independent on frequency (15-55 MHz). As sound velocity is concerned, no dispersion was observed between 5 and 55 MHz, within the limits of accuracy of our experimental method (Δc/c ≈ 0.5%). This behavior, as we said, is rather different from the observed one in simple fluids or binary mixtures. They seem to confirm the theoretical conclusions obtained by one of us (M.M.) on the basis of a generalization of the existing theories on sound propagation in one- and two- component systems (1). Briefly, the calculation in a ternary system is based on the generalization of the relation between sound velocity and adiabatic compressibility at constant composition

\[ c^{-2} = \kappa_{S,x_1,x_2} \]

c Considering complex and frequency dependent quantities.
In this way one takes into account the effect of the slow decaying critical composition fluctuations which lead to expressions for the sound absorption and velocity qualitatively in agreement with experiments.

REFERENCE

ULTRASONIC RELAXATION STUDIES ON POLYPEPTIDE SYSTEMS

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PROBLEM

Kinetic investigations of helix-coil transitions in poly-peptide solutions have been made in systems with the transition affected by a pH or solvent composition change (1, 2). We have studied helix-coil transitions induced only by temperature variation in the same solvent, e.g. poly-β-benzyl-L-aspartate (PBLA) in m-cresol (3). Dielectric data (frequency range 300 Hz to 3 MHz) for this system had been published by A. Wada (4,5).

EXPERIMENTAL

Ultrasoundic excess absorption has been measured by a resonator and a pulse technique (6) in the frequency range from 1 to 100 MHz between 25 and 80°C for PBLA (Miles-Yeda Ltd.; mol. weight 25000) at 1 to $5 \cdot 10^{-3}$ M in m-cresol with the solvent as the reference system; the transition has been verified by optical methods and viscosity measurements.

RESULTS

The ultrasonic absorption spectra at various temperatures clearly indicate the helix-coil transition of the poly-peptide with a transition temperature $T_c$ of ca. 50°C, in coincidence with our optical data. Between 2 and 10 MHz relaxation processes are observed below and above $T_c$; near 50°C the relaxational absorption appears to be drastically reduced. The temperature dependence of the "background" absorption correlates with the fraction of helicity. The relevance of the observed relaxational processes for the dynamics of helix-coil transitions will be discussed.

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(2) A. D. Barksdale and J. E. Stuehr, J. Am. Chem. Soc. (1972) 94 3334  
(3) Y. Hayashi et al., Biopolymers (1969) 8 403  
(5) A. Wada et al., Biopolymers (1972) 11 587  
L'eau présente un maximum de masse volumique au voisinage de 4°C et cet effet pourrait avoir une influence sur la vitesse et l'absorption ultrasonore dans ce liquide. Plusieurs auteurs (1,2) ont examiné ce problème mais aucune anomalie n'a été mise en évidence.

Nous avons effectué des mesures de vitesse de propagation et d'absorption des ondes ultrasonores dans l'eau à des fréquences comprises entre 5 et 15 MHz et dans un domaine de température de 0 à 10°C ; les résultats obtenus montrent une anomalie au voisinage de 4°C.

Plusieurs méthodes sont utilisées pour les mesures de vitesse : une méthode interfréquentielle en ondes entremêlées et deux méthodes d'impulsions (superposition d'impulsions et bouclage). Les résultats expérimentaux sont concordants et la figure 1 rend compte de l'effet observé. La variation ΔV/ΔT de la vitesse de propagation pour des écarts de 1°C montre un minimum aux environs de 4°C. Ceci se traduit par un point d'inflexion sur la courbe v = f(T).

Les mesures d'absorption ultrasonore ont été effectuées dans la même gamme de température par une méthode d'impulsions qui consiste à comparer l'amplitude d'un écho pour des trajets acoustiques de longueurs différentes. La figure 2 rend compte des résultats expérimentaux et bien que nos valeurs coïncident avec celles d'autres auteurs (3,4) elles montrent que la variation de α/ f² (rapport du coefficient d'absorption au carré de la fréquence) en fonction de la température apparaît plus importante autour de 4°C que dans le reste du domaine de température.

L'effet mis en évidence est légèrement supérieur à la précision des mesures. Nous nous employons à améliorer cette précision. Cet effet met en évidence une variation du coefficient de compressibilité de l'eau au voisinage de 4°C, car l'anomalie observée ne peut être expliquée par la seule anomalie de la masse volumique.

Bibliographie.
EIGHTH INTERNATIONAL CONGRESS ON ACOUSTICS, LONDON 1974

CONTRIBUTION TO THE DETERMINATION OF PHASE AND GROUP PROPAGATION VELOCITIES IN DISPERSIVE MEDIA

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The problem of determining the group velocities on the base of the measured phase velocities sometimes needs a more practical approach, than that given by the classical eq. /1/

The important problem of determining the phase velocities in dispersive media from the pulse measurements of group velocities has, until now, apparently not been raised.

The known relation between the phase velocity \( c_\varphi = \omega / \beta \) and the group velocity \( c_g = \partial \omega / \partial \beta \):

\[ c_g = c_\varphi - \lambda \frac{dc_\varphi}{d\lambda} \] /1/

when rewritten in the form:

\[ \frac{c_\varphi}{c_g} = 1 - \frac{fdc_\varphi}{c_\varphi df} \] /2/

allows for an easy graphic calculation of \( c_g \) from the \( c_\varphi / f_c \):

Rearranging and integrating eq. /2/ we finally obtain the expression for the phase velocity \( c_\varphi \) in function of the group velocity \( c_g / f_c \):

\[ \frac{c_\varphi / f_c}{f_c} = f_0 \int \frac{df}{c_g} \] /3/

which also allows for an obvious graphic calculation, requiring, however, the knowledge of \( c_g \) in the frequency interval from 0 to \( f_0 \). On the base of eq. /3/ it may be shown that, determining the velocity at the constant path 1 by varying the frequency and by measuring its interval, corresponding to the phase change \( 2\pi \), the known formula:

\[ c = \Delta f / 2\pi \] /c/

yields the value near to the average group velocity in the interval \( \Delta f \), although the observation of the phase shift is essential for the measurement /1/.

PRESSURE JUMP - VOLUME RELAXATION OF B₂O₃ GLASS IN THE GLASS TRANSITION REGION USING AN ACOUSTIC DILATOMETER

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INTRODUCTION

A recently developed ultrasonic dilatometer (1) has been modified and used in the direct measurement of pressure induced, time-dependent volume changes of B₂O₃ glass in the glass transition temperature region (250 - 350°C). These time-dependent volume changes are caused by structural changes in the glass which are much the same as the structural reorientations observed in common liquids at very high frequencies (2).

EXPERIMENTAL

Following a sudden (one second) pressure change, the volume of a glass sample is determined as a function of time over the time range 30 to 30,000 seconds. From the measured volume changes, the instantaneous and equilibrium compressibilities (βᵢ and βₑ) are evaluated, as well as parameters related to the relaxation spectrum (Fig. 1).

ANALYSIS

Time-temperature superposition is used to reduce the data collected at ten temperatures to a single curve. The data is treated in terms of a spectrum of relaxation times, and also using a memory function approach. The results are compared with previous temperature-jump and high temperature ultrasonic data. In addition, the measured values of thermodynamic parameters (compressibility, expansivity) are compared with model predictions.

REFERENCES

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ULTRASONIC AND BRILLOUIN SCATTERING STUDIES OF IONIC HYDRATE MELTS

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INTRODUCTION

Ultrasonic relaxation has been reported in ionic hydrate melts (1-4) but the data have generally been too limited to establish the distribution of relaxation times. Flow in such melts involves the disruption of relatively strong ion-ion and ion-water bonds.

EXPERIMENTAL

Ultrasonic absorption \( a \) has been determined as a function of frequency \( f \) (up to 400 MHz), water content, and temperature for Zn(NO\(_3\))\(_2\).\(x\)H\(_2\)O, ZnCl\(_2\).\(x\)H\(_2\)O, and Ca(NO\(_3\))\(_2\).\(x\)H\(_2\)O, using pulsed, send-receive apparatus. Hypersonic velocity and absorption have been measured using Brillouin scattering at 514.5\(Å \) with an argon-ion laser (single mode) and a piezoelectric scanning Fabry-Perot interferometer. Depolarized Rayleigh scattering also has been examined.

RESULTS

All three hydrate melt systems exhibit a distribution of relaxation frequencies rather than a single relaxation. The high frequency limiting values \( a/f^2 \) indicate that both shear and volume viscosities have relaxed. The deuterated ZnCl\(_2\).\(4\)D\(_2\)O and Ca(NO\(_3\))\(_2\).\(4\)D\(_2\)O have the same relaxation spectrum as the corresponding hydrate melts, although the low frequency limiting value \( a/f^2 \) is ~30% higher. The experimental points in Fig. 1 are for ZnCl\(_2\).\(4\)H\(_2\)O at 25°C. The dashed line is the single relaxation curve and the solid curve is the best fit using the Montrose-Litovitz treatment (5) involving fluctuations in an order parameter with \( \sigma^2/\Delta \Theta_0 \) = 0.5, \( f_0 = 20.5\)MHz, and \( f_p = 5.8f_0 \). For other melts \( \sigma^2/\Delta \Theta_0 \) is even lower (0.05 for Zn(NO\(_3\))\(_2\).\(3.8\)H\(_2\)O at 50°C), indicating that the predominant contribution to the structural relaxation is the diffusion of the ordering parameter. (Research supported by the U.S. Office of Naval Research, National Science Foundation, and NATO.)

REFERENCES

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APPLICATION OF WEHR'S EQUATION FOR DETERMINING THE VARIATION OF SONIC BULK MODULUS WITH PRESSURE FOR MINERAL OILS

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INTRODUCTION

Sonic bulk modulus $K_B$ is a function of density $\rho$ and propagation velocity $c$. To find the dependence of $K_B=f(p/T)$, one has to know the following functions: $\rho=f(p/T)$ and $c=f(p/T)$. The dependence of $c$ against pressure $p$ is determined by a direct measurement. The dependence of $\rho$ against $p$, basing on the measurement of $c$, has been given by Wehr's eqn./1/:

$$\rho = \rho_0 c_1^2 c_2^2 z \quad \text{where} \quad z = \frac{c_2^2}{c_1^2} \left[1 - \frac{\Delta p}{K_{\rho}} \ln \left(1 - \frac{z}{1}\right)\right]$$

Eqn. /1/ has been solved by approximation methods for a pressure jump $\Delta p = p_1 - p_2$ and ratio of specific heats $\gamma$.

CHOICE OF APPROXIMATION AND ITS EVALUATION

Wehr solved eqn./1/ and gave 1, 2 and a approximated solution. Approximation accuracy is determined by a correction factor $R_1$ which is for $p = 1000 \times 10^7$ Nm$^{-2}$ and $\gamma = 1.3$ equals $R_1 = \exp \left(\frac{\Delta x}{\gamma} - 1\right) = 1.01$. Assuming $\Delta p = 200 \times 10^7$ Nm$^{-2}$ it is obtained the first approximated solution of eqn./1/:

$$z_1 = c_2^2 c_1^2 \left[1 - \frac{\Delta p}{K_{\rho}}\right]^{-1} \left[\exp \left(\frac{\Delta p}{K_{\rho}}\right) \left(1 - \frac{\Delta p}{K_{\rho}}\right)\right]$$

The square bracketed factor is the correction factor $R_1$. For hydraulic fluid of $K_{\rho} = 15 \times 10^7$ kNm$^{-2}$ and $\gamma = 1.1; 1.25$ and 1.3 the following values of $R_1$ have been obtained accordingly: 1.00059; 1.0029 and 1.0032. Comparing the values of $R_1$ and $R_1$, it has been noticed that $R_1$ is always greater than $R_1$ thus a better accuracy of approximated solution is obtained with eqn./1/.

A Ramach method has been used for the evaluation of the approximated solution of eqn./2/. The method consists in determining distance of a fixed representation point $P/\gamma$ from the approximated point. Taking the solution as in eqn./2/, there is an error of 0.315% as compared with eqn./1/. This solution /eqn.2/ is more accurate.

REFERENCES

SOME PECULIARITIES OF A SONIC BULK MODULUS OF MINERAL OILS AGAINST TEMPERATURE AND PRESSURE

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INTRODUCTION

Measurements of propagation velocity of ultrasounds \( c/ \) and bulk moduli \( K_s/ \) vs \( T \) and \( p \) of petroleum products; fuels, lubricants and hydraulic fluids have shown an occurrence of irregularities of \( c/T, p/ \) and \( K_s /T, p/ \) functions.

EXPERIMENTAL

Irregularities of the S-shaped curvature for \( c/T, p/ \) and \( K_s/T, p/ \) functions have been found for a number of products. The above has been proved on the following compositions, Fig.1; mineral oil without irregularities /4/; oil with 2-methylnaphthalene /m.p. +34.4°C/ without irregularities /5/; oil with n-paraffin fraction /f.p. +11 +18°C/ with irregularities /1/. A shift of the irregularities for \( c \) and \( K_s \) functions has been found, depending on \( p \) and \( T \), and the viscosity of the products /2, 3/.

INTERPRETATION

It seems that the aforementioned peculiarities may be connected with a fractional formation of mesophases with anisotropic properties so characteristic of liquid crystals with short-range order. Probably for most petroleum products strongly associating hydrocarbons containing paraffin chains in a molecule are responsible for the mesophase formation.

REFERENCES

INTRODUCTION
The abnormal frequency dependence of sound absorption in some polymer solutions in the low MHz range has been identified as a relaxation process by which the distribution of monomer groups between the different minima of their rotational potential about the mean chain axis is perturbed by the temperature variation of the sound wave (1). The temperature dependence of position and magnitude of the maximum yielded information about the rotational potential within the polystyrene chain. This has also been the aim of investigations in polymethylmethacrylate (PMMA) solutions, where contradictory results already exist.

EXPERIMENTAL
The frequency range from 1 to 300 MHz is covered by two pulsed apparatuses, one with Soll type condenser transducers, one with surface-excited Quartz rod transducers; both nonresonant in order to allow measurements at arbitrary frequencies.

RESULTS
Syndiotactic PMMA: In all solvents (dioxane, xylene, chloroform) two maxima had to be used to represent the experimental findings within the accuracy of the measurement. Both maxima were proportional to the concentration of monomer units and shifted towards higher frequencies by a temperature increase (6 and 3.4 kcal/mole activation energy for the low- and high-frequency process, resp.). Whereas the amplitude of the slow process decreases strongly with increasing temperature and quality of the solvent, the fast process is not particularly sensitive to these parameters. A possible explanation could be that the fast process arises from monomer units on "regular" chain sequences while the slow process belongs to units near a thermal chain defect the concentration of which decreases with temperature and solvent quality. Bends in the chain could be such defects.

Isotactic PMMA: Only the fast process could be detected experimentally; an activation energy of 3.1 kcal/mole and an energy separation of 0.9 kcal/mole between the minima could be derived from the experiment.

REFERENCES
SOUND VELOCITIES IN SOME BINARY MIXTURES CLOSE TO THEIR CRITICAL SOLUTION TEMPERATURE

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A survey is given of recent experimental sound velocity data in the critical mixtures aniline-cyclohexane (U.C.S.T.30°C), water - 2 butoxyethanol (L.C.S.T.49°C) and water-triethylamine (L.C.S.T.18°C).

Use is made of a pulse superposition technique (0.6-2 MHz) by two ultrasonic resonators located at different height in the fluid so that the velocities W in both the coexisting liquid phases could be measured simultaneously. The data confirm the theoretical prediction (1) that the adiabatic compressibility \( \beta_s \) in the critical mixture remains finite. This is in contrast with the behavior in gas liquid critical systems where \( \beta_s \rightarrow \infty \) and \( W \rightarrow 0 \).

Analysis of the data reveals that over a large temperature range the difference between the sound velocities in both the coexisting phases changes according to a power law with an exponent fairly close to 0.34. At the critical composition in the water-triethylamine mixture an anomalous behaviour of W as a function of T is found. With reference to the thermodynamic relation (2)

\[
W^2 = \frac{V}{M \beta_T} + \frac{T V^2}{M C_v} \left( \frac{\delta p}{\delta T} \right)_V
\]

this anomaly can be identified as due to a divergent isothermal compressibility \( \beta_T \).

REFERENCES


Fig.1 Sound velocities in the mixture water-2 butoxyethanol.
Ultrasonic absorption in polymeric solids has been shown (1) to have a small hysteresis background throughout the temperature-frequency plane. Superimposed on the background are various relaxations. Zener showed (2) that when a viscoelastic material has a small hysteresis absorption

$$\alpha_\lambda = \frac{\pi^2}{2} \left( \frac{\partial \ln B}{\partial \ln \omega} \right) T$$

where $\alpha_\lambda$ is the absorption per wavelength, $B$ is the adiabatic bulk modulus, and $\omega$ is the circular frequency. Since $B$ is a function of $T$ and $\omega$, Eq. 1 can be written

$$\alpha_\lambda = -\left( \frac{\partial T}{\partial \ln \omega} \right)_B \frac{\pi^2}{2} \left( \frac{\partial \ln B}{\partial T} \right)_\omega$$

The preferred parameter to use in describing the anharmonic properties of a solid is the Gruneisen parameter, $\gamma$. There are two ways of expressing $\gamma$,

$$\gamma = -\frac{1}{2B} \left( \frac{\partial \ln B}{\partial T} \right)_\omega$$

where $\beta$ is the cubic coefficient of expansion, and

$$\gamma = \frac{6B}{C_p, i}$$

where $C_p$ is the specific heat per unit volume and the subscript $i$ denotes that only the inter-chain contribution to the specific heat should be used. The later restriction is peculiar to polymers and was first pointed out by Wada (3). Making use of the fact that the sound speed, $c$, is given by $c = \sqrt{B/\rho}$ and combining Eqs. 2, 3, & 4 yields

$$\alpha_\lambda = 2 \left( \frac{\partial T}{\partial \ln \omega} \right)_B \frac{\pi^2}{2} \frac{C_p, i \gamma^2}{\rho c^2}$$

It is interesting to compare the above equation with the result for phonon-phonon interaction,

$$\alpha_\lambda = T \frac{\pi^2}{2} \frac{C_p, i \gamma^2}{\rho c^2}$$

The similarity of Eqs. 5 & 6 is striking. However, Eq. 6 is not expected to explain the observed absorption in polymers since Eq. 6 is derived assuming that $\omega_T << 1$ where $\omega_T$ is the relaxation time for thermal phonons and is on the order of $10^{-12}$ sec. for polymers. The ratio of Eqs. 5 & 6 is

$$R = 2 \left( \frac{\partial T}{\partial \ln \omega} \right)_B$$

and the magnitude of the phonon-phonon interaction is about a factor of 100 smaller than that predicted here.

EIGHTH INTERNATIONAL CONGRESS ON ACOUSTICS, LONDON 1974

DISLOCATION DAMPING AND CHARGE EFFECTS IN IRRADIATED ALKALI HALIDES

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INTRODUCTION

Changes in anelastic and piezoelectric properties of bent alkali halide crystals at 85 kHz during neutron irradiation are interpreted in terms of dislocation interactions.

RESULTS

Using a simplified dislocation string model (1) and computer curve fitting, the parameters are found that give the best fit to the experimental period (modulus) curve. The theoretical and experimental damping curves are then compared and deviations are often found as shown in Fig. 1. The deviations are attributed to variations in the damping parameter B during irradiation which in turn correlate well with predicted changes in dislocation line charge (2).

CONCLUSIONS

Experiments indicate two regions with different characteristics
(1) the dislocation core region which involves pinning and charge effects,
(2) the compensating charge cloud region. Dislocation damping is a function of the average loop length as well as charges in the core and in the charge cloud region. During an interruption to irradiation, pinning and charging cease immediately but the electrical equilibrium of the cloud takes time to re-establish due to diffusion processes.

REFERENCES.

(1) J.S. Koehler, in "Imperfections in Nearly Perfect Crystals, (1952) 197.
INTRODUCTION AND EXPERIMENTAL RESULTS

The ultrasonic attenuation of quasi-longitudinal waves in the Z-direction of a ferroelectric single crystal of TGS has been studied in the vicinity of the Curie point as a function of an externally applied electric field. The increase of the attenuation with increasing strength of externally applied field $E$ at temperatures above the Curie temperature $T_c$ can explain a residual attenuation observed previously (1) above $T_c$ and related either to an existence of some needle-shaped domains or to attenuation due to fluctuation of polarization $P$ (2).

The experimental investigation has been extended to DTGS in order to get relaxation time $\tau$ and kinetic coefficient $\gamma$ in comparison to TGS. Fig.1 gives the attenuation change due to the electric bias at temperatures above $T_c$.

THEORETICAL ANALYSIS

The experimental results can be well explained using formulae (3): 

$$\alpha = \frac{2GP^2}{\rho \omega^3 (2fP^2 + EP^4)} \cdot \frac{\omega^2 \tau}{(1 + \omega^2 \tau^2)} \cdot \tau^{-1} \gamma (2fP^2 + EP^4)$$

where $G$ is the effective coefficient of electrostriction, $\rho$ is the mass density, $\omega$ is the sound velocity, $f$ is the coefficient at 4-th power of expansion of free energy, and $\omega$ is the angular frequency of the ultrasonic wave.

ELASTICITY AND INTERNAL FRICTION MEASUREMENTS ON REFRACTORY MATERIALS AT HIGH TEMPERATURES

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INTRODUCTION

Simultaneous measurements of internal friction and elasticity have been made by a new method developed from the technique of resonant ultrasonic thermometry (1,2, J.F.W. Bell et al to be published). This method is very convenient for high temperature work.

EXPERIMENTAL

Specimens were prepared in the form of a tuning fork cut into a rod or strip of material (3). A transmission line, typically 1m. long, couples the resonator to a magnetostriuctive transducer. The measurements are made by examination of echo signals. The strain amplitude was about 3x10^-5 and the frequency about 100 kHz. The report includes results for W, Ta, Re, Ir, Graphite, Sapphire and Si$_3$N$_4$ up to 2000°C. The figures illustrate the resonator and results for tantalum.

![Fig.1 Strip Tuning Fork Resonator](image)

![Fig.2 Internal Friction in Tantalum](image)

CONCLUSION

The measurements have established the reliability and convenience of this method. Automation is possible and has already been achieved for the elastic moduli.

REFERENCES

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ANELASTIC EFFECTS IN THE Pd/Ag/H SYSTEM

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INTRODUCTION

Vibration frequency and internal friction is measured over a temperature range from 80 K to 300 K in a Pd/Ag alloy containing 23% silver and variable amounts of hydrogen. Preliminary results obtained in the hydrogenated Pd/Ag alloys have already been reported (F. M. Mazzolai and F. A. Lewis, J. of Physics F. In press). More extensive results are available now.

RESULTS

Fig. 1 shows the measurements made in a circular plate annealed and then hydrogen loaded. A pronounced peak in the internal friction curves is brought about by hydrogen impurities; its mean activation energy and its mean limit relaxation time, measured for a hydrogen content of about 5% at., are 0.19 eV and 10^-12 s, respectively. The height of the peak shows approximately a linear dependence on the hydrogen amount. The relaxation is associated with a time spectrum.

DISCUSSION

Three possible mechanisms have been considered for explaining the internal friction peak: a) stress-assisted ordering of Ag-H pairs, b) movement of dislocations punched out by precipitated particles, c) stress-assisted dissolution and formation of precipitates. It may be that all these mechanisms operate at the same time, giving rise to different components of the spectrum. However, we believe the main contribution to be given by stress-assisted ordering of Ag-H pairs. With this interpretation the height is expected to vary linearly with hydrogen content and the activation energy to be close to that for hydrogen diffusion; both expectations seem to be confirmed by the experiment.
ULTRASONIC ATTENUATION IN NICKEL SINGLE CRYSTALS

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INTRODUCTION

To overcome the difficulties in interpreting the ultrasonic attenuation in ferromagnetic materials mentioned in previous papers (1) we measured the damping of transverse and longitudinal waves in nickel single crystals as function of the intensity of magnetization under various conditions.

EXPERIMENTAL

The specimens had coercivities about 0.100e after thermal treatment. The magnetically part $\Delta \omega$ of the attenuation coefficient depended strongly upon the directions of sound propagation and polarization and of magnetizing field (Fig. 1), and upon plastic deformation (Fig. 2).

![Graph](image1)

**Fig. 1** Damping of various ultrasonic waves in soft nickel

![Graph](image2)

**Fig. 2** Damping of a transverse wave in deformed nickel

DISCUSSION

The ultrasound waves are damped by micro eddy currents caused by local periodic rotations of the magnetization. Using Néel’s phase theory (2) we can interpret curves as shown in Fig. 1 for soft material. In plastically deformed crystals dislocations take part in the magnetoelastic coupling.

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GENERAL DISPERSION LAW FOR A CONDUCTIVE MAGNETOELASTIC MEDIUM

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INTRODUCTION

We have calculated a general dispersion law for a cubic elastic conductive medium in which the magneto-elastic coupling and the magnetic anisotropy energy parameters can be large. Also, the direction of the external field is taken to be arbitrary and it is not assumed to be parallel to the internal field. Previous calculations have been restricted to insulating mediums with the further implicit assumption of collinearity between the two fields. This assumption is not even valid in systems where the magnetic anisotropy is small. In our case we are considering the rare earth-transition metal alloy systems which have large anisotropy and large magnetostriction constants. Hence these materials may permit high frequency transducers.

RESULTS

We have combined Maxwell's equations and the equation of motion for the magnetization in a consistent manner without the assumption of the collinearity of the fields to yield a general dispersion law.

\[ \begin{align*}
(w^2 + c_{11}k^2)(w^2 + c_{44}k^2) & \quad \text{(magnetic secular equation)} \\
-\alpha_2^2(w^2 + c_{44}k^2)^2(2B_1/M\rho)^2 & \quad \text{M}\kappa^2 \left\{ \left(1-\alpha_2^2\right)[b+\eta(\Omega + a)] + \kappa_2 \right\} \\
-(w^2 + c_{11}k^2)(w^2 + c_{44}k^2) & \quad \text{(B}_2/M\rho) \quad \text{M}\kappa^2 \left\{ \left(1-3\alpha_2^2+4\alpha_4^2\right)\eta\Omega + \\
(\eta a+b)(1-4\alpha_2^2+5\alpha_4^2) & \quad +(c+\eta d)\alpha_2^2(1-\alpha_2^2) + \kappa_3 \right\} \\
+\alpha_2^2(1-2\alpha_2^2)^2\beta(w^2 + c_{11}k^2) & \quad (B_2/M\rho)^4 \quad (M\kappa^2)^2 \\
+\alpha_4^2(1-\alpha_2^2)^2\beta(w^2 + c_{44}k^2) & \quad (2B_1/M\rho)^2(2B_2/M\rho)^2(M\kappa^2)^2 = 0 \quad (1)
\end{align*} \]

where \( c_{ij} \) is defined as \( C_{ij}/\rho \) and \( i = 1 \) and 4. \( c_{11} \) and \( c_{44} \) are the elastic constants and \( \rho \) the crystal density. \( B_1 \) and \( B_2 \) are the magneto-elastic constants. The rest of the parameters are defined in Ref. (1). Equation (1) is seventh order in \( k^2 \), where \( k \) is the propagation constant and we have solved all of the \( k \) values by computer. There are 3 acoustic and 4 magnetic branches. The two sets of branches repel each other for some special directions of the external field. As a specific example the effect of magneto-elastic interaction on wave propagation and magnetic resonance in \( N_3 \) and \( DyFe_2 \) will be discussed.

EIGHTH INTERNATIONAL CONGRESS ON ACOUSTICS, LONDON 1974

ANOMALIES D’ÉLASTICITÉ AU VOISINAGE DE LA TRANSITION DANS RbCdF₃

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INTRODUCTION

Parmi les isolants diélectriques AMF₃ de symétrie cubique à température ambiante, RbCdF₃ joue un rôle particulier en raison de l'existence à 124 K d'une distorsion quadratique avec c/a > 1 (1).

CONDITIONS EXPERIMENTALES

Nous avons obtenu des monocristaux de RbCdF₃ par la méthode de Bridgman-Stokbarger en tube scellé de platine (2). Deux échantillons monocristallins de dimensions respectives 12x9x8 mm, et 11x6x7 mm nous ont permis de mesurer les vitesses des ondes ultrasonores longitudinales et transversales, pour les directions [100] et [110], par la méthode de résonance en ondes entretenues et par la méthode d’impulsions. La figure 1 représente les variations en fonction de la température, des constantes élastiques dans la phase cubique et, quand les mesures de vitesses sont possibles, les variations de υ² dans la phase quadratique polydomaine.

DISCUSSION

Il est intéressant de remarquer que C₄₄ se comporte très différemment de C¹₁ et C₁₂. En effet, quand la température diminue, on observe d’abord une augmentation de C₁₁ et C₁₂ dans la phase cubique, puis leur diminution brutale au niveau de la température de transition. Pour C₄₄, cet effet apparaît environ 50 K au dessus de la transition. Par contre, dC₄₄/dT est toujours négatif, même à la température ambiante.

CONCLUSION

Ce comportement est à rapprocher de celui de K MnF₃ au voisinage de la transition de type antiferrodistorsif à 184 K. La similitude de ces résultats nous permet de supposer que la quadratisation s'effectue par un mécanisme de rotation d'octaèdres, comme dans K MnF₃ (3). La détermination des courbes de dispersion de phonons et du spectre de diffusion Raman, actuellement en cours, devrait nous permettre d'observer l'existence éventuelle d'un soft mode et de confirmer ou d'infirmir l'hypothèse des rotations d'octaèdres.

REFERENCES

CONSTANTES ÉLASTIQUES DE KNiF$_3$ - VARIATIONS AVEC LA TEMPÉRATURE

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Dans le cadre des travaux effectués sur les anomalies de propagation d'ondes ultrasonores au voisinage de la température de Néel dans les isolants antiferromagnétiques de type AMF$_3$ (1) (2), nous avons étudié KNiF$_3$ qui présenta l'intérêt de conserver sa symétrie cubique au-dessous de la température critique $T_N = 246$ K (3).

Les constantes élastiques ont été déterminées à partir de la mesure des vitesses de propagation d'ondes ultrasonores dans des échantillons monocristallins préparés par la méthode Bridgman-Stockbarger. Des mesures indépendantes sur plusieurs échantillons orientés suivant les directions [100] ou [110] (dimensions en mm : 16x8x7 et 5x5x4) ont donné, à 295 K en prenant $\rho = 3.99$ g cm$^{-3}$, les valeurs suivantes :

- $C_{11} = 15.82 \pm 0.05 \times 10^{11}$ dynes cm$^{-2}$
- $C_{12} = 4.85 \pm 0.10 \times 10^{11}$ dynes cm$^{-2}$
- $C_{44} = 4.03 \pm 0.02 \times 10^{11}$ dynes cm$^{-2}$

Nous avons suivi l'évolution de ces constantes élastiques en utilisant une méthode de résonance avec détection optique. Les courbes obtenues, représentées sur la figure ci-contre, mettent en évidence, quand la température décroît, une augmentation sensiblement linéaire des constantes élastiques avec toutefois, pour $C_{11}$, une légère courbure que l'on peut attribuer à une valeur élevée de la température de Debye ($\Theta_D = 475$ K).

En outre, nous noterons essentiellement la différence avec l'antiferromagnétique RbCoF$_3$ (2) pour lequel une distorsion quadratique accompagne l'ordre magnétique. En effet, dans KNiF$_3$, la diminution des constantes élastiques est très rapide au voisinage de $T_N$ et cet effet de la transition n'apparaît en aucune façon avant la température critique comme dans les transitions à caractère structural.

Les fluctuations de spin au voisinage de $T_N$ et l'existence de domaines magnétiques pour $T < T_N$ peuvent expliquer la disparition des résonances dans la phase antiferromagnétique. Des mesures d'absorption d'ondes ultrasonores au voisinage de la transition devraient permettre une étude plus significative des interactions spins-phonons.

REFERENCES
EIGHTH INTERNATIONAL CONGRESS ON ACOUSTICS, LONDON 1974

NON-COLLINER INTERACTION AND SECOND HARMONIC GENERATION OF ELASTIC WAVES IN CRYSTALS

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The second approximation of the wave equation for elastic waves in anisotropic medium is considered. The square terms in the nonlinear equation of state and square terms of the strain tensor are taken into account [1, 2]. The solution is obtained by the slowly varying envelope approximation for the given field. In nondispersion medium the resonance solution is [3]:

\[ U = \sqrt{S_k k' k'' U''} \]  \hspace{1cm} (I)

where \( U \) is amplitude of a resulting wave, \( k', k'' \) and \( U', U'' \) are the wave vectors and amplitudes of interacting waves respectively, \( X \) is a distance of the resulting wave propagation, \( \Gamma \) is a parameter of nonlinear acoustic interaction, which characterizes the efficiency of combination wave generation [3]. For the arbitrary direction of propagation of the interacting waves and for any crystal in the case of three frequencies interaction one can obtained

\[ \Gamma = \sum_{i=1}^{3} N_i \left( \frac{C_{ijk}^{(i)}}{\rho V^2} \right) \]  \hspace{1cm} (2)

where \( C \) is polarization vector of the resulting wave, \( \rho V^2 \) is a corresponding eigenvalue of the linear equation, \( N_i \) is a unit vector of "nonlinear force":

\[ N_i = C_{ijk} n' i j k + C_{ijk} n' i k j - n' i n' j k \]  \hspace{1cm} (3)

where \( C_{ijk} \) and \( C_{ijk} \) are the second and third order elastic modulii, \( n' \) and \( n'' \) are unit vectors of interacting waves.

Taking into account the symmetry of interactions with difference frequencies generation relative to break down phonons the number of interaction types decreases to 16 instead of 21 in general.

The solution for second harmonic generation of shear waves predict some polarization effects [4]. For example, for trigonal crystals the components of the "nonlinear force" for C-direction will be:

\[ N_x = C_{xxx} \sin 2\psi \]  \hspace{1cm} \[ N_y = C_{yyy} \cos 2\psi \]  \hspace{1cm} (4)

where \( \psi \) is displacement vector's angle to X direction.

According to (4) rotation of the displacement vector of the fundamental wave by the angle \( \psi \) gives for the second harmonic the polarization under \( 2\psi \). The experimental observation of the polarization effects for shear second harmonics and collinear interactions were carried out in Si, SiO_2 and LiNbO_3 single crystals.

ULTRASONIC WAVES IN CRYOGENIC AND BOILING LIQUIDS

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INTRODUCTION

Cryogenic liquids (and ordinary liquids not far off boiling state) contain the vapour bubbles that have an influence on the propagation of ultrasonic waves.

FORMULATION OF PROBLEM

The plane ultrasonic wave of frequency \( \omega \) in a liquid that contain the bubbles of the radius \( R \) and the concentration \( n \) is considered. The wave number may be written as (1,2)

\[
k^2 = k_0^2 \left[ 1 + \frac{4}{3} \pi n R^3 \rho c_o^2 K_b (1 - \rho^2 R^2 K_b) \right],
\]

where \( c_o \) - sound velocity without bubbles, \( \rho \) - density of the liquid, \( k_0 = \omega / c_o \), \( K_b \) - compressibility of a bubble. \( K_b \) for the vapour bubble can be determined as (3)

\[
K_b = - \frac{3}{R} \frac{dR}{dp} = 3A \left[ \frac{(D/2\omega)^{1/4}}{R} + j \frac{(D/2\omega)^{1/4}}{R} + j \frac{(D/\omega)}{R} \right] + B,
\]

where \( D \) - coefficient of temperature conductivity, \( A \) and \( B \) - some constants,

\[
A = \frac{\alpha}{L \rho} \left( \frac{dT}{dp} \right), \quad B = \frac{1}{3} \left[ \frac{c_s}{L} \left( \frac{dT}{dp} \right) + \frac{1}{\rho'} \left( \frac{dp'}{dp} \right) \right],
\]

\( \alpha \) - coefficient of heat conductivity, \( L \) - heat of vaporisation, \( c_s \) - heat capacity of the vapour at saturation, \( T, p \) and \( \rho' \) - temperature, pressure and density of the vapour at saturation correspondently.

PRINCIPAL RESULTS

From eqns. (1) and (2) we have that the attenuation coefficient \( \alpha \) and the sound velocity \( c \) are as follows:

\[
\alpha \simeq 2\pi n p c_0 \omega \left[ AR^2 (D/2\omega)^{1/4} + A (D/\omega) \right],
\]

\[
c \simeq c_0 \left\{ 1 - 2\pi n p c_0^2 \left[ BR^3 + A R^2 (D/2\omega)^{1/4} \right] \right\}.
\]

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(2) G.T.Trammell, J.Appl.Phys. (1962) 33 1662;
ULTRASONIC RERADIATION FROM RAYLEIGH AND LAMB WAVES
AT LIQUID-SOLID INTERFACES

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INTRODUCTION

The surface wave excited by a sound beam incident on a liquid-solid bulk interface at the Rayleigh angle reradiates energy into the liquid at the Rayleigh angle. The reradiation and specular reflection produce a resultant sound field partially described by Schoch's "beam displacement" (1), and recently more completely by Bertoni and Tamir (2).

In a similar manner, Lamb waves can be excited at liquid-solid plate interfaces to produce similar "beam displacements" (3).

THEORETICAL CONSIDERATIONS AND EXPERIMENTAL RESULTS

Phase velocities V of free-plate Lamb modes depend on the product fd (frequency x plate thickness) and the longitudinal and shear velocities of the solid. The lowest few Lamb modes are shown for brass in Fig. 1, and for Plexiglas in Fig. 2.

If the plate is immersed in water, loading changes the V. New values are found by adding, to the free-plate equations, terms containing the ratio \( r = \frac{\text{density of water/density of the solid}}{\text{density of the solid}} \). V can be expressed in terms of the angle of sound incidence \( \alpha \), from Snell's law.

Calculations for brass (\( \rho = 8.6 \)) show that the values of V differ very little for free or for loaded plates (Fig. 1), while for solids with \( \rho \approx 1 \), the two sets of curves differ markedly (Figs. 1 and 3).

Schlieren techniques are used to experimentally verify the results. For a given solid and a fixed fd, those \( \alpha \) were determined where "beam displacement" occurred, giving values of V. Results (Figs. 2 and 3) show that a free-plate approach should not be used for plastic, ice, and other low-density plates in water.

Acknowledgment: Supported by the Office of Naval Research, U.S. Navy.

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PULSE COMPRESSION FILTER USING BULK ELASTIC WAVES AT MICROWAVE FREQUENCIES

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INTRODUCTION

Microwave acoustic signal processing has so far been accomplished exclusively by elastic surface waves. Now signal processing can be accomplished with elastic bulk waves at higher frequencies.

TECHNIQUE

A bulk wave pulse compression filter has been designed and fabricated for use between 1.0 and 1.5 GHz. The method of operation is illustrated in Fig. 1.

A lithium niobate plate was fabricated to have one end free and the upper surface polished flat and orthogonal while the other end and lower surfaces were roughened and tilted with respect to the polished surfaces, to prevent internal reflections and render incoherent any scattered energy.

A broadband thin film piezoelectric mosaic (1) transducer was fabricated on the polished end face at the upper edge adjacent to the larger polished face. An interdigital grid with continuously varying finger width and spacing was fabricated on the upper polished surface by means of photolithography and photoresist techniques.

When a linear frequency modulated rf signal was applied to the mosaic transducer, the frequency modulated bulk compressional wave launched propagated just below the polished surface, allowing its presence to be sensed by the interdigital grid. As indicated in Fig. 1, the high frequency end of the grid was adjacent to and the low frequency end remote from the launching transducer.

EXPERIMENTAL MEASUREMENTS

By arranging the frequency to sweep from 1.0 GHz to 1.5 GHz in 3 μS, the FM pulse was compressed to a pulse width of 2 nS. In this case,

\[ \text{Pulse compression ratio} = \Delta f \Delta t = 5 \times 10^8 \times 3 \times 10^{-6} \]
\[ = 1.5 \times 10^3 \]

Fig. 1 Construction of bulk wave pulse compression filter.

REFERENCES

INTRODUCTION

The dispersion characteristics of two classes of fibrous composites are investigated experimentally by observing the propagation of low-amplitude, sinusoidal waves through the composite (standard water-bath techniques were used). The first class of composites, unidirectional elastic fibers in an elastic matrix, was chosen for study because its only dispersive mechanism is geometric. The second class of composites, elastic fibers in a viscoelastic matrix, was chosen for study to illustrate the interplay between geometric and viscoelastic dispersion.

MATERIALS

The elastic-elastic composite used in this study was composed of tungsten wires embedded in an aluminum matrix. It was studied in two constituent ratios, 2.2 and 22.1 percent tungsten wires by volume. Two elastic-viscoelastic composites were used in the study. The first, a cloth-laminate quartz phenolic, was selected because its relatively fine structure emphasized viscoelastic dispersion. The second composite, stainless steel wires embedded in an Epon 828-2 matrix, was selected for its relatively course structure which emphasized geometric dispersion.

DISCUSSION

The dispersion spectra for the elastic-elastic composite demonstrate that geometric dispersion in fibrous composites is manifested as a wave filtering phenomenon in which periodic waves are selectively transmitted or reflected by the composite. The dispersion spectra for the elastic-viscoelastic composites demonstrate that either viscoelastic dispersion or geometric dispersion can be the dominant mode of dispersion in a particular composite, depending on the internal structure of the composite.

*This work was supported by the United States Atomic Energy Commission.
EINLEITUNG
In der BCS-Theorie der Supraleitung (1,2) wurde für die Übergangstemperatur $T_u$ eine Gleichung aufgestellt, die einen Zusammenhang mit der Debyeschen Temperatur $\Theta$ herstellt:

$$ T_u = \frac{1.14 \Theta \exp \left( \frac{-1}{\Theta} \right)}{k_b} , \quad \Theta = \frac{k}{3\pi^2 n} = \frac{k_B^2}{2\pi^2 n^2} \frac{3}{\frac{1}{u_x} + \frac{1}{u_y}} .$$

In diesen Formeln ist $E$ die Zustandsdichte der Elektronen an der Fermikante, $\Theta$ eine die Elektron-Phonon-Wechselwirkung beschreibende Größe, $V$ das Molvolumen, $u_x$ die Geschwindigkeit von longitudinalen, $u_y$ die von transversalen Schallwellen.

DER QUOTIENT $\Theta/T_u$
Wenn man auf Grund des Prinzips der korrespondierenden Zustände $T_u$ als Funktion der kritischen Temperatur $T_c$ darstellt, erhält man parabelähnliche Kurven (3), die aber für kleine Werte von $T_u$ ineinander fließen. Es wurde daher nach einer anderen Form des Prinzips der korrespondierenden Zustände gesucht, um die verschiedenen Gruppen der Supraleiter besser trennen zu können. Diese Form wurde in der Gestalt des Quotienten $\Theta/T_u$ gefunden. Auch wenn man berücksichtigt, dass $\Theta$ etwas zu einfach definiert und auch etwas temperaturabhängig ist, und die Übergangstemperaturen nicht immer genau genug bestimmt sind, ergibt sich der merkwürdige Befund, dass diese Quotienten sich um mittlere Zahlenwerte der Gestalt $\Theta/T_u = 2^n$ gruppieren.

Es handelt sich dabei um die Zahlen

$$4^1, 7^1, 11^2, 18^1, 29^1, 47^1, 76^1, 123^1, 199^1, \ldots$$

Da offenbar jede Basiszahl gleich der Summe der beiden vorhergehenden ist, könnte es sein, dass zu dieser Zahlenfolge noch die Werte $1^2$ und $3^2$ gehören. Dieser Befund dürfte für die Diskussion des Exponentialausdrucks in der BCS-Theorie von Interesse sein.

Nimmt man die Formel $\Theta/T_u = 2^n$ als gegeben hin, so liefert die Molekularakustik für die Übergangstemperatur bei Isotopen einen Ausdruck der Form $T_u = \frac{c_H f}{W} \cdot T$. Die BCS-Theorie lieferte bisher nur den zweiten Faktor. Durch Hinzunahme des Molvolumens kann erläutert werden, dass es Elemente ohne Isotopieeffekt gibt, und andere, bei denen die Übergangstemperatur mit steigender Isotopenmasse steigt.

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EIGHTH INTERNATIONAL CONGRESS ON ACOUSTICS, LONDON 1974

ACOUSTIC PROPAGATION MODES AT THE BOUNDARY BETWEEN Bi$_{12}$GeO$_{20}$ AND LIQUID MIXTURES

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INTRODUCTION

Two acoustic modes may propagate along the boundary surface between a solid and a liquid: a Stonely wave, with velocity close to the bulk wave velocity in the liquid, and a leaky Rayleigh wave, with velocity slightly above that of the Rayleigh wave propagating on the free surface of the solid.

A case is here considered, where the two modes propagate almost with the same velocity and thus have very close characteristics.

THEORY

If the solid is an elastically isotropic one, the velocity $v$ of non-trivial solutions must satisfy (1) the equation:

$$(1-2s)^2 - 4s[(s-1)(s-q)]^{1/2} = -\rho^{-1}(s-q)/(s-r)$$

where $s=(v_T/v_C)^2$, $r=(v_T/v_L)^2$, $q=(v_T/v_L)^2$, $\rho$ is the density of the solid relative to that of the liquid, $v_T$, $v_C$, and $v_L$ respectively are the velocities of the transversal and longitudinal waves in the solid and that of the compressional one in the liquid.

In Fig. 1 the dependence is reported of $v/v_R$ ($R$ stands for Rayleigh) vs. $v_T/v_L$ for the case of Bi$_{12}$GeO$_{20}$ crystal in contact with water-alcohol mixtures. It is to be noted how the two velocity curves approximate each other at values of $v_T/v_L$ close to unity; in this range of values, the two modes have very similar wave profiles.

REFERENCES

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PRESSESS INTERNES ET DIMENSIONS MOLECULAIRES ACOUSTIQUES
DANS LES SOLUTIONS AQUEUSES

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INTRODUCTION

La vitesse de propagation de l'ultrason dans les liquides en corrélation avec les valeurs de la densité et de la tension superficielle, offre la possibilité de connaître certains grandeurs moléculaires, inaccessibles par ailleurs.

PARTIE EXPERIMENTALE

Nous avons mesuré la vitesse de l'ultrason par une méthode de diffraction dans les solutions aqueuses de halogènes à des températures et concentrations différentes, en déterminant aussi les densités et les tensions superficielles qui y correspondent.

PARTIE THEORIQUE

En exprimant la pression interne en fonction de la vitesse (1):

\[ P_1 = \frac{v^2 \rho}{2 \chi} \]  

(1)

respectivement en fonction de la tension superficielle et du diamètre moléculaire (2), sur base de considérations thermodynamiques:

\[ P_1 = \frac{\sigma}{\phi} \]  

(2)

\[ \phi = 2\pi \chi \frac{\sigma}{\rho v^2} \]  

(3)

on obtient:

Le calcul des valeurs à l'aide des relations (1) et (3) permet aussi bien l'étude qu'à l'effet de l'agitation thermique sur le champ des forces, que l'intervention des ions, en fonction de leurs propriétés spécifiques.

BIBLIOGRAPHIE

UNE ÉQUATION DE PROPAGATION DES ULTRASONS DANS LES LIQUIDES

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INTRODUCTION

Les calculs basés sur la représentation de la phase liquide par extrapolation du modèle du gaz réel, respectivement du solid cristallin (1) ont permis l'élaboration de certaines équations de la vitesse de l'ultrason, dans leur grande majorité en désaccord avec les résultats expérimentaux.

PARTIE THÉORIQUE

Considérant la participation simultane des forces élastiques et des heurts moléculaires dans le mécanisme de la transmission des perturbations ultrasonores, on obtient pour la vitesse de propagation :

$$v = v_0 e^{an}$$

où:

$$a = \ln \left( \frac{1}{A} \frac{\bar{V}_M}{v_0} \right); \quad n = \frac{S + D}{S}$$

$\bar{V}_M$: vitesse moyenne des molécules, $A$: rendement de heurt, $S$: distance intermoléculaire, $D$: diamètre moléculaire.

La relation (1) se vérifie aussi bien pour les liquides ideaux que pour les réels y compris l'eau et les solutions aqueuses, relevant, aussi en accord avec les résultats expérimentaux, les sauts de vitesse aux points de changement d'état.

La relation permet de déduire une expression indépendante de température :

$$v \cdot v^{1/a(n-1)} = R_A$$

qui pour $a(n-1) = 9$, devient identique à la formule empirique de Rao.

BIBLIOGRAPHIE.

Experimental and theoretical investigations of the second harmonic generation using Blustein-Gulyaev waves on CdS plate and parametric interaction of elastic surface waves are described.

Blustein-Gulyaev wave second harmonic generation.

The 12 MHz Blustein-Gulyaev mode generation and second harmonic detection were carried out by means of interdigital transducers with electrodes parallel C-axis of CdS crystal. The relaxational dependence of the second harmonic amplitude on crystal conductivity was measured. Comparing the first and second harmonics acoustic powers the magnitude of the crystal nonlinear parameter for Blustein-Gulyaev wave was shown to be equal 450 under optimum conductivity, i.e. approximately two orders higher than its dark value.

Assuming the semiconductor dielectric permittivity to be complex the inhomogeneous wave equation describing the elastic wave-electron interaction in piezoelectric semiconductor under the quadratic approximation was derived. The solution of this equation allows to determine the second harmonic characteristics and their dependence on semiconductor parameters. For shear wave or Blustein-Gulyaev mode propagating along C-axis of CdS crystal the expression of the nonlinear acoustical parameter \[ \Gamma \] takes the form:

\[
\Gamma = \frac{(e_{15})^3 \mu (\omega_c/\omega)^2}{c_{44}^2 \varepsilon \varepsilon_s \left[1 + (\omega_c/\omega)^2\right]^2}
\]

where \( e_{15} \) - piezoelectric constant, \( c_{44} \) - elastic modulus, \( \varepsilon \) - dielectric permittivity, \( \varepsilon_s \) - wave velocity, \( \mu \) - mobility, \( \omega_c \) - relaxational frequency, \( \omega \) - wave frequency.

Surface waves parametric interaction.

Nonlinear interaction of 21.5 MHz contradirectional surface waves and idler generation due to the presence of the double frequency pump field in layered systems LiNbO\(_3\) -CdS, LiNbO\(_3\) -CdSe was studied experimentally. The influence of the conductivity and liquid layer between the semiconductor and substrate thickness was investigated. It was shown that the presence of liquid layer [2] considerably increases back wave generation efficiency.

References.
Cette étude sur les ultrasons est destinée à vérifier le principe d'une méthode de contrôle non destructif de matériaux. Elle devrait permettre la mesure de divers paramètres physiques, la détection d'inhomogénéités à l'échelle microscopique (telles que la dislocation) et l'évaluation de défauts à l'échelle macroscopique (tels que criques, retassures, soufflures).

La méthode considérée fait appel à l'analyse du signal ultrasonore ayant traversé un échantillon métallique d'épaisseur invariable. Cette analyse est menée à partir du relevé des variations de phase et d'amplitude du signal. De ce fait, les informations recueillies sont plus complètes que celles obtenues par les méthodes usuelles, liées à la seule mesure de l'amplitude. Par exemple, pour une fréquence ultrasonore de 10 MHz, une différence de phase de $10^{-2}$ degré (détectable à l'aide d'un interférogramme) est capable de déceler une variation d'épaisseur de $10^{-2}$ mm et une variation $\Delta \nu$ de la vitesse du son dans l'échantillon ($\nu \approx 6000$ m/s) égale à 7 m/s. Une étude de l'hétérogénéité et de l'anisotropie des matériaux orientés est donc possible.

Les études théoriques font appel à la propagation dans les cristaux (mise en évidence de la diffusion due à la grosseur des grains - calcul de l'amortissement et de la vitesse de phase).

Les premières études expérimentales ont conduit au relevé de cartes d'amplitude et de phase de signaux ayant traversé des échantillons sains (calcul de l'atténuation) servant de référence à des échantillons inconnus.

Une autre série de mesures utilise un banc équipé de transducteurs électrostatiques à large bande. Cette méthode permet de connaître la loi d'atténuation de l'échantillon en fonction de la fréquence.

Les études se poursuivent actuellement pour comparer les deux méthodes et connaître leurs domaines de validité.
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EFFECT OF VOIDS ON STRESS-WAVE EMISSION FROM CARBON-FIBRE COMPOSITES

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INTRODUCTION

The stress wave energy (\(E^S\)) release and the mode of failure are investigated for carbonfibre/epoxy composites (CF/EC) as a function of the void content.

EXPERIMENTAL

Three specimens containing the same fibre (diam. 0.125 mm) content were moulded each for a different dwell time, respectively 0, 11.5 and 21 minutes. (1). Three-point bending tests were carried out on an Instron machine. Stress wave energy (\(E^S\)) was calculated from \(E^S = \sum V_i^2\) where \(N_i\) is the number of pulses of amplitude \(V_i\). (2). Waveforms and modes of failure were recorded photographically.

DISCUSSION OF RESULTS

Figs. 1(a), (b) and (c) relate to the variation of \(\sum E^S\) with time of stress application for specimens having 1.4%, 7% and 24% void content respectively. Curve (a) shows the greater \(\Delta (\log \sum E^S)/\Delta t\) just before fracture, but exhibits a more gradual initial slope compared with the specimens of greater void content (\(\sum E^S\) is the bent line 'envelope'). For a large void concentration (c) the stress interaction between voids appears to appreciably lower \(E^S\). Also the time to fracture is enhanced by the presence of voids.

REFERENCES

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FIG. 1 VARIATION OF \(E^S\) WITH TIME OF STRESS APPLICATION
ULTRASONIC FREQUENCY ANALYSIS IN NON-DESTRUCTIVE TESTING

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The frequency distribution of a broadband ultrasonic pulse—scattered from a discontinuity—depends on the size and orientation of the discontinuity as we have shown in previous papers (1,2). For a flat laminated type of discontinuity the reflected spectrum appears in the form of frequency maxima and minima. This behavior of the spectrum is explained (2). The presence (or the lack) of such characteristic frequency maxima is applied in several areas of nondestructive testing:

1. FLAW IDENTIFICATION
Figure 1 shows the difference in frequency spectrum observed for burst type (crack like) and porosity type (spherical) defects in a titanium sample.

2. FLAW SIZE DETERMINATION
The size d of a randomly oriented flaw is given (2) in terms of the frequency spacing between consecutive maxima as

$$d = V[\Delta f \sin \theta]^{-1}$$  (1)

where V is the sound velocity and θ is the flaw angle orientation. In Figure 2 the actual sizes of seven artificial flaws in a stainless steel block are plotted vs. the measured values by frequency analysis and by the use of a conventional technique.

3. TRANSDUCER ALIGNMENT
4. TUBE WALL THICKNESS MEASUREMENT
5. NOTCH DEPTH MEASUREMENT

Fig. 1 Flaw Identification

Fig. 2 Comparison of Flaw Size Determination Techniques

REFERENCES
LATERAL RESOLUTION IN DETECTION OF UNDERGROUND PIPES WITH IMPULSIVE SOUND

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INTRODUCTION

The authors designed a detection method of pipes buried underground by pulse echo technique with impulsive sound and study towards its practical use is proceeding.
In this paper are described experimental results take to fine the minimum interval at which laid two steel pipes buried at the same depth in a model sand bath can be detected separately.

MEASUREMENT AND RESULT

An electromagnetic induction type sound source and four piezoelectric type receivers were used as an impulse sound source and as a receiver respectively.
The measurements were taken in a model sand bath of 11 m in length, 8 m in width and 2 m in depth. Steel pipes of 8 cm in diameter were used as reflective bodies. All of ten steel pipes buried at a depth of 1.2 m. The five paired pipes were laid at five steps of interval of 10, 20, 30, 40 and 50 cm. The sound source was placed just above one of the paired pipes and the four receivers were arrayed perpendicular to the pipes and symmetrically with regard to the sound source at intervals of 50 cm. At the place where the experiment was taken, the outputs of the four receivers were recorded in a magnetic tape by a tape recorder.
A signal processing and a cross sectional display were made at our laboratory. The signal processing technique is similar to a polarity correlation technique. Three variable delay lines for the received signals were used to obtain the cross sectional display.
As a results, it was made clear that the two pipes laid at an interval of more than 30 cm can be displayed separately.

Fig. 1 shows the clearly separated images of the two steel pipes laid at a depth of 1.2 m and an interval of 50 cm.

Fig. 1 Cross Sectional Display of Underground Effective Range: 1.0 m to 1.4 m
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AMPLIFICATION OF BLEUSTEIN-GULYAEV WAVES IN TETRAGONAL
AND HEXAGONAL CRYSTALS

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INTRODUCTION

The formula for the amplification coefficient of Bleustein-Gulyaev waves in piezoelectric semiconductors (1) is approximately the same as that of White for shear volume waves (2). Lithium iodate (class 6) is one of the dielectric crystals in which the above waves may exist (3-4). Therefore, the amplification of Bleustein-Gulyaev waves in the system: piezoelectric crystal-semiconductor, is very interesting.

THEORETICAL ANALYSIS

The analysed system consists of two half spaces which are filled up by a piezoelectric crystal (class 4 or 6) and a semiconductor, respectively. Between the media there is an infinitesimal vacuum gap. From the equations of motion, the electrostatic equations, the constitutive relations for both half spaces and from the boundary conditions one can obtain the expressions for the amplification coefficient and velocity of Bleustein-Gulyaev waves. The conditions of existence of these waves in the analysed system are also discussed. Detailed calculations of the amplification coefficient and velocity are performed for a system which consists of silicon and lithium iodate (LiI03). For example, at the frequency f = 30 MHz, the maximum value of the amplification coefficient is of the order 100 dB/cm.

CONCLUSIONS

One can obtain analytical expressions for the amplification coefficient and velocity of Bleustein-Gulyaev waves in a piezoelectric crystal-semiconductor system. Lithium iodate may find application as the effective piezoelectric material in surface wave amplifiers utilizing Bleustein-Gulyaev waves.

REFERENCES

ULTRASONIC SURFACE WAVES AND ELECTRIC CHARACTERISTICS OF PIEZOELECTRIC SEMICONDUCTING CRYSTALS

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FORMULATION OF THE PROBLEM

It is known that Rayleigh waves are used for determining mechanical characteristics of a solid (1). The purpose of this work is an attempt to extend this method for studying electric characteristics of surface layer of a piezoelectric semiconductor. Our first experiments of this kind (2) showed that the use of Rayleigh waves and "field-effect" method permits to receive an information about traps in a thin surface layer of a CdS-crystal.

RESULTS AND CONCLUSIONS

The experiments carried out with several samples of CdS-crystal have shown that electronic attenuation permits to get information about surface conductivity \( \sigma \) and about qualitative law of its distribution in \( \approx 60 \) microns thick surface layer. Electronic amplification data allowed to get information on effective drift mobility in \( \approx 5 \) microns thick surface layer of CdS and on its dependence on \( \sigma \). The use of Rayleigh waves seems promising for studying surface states of piezoelectric semiconductor too, as experiments (3) and our calculations revealed strong dependence of both electronic attenuation and gain of these waves on surface states of piezoelectric semiconductor.

REFERENCES

THEORETICAL AND EXPERIMENTAL STUDY OF PSEUDO-INTERDIGITAL SURFACE-WAVE TRANSDUCERS: THEIR USE IN ACOUSTIC DELAY LINES

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INTRODUCTION

The structure of pseudo-interdigital Rayleigh-wave transducers consists in a metallic grating deposited on the quartz crystal surface and a counter electrode separated from the grating by a dielectric layer. The acoustical efficiency is comparable to that of the interdigital transducer but, for a given width of the strips, the fundamental resonant frequency is twice higher.

THEORETICAL STUDY

The electric potential distribution from a pseudo-interdigital transducer has been calculated in an isotropic approximation, under small piezo-electric coupling conditions a) with a computer (1 to 6), using a numerical relaxation technique (Frankel-Young's method); the potential maps are given as thickness e and permittivity E of the dielectric layer are varied b) according to an approximate analytical method where each strip is replaced by a series of regularly spaced electric lines.

Efficiency factor E is defined and its value, as a function of e, E and the relative width a of the strips are shown. Expressions for the spatial harmonic content are derived from the superficial potential distribution and compared to the case of interdigital combs (7).

EXPERIMENTS. USE IN ACOUSTIC DELAY LINES

Pseudo-interdigital transducers were constructed and tested (8) (from 20 to 300 MHz). Several types of delay-lines are described in detail:

a) Large bandwidth, constant delay-line; time delay: 10 ± 0.02 µs, 3db B.W.: 8 MHz, central frequency: 32.5 MHz, insertion losses: 40db.

b) Continuously variable delay-line, by moving the counter electrode (9); central frequency: 20 MHz, time delay varying from 1 µs to 12 µs, adjusting accuracy: 20 ns, B.W.: 400 KHz, 1.20 < gain < 1.92.

c) Dispersive delay-lines; central frequency: 30 MHz, B.W.: 7 MHz, compression ratio: 30; insertion losses: 50 db; spurious signal level: -20 db (without correction).

Some other possibilities are mentioned: large bandwidth delay-line of type (b); delay-line with multiple discrete time-delay...

REFERENCES

ACOUSTIC EMISSION: THE SOUND OF STRESSED MATERIAL

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INTRODUCTION

Mechanical deformation of stressed materials produces elastic stress waves known as acoustic emissions (1). By detecting and analysing these signals it is possible to locate cracks and flaws in stressed structures and to monitor their growth. The usual form of the observed signal is a ringdown waveform (Fig.1) representing the response of the structure to a local impulsive excitation.

WAVE PROPAGATION

Between the source and the detecting transducer the acoustic emission wave undergoes attenuation, dispersion and multiple reflections (1) within the transmitting structure. This affects both source location and quantification of the emission data.

STRUCTURAL CALIBRATION

Simple techniques have been evolved for measuring the response of a structure to simulated emissions (3). Using a piezoelectric transducer as an exciter, ringdown counts at the detector are measured as a function of input pulse amplitude. The resultant curve characterises the structure/instrumentation combination, providing a reference against which naturally occurring acoustic emissions have a quantitative significance. Typical experimental results will be presented.

SOURCE LOCATION

Dispersion has a major influence on the emission waveform during propagation, since the wall thickness of the test structure is often comparable with the wavelength of sound in the frequency region employed. The implications for source location (4) will be discussed.

CONCLUSIONS

Acoustical wave theory has an important bearing on the practical application of acoustic emission in non-destructive testing. This subject area is understood in broad outline but there is a continuing need for more detailed investigation.

REFERENCES

ULTRASONIC ASSIST IN DRYING OF FINE PARTICLES

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INTRODUCTION

The rate of drying fine, heat sensitive particles by normal procedure is a relatively slow process. The introduction of ultrasonics into a drying system was found not only to accelerate the drying rate at relatively low temperatures but also extended the constant drying rate period.

EXPERIMENTAL

Airborne ultrasonics at a frequency of 20kHz with intensities up to 170 db were used. The ultrasonic radiation was normally introduced directly onto the top surface of the bed of particles being dried. In some experiments, the ultrasonics were introduced through the bottom of the container holding the particles. In most of the experiments air flowed directly across the top layer of particles. However, a series of runs were made in which the air flowed through the bed of wet particles. The particle sizes used in this study ranged from 0.1 to 1000 microns with bed depths from 0.6 to 10cm. The materials studied included plastic powders, fine coal, and sludge.

RESULTS

Ultrasonics when applied to the top surface increased the drying rate as shown in Fig.1. Ultrasonics extended the constant drying rate period as long as Biot's relationship held. The formula is: \( f > \frac{\nu}{kd^2} \) where \( f \) is the frequency in cps, \( \nu \) is kinematic viscosity in cm\(^2\)/sec., and \( d \) is channel diameter in cm (1).

Ultrasonics was found to have an effect on the drying rate to a depth of approximately 5cm in a bed of particles.

REFERENCES

EIGHTH INTERNATIONAL CONGRESS ON ACOUSTICS, LONDON 1974

NON-LINEAR EFFECTS ASSOCIATED WITH HIGH-POWER ULTRASONICS

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INTRODUCTION

Recently a detailed review of non-linear elastic wave propagation in solid materials has been published(1). In the present work the aspects of non-linear elastic wave propagation in real crystalline solids pertinent to the field of high-power ultrasonics are presented.

THEORY AND EXPERIMENT

Non-linear elastic waves differ from linear elastic waves in several important aspects. An initially sinusoidal longitudinal stress wave of a given frequency distorts as it propagates and energy is transferred from the fundamental to the harmonics which appear. The degree of distortion and harmonic generation is directly dependent on the amplitude of the wave. A pure mode longitudinal non-linear wave may propagate alone, but a pure mode transverse non-linear wave cannot propagate without the existence of an accompanying longitudinal wave. On the other hand, a non-linear transverse wave does not distort when it propagates in a defect free solid. Non-linear elastic waves can interact with other waves in the solid. At the intersection of two ultrasonic beams a third ultrasonic beam can be generated. Interaction with thermal vibrations causes energy loss from the wave and heat generation in the test specimen. The degree of interaction in all cases is directly dependent on the amplitude of the wave. In the present work consideration is given to non-linearities caused by superimposed static stress fields, by non-linearities caused by lattice defects, and by non-linearities caused by the finite amplitude of the elastic wave itself.

REFERENCES

ULTRASONIC PRECIPITATION OF AEROSOL PARTICLES USING A PIEZOELECTRIC Emitter

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INTRODUCTION

In a previous paper (1) we studied the characteristics of a piezoelectric transducer for air-borne ultrasound based on a new type of flexural vibrating stepped plate (2). The high directivity and efficiency of this emitter disclose new possibilities in the ultrasonic coagulation of aerosols. We present here the first experimental results using this emitter in the precipitation, under static conditions, of an aerosol constituted by the exhaust smoke from an internal combustion engine.

EXPERIMENTAL

Observations using an electron microscope showed that the linear size of the aerosol particles varies from 0.2-2 microns. Consequently the emitter was designed for a frequency of 22 kHz (3).

The precipitation chamber was a plastic tube of square section. The selected dimensions (90x12x12 cm) were much greater than the wavelength in order to obtain a uniform distribution of the standing waves in the chamber.

The sound pressure at the antinodes and the frequency were controlled by an acoustic probe situated in the radiation axis.

The aerosol settling rate was measured by the changes in light transmission of the smoke during the process. Fig. 1 gives some results of the experiments achieved with an initial concentration of the aerosol of about 7 g/m³. The settling time is very high for spontaneous precipitation (p=0). Under the action of ultrasonic energy the process is accelerated. For comparison, precipitation curves for two different pressures are shown.

REFERENCES

EIGHTH INTERNATIONAL CONGRESS ON ACOUSTICS, LONDON 1974

FATIGUE STRENGTH OF STRUCTURAL MATERIALS AT SONIC AND ULTRASONIC FREQUENCIES OF LOADING

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Investigations of fatigue strength in wide range of frequencies of loading (from infra- to ultrasound) have been so far carried out on traditional materials: steel, aluminium, copper (1, 2, 3). The purpose of our work was to increase the number of materials by including there some modern heat-proof steels, alloys on the basis of nickel, titanium and aluminium. The results of the tests on some of them are shown on Fig. 1.

Fatigue tests on the specimens of the above-mentioned materials were carried out with low (infrasonic) frequency on hydraulic vibrator, with average sonic frequencies on standard electrodynamic rig, and with high sonic and ultrasonic frequencies on magnetostrictive installations worked according to the design suggested by W. Mason and constructed by E.A. Nappiras. The tests were carried out at longitudinal and transversal vibrations of the specimen.

The main results: a) specimens from different metals and alloys being rather effectively cooled, their fatigue strength monotonously increases with the increase of frequencies; b) in tests without cooling with rather high frequencies of loading fatigue strength decreases due to the local overheating of the material; c) effect of frequencies of loading on fatigue strength is small for steels and alloys in high strength and low plasticity state.

REFERENCES


Fig. 1 Experimental results of the frequency effect on fatigue limit of metals (specimens from titanium alloys BT22 and OT4-1 and aluminium alloy D16T were tested without liquid cooling)

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MEASUREMENT OF THE ACOUSTICAL OUTPUT POWER OF ULTRASONIC HIGH POWER TRANSDUCERS USING AN ELECTRICAL HIGH-FREQUENCY WATT-METER

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Introduction

We have discovered a new method of measuring the acoustical output power of ultrasonic power transducer by making some modifications on their differences of electrical input power which are measured at the same vibrational velocity under on load and no load.

Measuring Principle and Method

The following eqn.(1) for acoustical output power $\dot{W}_a$ of transducer by our method can be obtained.

$$\dot{W}_a = (\dot{W}_L - \dot{W}_N) - (\dot{W}_{dL} - \dot{W}_{dN})$$  (1)

Keeping either the exciting current or voltage in sine wave form, electrical input power $\dot{W}_L$ (on load), $\dot{W}_N$ (no load) and exciting current or voltage at the same vibrational velocity under on load and no load are measured. $\dot{W}_{dL}$ and $\dot{W}_{dN}$, which represent the power for the internal magnetic or dielectric losses of the transducer under on load and no load, are measured at a damped frequency of transducer regard as exciting current or voltage which are measured by the above measurement. The results of the above are substituted into eqn.(1) to calculate the $\dot{W}_a$.

Measuring example and results

In order to determine the validity of this measuring radiated acoustic power $\dot{W}_a$ was measured by the calorimeter to be compared with the results by our method. As a transducer, a 28kHz π type magnetostrictive ferrite transducer provided with water acoustic load was used. As a wattmeter, a wattmeter using semiconductor Hall effect element was used. As a calorimeter, a Dewar bottle (capacity max. 700cc) was used. Fig.1 shows the measured results, which are fairly close to the values, the difference being within 2~3%.

Conclusion

This measuring method has provided a good prospect for the simple measurement of the acoustical output power of transducer in the various applications of ultrasonics.

Fig.1 Measured value of acoustical output power by wattmeter (o) and by calorimeter (x) vs. vibrational velocity at end surface of transducer.
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A CONTACTLESS REAL-TIME METHOD OF MEASURING POWER IN AN ULTRASONIC WAVE GUIDE

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INTRODUCTION

Many applications of high energy ultrasound make it necessary to measure ultrasonic power in real time. A contactless method was developed for power measurements in metal wave guides with radii much smaller than the wave length.

PRINCIPLE OF THE METHOD

The method consists in the determination of the time average of the product of instantaneous values of stress and acoustic velocity. A wave guide is surrounded by a concentric metal electrode, which is polarized with a constant voltage $U$ through a resistor $R$. Air gap thickness $d_0$ between the electrode and the wave guide constitutes a fraction of a mm. Once a wave propagates in the wave guide, its radius $a$ varies periodically thus changing the electrode capacitance $C_0$. If $RC_0$ is much greater than the wave period an alternating voltage $e$ appears on the electrode. Then the normal stress in the wave guide direction $Z$ equals

$$T_{zz} = \frac{E \cdot d_0 \cdot e}{C_0 + \frac{1}{v \cdot a \cdot U \cdot C_0}}$$

where $C'$ - stray and wiring capacitances of the electrode, $E$ - Young modulus, $v$ - Poisson ratio. The acoustic velocity is determined by the electrodynamic method /2/.

EXPERIMENTAL

A probe developed for measuring stress and acoustic velocity was calibrated according to Eq. /1/, and by means of a microscope, respectively. The phase shift occurring in the probe between these magnitudes was measured and eliminated. Power measurements were performed in the wave guide coupled with two magnetostrictive transducers in the dummy load system and also with the standing wave method, in which the acoustic velocity was determined. Preliminary measurements show the powers measured to coincide with an accuracy of 1 dB or 1.5 dB.

REFERENCES

POWER SUMMATION OF VIBRATION ENERGY

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In conventional (one-dimensional) ultrasonic vibration systems, mechanical outputs are limited to 1 or 2 KW. Actually, such industrial applications of ultrasonics as welding of plastics or drawing of steel pipe etc. require 10 KW or more. The authors have devised a concentrator of ultrasonic power into a load, resulting more than 15 KW of mechanical power be available. We devised three types of concentrators such as L-L$^1$), L-L-L and R-L ones. In this paper, we explained summation by L-L type concentrators, for example.

An L-L type converter has such a construction that two half wavelength resonant rods are crossed at their nodes, thus having four free ends (Fig.1). We can summarize vibrational energy into one of these free ends from the others, and vice versa, considering vibrational phase. This resonator has two fundamental resonant frequencies. Efficiency of summations, defined as the ratio of output power and input one, is very high because this device has Q-factor more than $10^4$.

The authors trially made two concentrators in cascade (Fig.2), and attached five transmitters ($T_1$-$T_5$) besides one receiver (Rec.). Fig.2 shows electric power dissipated in the resistance ($R_L$), load of receiving transducer, changing numbers of excited transmitters. Parameters are power given to a single transmitter. From Fig.2, for example, we have confirmed the summation of ultrasonic power into single load. Division of power from single source into plural loads can be made, of course.

1) 7th ICA, 25-U-7
Le problème de protection de l'atmosphère des éjections industrielles devient à présent un des plus actuels. La difficulté essentielle consiste à précipiter de minces aérosoles dispersifs. Nos recherches des années dernières ont montré les résultats perspectifs de l'utilisation de l'agrandissement préliminaire des particules aérosoles dans de puissants champs acoustiques.

Dans notre rapport nous avons présenté les résultats des recherches théoriques et expérimentaux du processus cinétique de coagulation des aérosoles industriels.

Les recherches de l'interaction spatiale des particules aérosoles dans les champs acoustiques ont montré que le rapprochement efficace des particules voisines a lieu dans une étroite zone spatiale des angles près de la direction des oscillations. Cela a permis en utilisant le modèle de la Symétrie spatiale des interactions puissantes pour calculer le changement de la concentration nombre des particules du réel aérosol industriel pendant la sonorisation en fonction des paramètres des champs acoustiques et de ceux du flux de poussière et de gaz tenant compte des changements du coefficient de diffusion des particules dans de puissants champs sonore et de la densité des agrégats formés.

Selon l'équation de la regression du processus de coagulation acoustique obtenue expérimentalement le changement de la concentration nombre des particules dépend beaucoup de l'intensité du son, du temps de sonorisation et de la concentration primaire de l'aérosol. Le changement de fréquences du son et de température du flux de poussière et de gaz dans une large bande n'exerce pas l'influence sur les processus de coagulation des particules dans les champs acoustiques. A l'aide de microscope à electron nous avons fait les recherches de structure, de dynamique de croissance et des paramètres physiques des agrégats produits par la coagulation acoustique.

L'utilisation de coagulation acoustique de basse fréquence en combinaison de précipiteur optima permet de résoudre avec succès le problème de protection de l'atmosphère des éjections industrielles.

Références
INTRODUCTION
The propagating surface wave in a piezoelectric solid is accompanied inside and outside the solid by traveling electric field. The wave velocity depends upon piezoelectric and dielectric properties as well as elastic properties. For device applications it is important to evaluate the strength of piezoelectric coupling to surface waves, that is the electric field associated with propagating surface waves of given amplitude. This information we can obtain in several ways but also quite easy from the change in wave velocity produced by eliminating the storage of electric energy outside the solid.

MEASUREMENTS
Measurements of surface wave velocity change were performed on a special digital wave velocimeter type SA-1000, at 10 MHz.

![Diagram of SA-1000 sing-around velocimeter for surface waves.]

The change in wave velocity was produced by application of a very thin metallic film on the surface of the quartz crystal between the transducers.

RESULTS
The effective coupling constant was calculated from the relation:

\[ k = \frac{2 \Delta c}{c_0} \]

where \( c_0 \) is the wave velocity with an electroded surface and \( \Delta c \) is the change in velocity produced by removing the metallic film. For quartz \( k = 0.009 \) was obtained.

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EIGHTH INTERNATIONAL CONGRESS ON ACOUSTICS, LONDON 1974

PRACTICAL ULTRASONIC PULSE SPECTROSCOPY BY FOURIER TRANSFORM

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INTRODUCTION

An investigation has been made of the frequency dependence of amplitude and phase information when broad band ultrasonic pulses, in the region 1 - 30MHz, are reflected from layered targets.

It is shown that phase information has more general application in the study of layered media than amplitude information.

EXPERIMENTAL

An on-line computer performing Fourier analysis of sampled ultrasonic pulses allowed both amplitude and phase information to be studied. Layers of various acoustic impedances, velocities and attenuations have been investigated in several configurations. In particular, results have been obtained for layers of magnetite grown on mild steel. In all cases excellent agreement between experiment and theory have been achieved.

THEORETICAL ANALYSIS

The reflection of ultrasonic waves from layered media has been investigated by several authors (e.g. 1). The present work extends these results, using Fourier Transform techniques, to pulsed systems. The effect of attenuation within the layers is also considered and shown to have a significant effect on both amplitude and phase in the frequency range covered. The phase change as a function of frequency from a single layer (thickness d, velocity of sound c) in which the attenuation (a) is non-zero is given by equation (1).

\[ \phi(f) = \tan^{-1} \left[ \frac{-r_2 e^{-2ad} (1-r_1^2) \sin \theta}{r_1 + r_2 e^{-2ad} (1+r_1^2) \cos \theta + r_1 r_2 e^{-4ad}} \right] \]  

(1)

\( r_1, r_2 \) = reflection coefficients at the front and back interfaces respectively and \( \theta = \frac{4\pi fd}{c} \).

REFERENCES

A systematic method for the determination of the electroacoustical characteristics of piezoelectric transducers

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The wide application of piezoelectric transducers (PT) in various scientific fields leads to many problems in the design of transducers. It may be noticed that there are quite different conceptions and approaches (1) regarding the design of PT from the standpoint of various scientific fields.

In order to investigate and present a general approach to the analysis of PT systematically the author has obtained the following results owing to the study of the approaches (2) coming from the theory of electromechanical transducers, the electric input impedance, the equivalent circuit and the transmission of plane waves through multiple-layers as follows:

PT can be included into a general theory from which all practical solutions result. The matrix method analysis has been chosen which uses the solution of wave equation and at the same time includes the electrical and mechanical sides of a transducer. The method enables the obtaining of a general mathematical description of all electroacoustic characteristics as well as a simple, synoptic approach to the analysis of the problems connected with the design of a transducer. The characteristics of composite transducers are particularly suitable in contemplation. In calculation has been noticed that it is advisable to use normalized quantities as they are used in the theory of transmission lines and microwave circuits as well.

The most useful way in the analysis of transmitting characteristics is to start from the "e" equation. In this way one can obtain the parallel junction of both the static and motional impedance so that the obtained normalized form of conductance corresponds directly to radiated acoustic power. Five basic technical constructions of PT which are mostly found in the ultrasonics have been checked by this treatment. By means of the introduced term "acoustic gain" one can compare the effectiveness of any complex form of a transducer relating to the simplest reference form, piezoelectric plates or bars with symmetrical load or air-backed.

Out of the obtained characteristics it is possible to define the receiving response by means of the reciprocity theorem, as well as the Q factor and efficiency. The obtained basic theoretical results have been checked experimentally.

REFERENCES

EIGHTH INTERNATIONAL CONGRESS ON ACOUSTICS, LONDON 1974

SOUND SOURCES FOR PRODUCING INTENSE ULTRASONIC FIELDS IN SMALL REGIONS IN AIR

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PRINCIPLE

A circular diaphragm is driven at its center C and is in a flexural vibration with several nodal circles \( N_1, N_2, \ldots \) on it, as shown in Fig.1. P is a point on the center axis. An intense sound field ought to be produced in a small region about P if

\[
PN_2 - PN_1 = PN_3 - PN_2 = \ldots = \lambda_a / 2,
\]

where \( \lambda_a \) is the wavelength in air.

EXPERIMENTAL RESULTS

The construction of one of our sound sources is shown in Fig.2. The diaphragm has seven nodal circles.

The experimental results for this sound source are summarized as (i) for small amplitudes, the resonance frequency is 18.4 kHz, the quality factor is 300, and the efficiency is 80 \%, (ii) for 10 W of electric power, the sound pressure level is 164 dB at the point P (PC=2 cm), (iii) for 10 W, the peak-to-peak value in amplitude is 15 \( \mu \)m at the edge of the diaphragm, and (iv) the value of

\[
\left[ PN_i / (\lambda_a / 2) - i \right] \cdot 180^\circ
\]

fluctuates within \( \pm 50^\circ \) for \( i=1,2,\ldots,7 \).

DISCUSSIONS

Theoretical formulas are obtained in the case of a free vibration of a circular diaphragm with a mechanical impedance at its center. It is found that the experimental results are in good agreement with the calculated ones about (i) the radius of the nodal circles, (ii) the relative amplitude of the diaphragm at the loops, (iii) the relative sound pressure distribution, (iv) the sound pressure level at the point P for electric power 10 W, and (v) the acoustic power for electric power 10 W.
Intensity of sound radiated by high power magnetostrictive transducers into liquid without cavitation is limited because of nonlinear effects in magnetostrictive materials. The dependence of fundamental component amplitude of the magnetostrictive stress $\sigma_1$ upon the amplitude of sinusoidal excitation induction $B_1$ in rod specimens made of different magnetostrictive materials was measured quasistatically under various bias magnetic field $H_0$. All the experimental curves $\sigma_1(B_1)$ had a maximum like described for nickel in (1) and ferrites in (2). The maximum values of $\sigma_1 = \sigma_{1\text{max}}$ increased with growing $H_0$; corresponding values of $B_1 = B_{1\text{max}}$ lightly depend on $H_0$ and were roughly equal to $B_s/2$, where $B_s$ is the saturation induction.

From the theoretical analysis of the magnetoelastic part of the thermodynamical potential including 4th order terms we have for $\sigma_1$ (3):

$$\sigma_1 = -EB_0B_1 \left[ \lambda + \lambda_1\left(4B_0^2 + 3B_1^2\right) \right],$$

where $E$ is the Young modulus, $\lambda$ and $\lambda_1$ - magnetoelastic constants, which may be determined from low amplitude measurements. The calculation gives $\sigma_{1\text{max}} = 2/3 \cdot aB_{1\text{max}}$, where $a = \sigma_1/B_1$, measured under small $B_1$ and given $H_0$. The discrepancy of theoretical and experimental dates in the table may be due to neglecting of high order terms in calculations and of the decrease of $B_0$ in the presence of big alternating induction.

<table>
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</table>

REFERENCES

THE DETECTION OF MICRO-INCLUSIONS IN FLOWING LIQUIDS

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INTRODUCTION

Many techniques have been evaluated to detect particles or gas/vapourous bubbles in flowing liquids. These techniques include pulse-echo, doppler, phase change, using separate transmitter and receiver, and it is found that the continuous wave technique is the most suitable.

In a previous paper (1), the basic continuous wave technique was discussed and preliminary experiments were reported.

THEORETICAL ANALYSIS

Fig. 1 shows the attenuations of ultrasonic radiation due (a) to bubbles and (b) to particles. The bubble resonant frequency $f_o$ shown is given by the formula:

$$f_o \cdot r = 3.28 \cdot 10^{-3} \text{ kHz.metre in water (2)}$$

$r$ being the bubble radius.

The Q factor of the resonance is approximately inversely proportional to the ratio of the gas pressure in the bubble to the hydrostatic pressure, and is a measure of the vapourous cavitation that could occur.

Particles on the other hand show no resonance phenomena and absorb ultrasonic energy in proportion to the particle area.

EXPERIMENTAL

Using narrow ultrasonic beams at right angles to the direction of liquid flow, good results have been obtained in discrete and collective bubble detection, working in the region AB of the graph of Fig. 1. In this region the absorption is proportional to the bubble area. Using suitable display techniques it is possible to distinguish between bubbles and particles. A range of instruments have been produced from the results of this work, which cover a wide variety of applications.

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EIGHTH INTERNATIONAL CONGRESS ON ACOUSTICS, LONDON 1974

THE APPLICATION OF A NEW ANALOG METHOD IN MEASURING ULTRASONIC TRANSDUCERS

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Medrzycki J

INTRODUCTION

The new analog method for the simulation of nonlinear functions has been applied in measuring impedance of electroacoustical transducers. This method gives possibilities of measuring frequency dependence of real and imaginary parts of complex impedance and admittance. The measured variables can either be recorded on an audio recorder, or fed to the computer for further processing (1).

THEORETICAL ANALYSIS

Nonlinear functions are represented in the compound form, where component functions - time models are generated by solving eqn., 1/

\[ \dot{x} + \omega^2 x = 0 \]

As an intermediate variable the autonomic time \( \psi \) is used. Variables ReZ and ImZ are obtained by realization of time models

\[ \text{ReZ} = \frac{1}{Z} \int_{0}^{\psi} Z \, d\psi \]

\[ \text{ImZ} = \frac{1}{Z} \int_{0}^{\psi} Z \, d\psi \]

and \( Z \) is found by

\[ Z = \int_{0}^{\psi} \frac{E}{U} \, d\psi \]

These operations can be repeated with the operating frequency of 160-2000 cps. The obtained accuracy of simulation is of 0.04%.

REFERENCES

A METHOD FOR MEASURING THE ELECTROMECHANICAL COUPLING FACTOR OF PIEZOELECTRIC MATERIALS BY MEANS OF TRANSIENT RESPONSE ANALYSIS

Kikuchi Y  
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In our previous paper (1), we performed a transient response analysis of piezoelectric thickness mode transducers, and have given general equations when arbitrary electrical and acoustical loads (resistive loads) are connected to them. In the equations, if the mechanical impedances of the acoustical loads are put to zeros, the Laplace transform of the current $I_S(t)$ that flow the plates becomes as follows:

$$I_S(p) = \frac{C_0V_S(p)p^2}{\tau(p-a_1)(p-a_2)} \left[ 1 - \frac{4\alpha}{\tau(p-a_1)(p-a_2)} \right] \sum_{n=1}^{\infty} \frac{(-1)^n}{n!} \left( \frac{p^2+\frac{p}{\tau+2\alpha}}{(p-a_1)(p-a_2)} \right)^n e^{-np\tau}$$

where, $C_0$: Clamped Capacitance of a piezoelectric plate, $\tau=\frac{C_0R_S}{R_S}$, $R_S$: Internal resistance of an electrical source, $\alpha_1=\frac{-1+\sqrt{1+8\alpha\tau}}{2\tau}$, $\alpha_2=\frac{-1-\sqrt{1+8\alpha\tau}}{2\tau}$, $\alpha=k^2/T$, $k$: Electromechanical coupling factor, $T=\frac{1}{v_o}$, $t$: Thickness of the plate, $v_o$: sound velocity, $p$: Laplace operator.

Current $I_S(t)$ is obtained by performing inverse transform of Eq. (1). Fig.1 shows four examples of waveforms of current $I_S(t)/I_S(0)$ when the coupling factor $k$ are 0.1, 0.3, 0.5 and 0.7 in the case that $\tau/T=0.3$ and that the exciting voltage is a unit step, i.e., $V_S(t)=V_0U(t)$, ($V_S(p)=V_0/p$). As is easily seen in Fig.1, the current at $t=T$, i.e., $I_S(T)/I_S(0)$, increases monotonously with the increase of $k$. Fig.2 shows $I_S(T)/I_S(0)$ as a function of $k$ by taking $\tau/T$ as a parameter. This figure is used for the measurement of electromechanical coupling factors.

Fig.1

Fig.2

REFERENCE


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ULTRASONIC TRANSDUCER FOR AIR RADIATION

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INTRODUCTION

The modes of vibrating circular plate, driven at its center by electromechanical vibrator are quite complex. Consequently, directional characteristics of the radiator seems to be also complex. To improve them, following methods have been devised to radiate the sound of the same phase, the acoustical absorbing materials are put on the adequate places of the plate, and plates with some steps on its surface are used to unify the phase at the radiation surface in the air(1).

In this report, the desired results are obtained by use of the ringed conical horn of which radiation surface is devised to give a piston motion.

EXPERIMENTAL

The mechanism of the transducer is easily understood by Fig.1(a,b); Parts of sound radiated by vibrating plate (about 39 mils thick) which are counter phase against the air-born waves radiated from horn surface are eliminated by acoustical absorbing materials (shown in Fig.1).

Thus the approximate piston motion of radiation surface is attained by the multiplication effects of vibrating plate and conical horn.

In Fig.2(a,b) and Fig.3(a,b), some examples of directivity pattern are shown. These patterns are obtained when the horn are placed about 3.9 mils in front of the plate (dia., 2.33 inches) vibrating by nodal circule modes ($m=4$, $n=0$).

In Fig.2 and 3, the length of horn is $5\lambda$ and $5\lambda$ respectively where $\lambda$ is sound length in the air. And a,b are corresponding ones in Fig.1.

REFERENCE

EIGHTH INTERNATIONAL CONGRESS
ON ACOUSTICS, LONDON 1974

SOFAR PROPAGATION IN THE OCEANS OF THE SOUTHERN
HEMISPHERE

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INTRODUCTION

Although acoustic propagation in the SOFAR channel has been studied for some time now (1), most experiments have been conducted in the Northern Hemisphere. Using data which has become available recently, we have attempted a survey of the general SOFAR propagation characteristics to be expected in the Southern Hemisphere.

EXPERIMENTAL RESULTS

Long range acoustic propagation data have been obtained in several recent experiments in the South Pacific, notably Project KIWI ONE (2): by comparison of acoustic propagation data with oceanographic data we can identify regions having different SOFAR characteristics.

CONCLUSIONS

The comparison shows that a major factor affecting SOFAR propagation conditions in the South Pacific is the influx of cold Antarctic Intermediate Water. This flows northward from the Antarctic Polar Front (3) at a depth of about 1000 metres. The evolution of velocity profile shape (figure 1) from the shallow axis Antarctic profile to the classic SOFAR profile found near the Equator can be linked to the relative amount of cold water present. One consequence of this change in velocity profile shape is that at mid-latitudes SOFAR signals do not in general terminate immediately following the rain pressure peak (K.M. Guthrie, to be published). Furthermore there is some evidence that regions which exhibit low frequency acoustic attenuation which is in excess of the extrapolated absorptive loss may be associated with regions of relatively high velocity water-mass movement. Based on this experience in the South Pacific, we make some tentative regional predictions of both SOFAR propagation arrival structure and low frequency attenuation in the South Atlantic and South Indian Oceans.

REFERENCES

SOUND SPEED DISPERSION AND FLUCTUATIONS IN THE UPPER OCEAN

Medwin H
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Simultaneous data of ocean microstructure and sound phase shift measured from a stable platform in Bass Strait, Australia, have provided new relations between the statistics of the medium and the statistics of the local sound phase speed near the sea surface in the open ocean. Because of dispersion due to ambient bubbles (1), average sound phase speeds in the frequency range 15 to 100 kHz differ as much as 2.5 m/sec from the 3 MHz "precision" velocimeter values down to depths of 7.6 m in the presence of wind speeds of 25-30 knots. These differential propagation speeds imply average total bubble volume fractions of the order of $10^{-7}$ with standard deviations approximately one-fifth of the mean value. The differential sound speed appears to increase approximately proportional to the increase in wind speed. The relative differential speed decreases with increasing depth roughly as the cube of the ambient static pressure.

Under these ocean conditions the predominant cause of the local sound phase fluctuations at 24.4 kHz and 95.6 kHz is shown to be bubble activity rather than temperature fluctuations. At 24.4 kHz, the phase fluctuations are due to the random change of volume fraction of bubbles. At a frequency such as 95.6 kHz, where there may be a large resonant bubble population, the predominant part of the frequency spectrum of the sound phase modulation is caused by the changing bubble radius due to the varying ambient pressure produced by the ocean surface wave. The sound phase spectrum therefore has the same peak frequency and mimics the $F^{-5}$ frequency dependence of a wind-generated ocean wave height spectrum to two octaves higher than the frequency of the peak power. (See figure) At still higher phase modulation frequencies, the surface pressure effect drops low enough for temperature-caused fluctuations to dominate.

This research was part of Project BASS, Basic Air Sea Studies, conducted in coordination with, and with the assistance of, the Royal Australian Naval Research Laboratory.

REFERENCES

A SIMPLIFIED RAY-TRACING METHOD FOR FAST CALCULATION
OF A SPATIALLY-AVERAGED PROPAGATION LOSS MATRIX

Bachmann W  
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de Raigniac B

The conventional ray tracing method for calculating propagation loss of underwater sound waves is based on the approximate assumption of constant energy in ray bundles of differentially small cross-section. Achieving a sufficiently high density of intensity values in the range/depth plane requires a very high number of rays to be traced (order of 100 rays per km max. range). Even at this effort the results are meaningful only in caustic-free regions. Refinements of ray theory which take more into account the wave character of sound lead to rather precise descriptions of a monochromatic sound field, including the strong fluctuations over distances of a few wave lengths in caustic regions. However, due to modulations of ray direction by inhomogeneities in the medium and on the boundaries the uncertainty of the true location of the caustic usually exceeds by far the fine resolution of this sound field description.

For estimating propagation loss and reverberation levels at sonar frequencies (1 to 10 kHz) one needs to know the spatially averaged sound intensity field with a resolution cell size much larger than one wave length. As a first attempt to verify this concept a propagation loss model has been developed (1) which contains the following two averaging processes: a) Conservation of energy is assumed for ray bundles with finite opening angle \( \theta \); hence a difference quotient replaces the differential quotient in the usual ray theory formula for intensity, \( I \propto dI/dz \).

b) The range/depth plane \((x/z)\) is divided into a regular pattern of layers and columns. Contributions of every ray-bundle are incoherently summed in the proper boxes. This results in a matrix description of the intensity field.

(Typical box size: \( \Delta x = 500 \text{m}; \Delta z = 50 \text{m} \)). The vertical spacing \( \Delta z \) of a ray bundle - and therefore the size of the averaging window - increase with range, thus automatically adapting to the increasing uncertainty of intensity distribution versus depth.

Results of a corresponding small FORTRAN program (500 statements) seem to agree considerably better with deep- and shallow water, non-monochromatic, 3 hrs-averaged propagation loss measurements (at 5 to 50 km range) than those of much more sophisticated programs (2).

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(1) W. Bachmann (1973) 217, in "Fortscritte der Akustik DABA-73", VDI-Verlag, Düsseldorf/Germany, (in German)

(2) B. de Raigniac, W. Bachmann and J.S. Cohen

ACUSTICA (1974) 20
CURRENT EFFECTS ON THE PROPAGATION OF SOUND WAVES IN SHALLOW WATER

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INTRODUCTION

Current effects on the propagation of sound-waves in shallow water of about 100m depth are calculated theoretically. The propagation losses of sound-waves in up or down currents of water are calculated and an about 30dB maximum difference of losses is presented.

THEORETICAL

In shallow water, only the RBR and/or BRR propagation will contribute to the long range propagation. The number $j$ of bottom reflections of the RBR mode has been calculated by the following equation,

$$ j = \left[ (g \pm \nu) R \cos \theta_b / \nu - (\sin \theta_b + \sin \theta_s) \right] / 2 \sin \theta_b + 1, \quad (1) $$

where $\cos \theta_b (\nu / \nu) \cos \theta_s, g$ is the vertical gradient of sound velocity, $\nu$ is that of current velocity and $\theta_b$ is the radiation angle. According to K.V. Mackenzie (1), the sound intensity at the horizontal range $R$ will be expressed by the following equation,

$$ I(\text{dB}) = -10 \log RH + 10 \log \sum_{j=1}^{\infty} r^j [\sin \theta_j] \theta_j^2, \quad (2) $$

where $H$ is the depth of water and $r$ is the energy reflection coefficient of the bottom.

NUMERICAL EXAMPLES

For negative constant $g$ and $\nu, H=100m$ and source depth is 50m, calculated losses vs $R$ are shown in Table 1. The reference level is at a unit distance from the source.

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<th>$\nu (\text{sec}^{-1})$</th>
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</table>

REFERENCE

Stationary wave acoustic fields, satisfying Helmholtz equation, and the difference methods of the computation of such fields are considered in this paper. In this case the derivatives in the equation are approximated by the finite differences, and the region is covered by the grid. The diffraction problems in the open infinite regions, the diffraction problems of the waves on the aperture in the shield and on the diaphragm in the waveguide are investigated. The iterative alternating Schwarz method at small dimensions of the intermediate region is used jointly with the difference approximation method. The high effectiveness of this method is shown to derive the numerical calculations of the diffraction problems.

The difference approximation method is applied also to the problem on the wave field in the finite region, filled by the nonuniform absorptive medium. The region is also covered by the grid, the joints of which are divided into two families: the families of even and uneven joints. To decide the difference problem this division allows using the iterative circuit being one of the circuits of the fractional steps method. The high rate of the iterative circuit convergence is used to decide the wave problems. The examples are given.

REFERENCES:

GENERAL METHOD FOR LOCATING CAUSTIC POINTS ALONG A SPECIFIED RAY

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INTRODUCTION

This theoretical paper develops a new method for rapidly determining caustic points of a ray family. The method applies to any profile for which the sound speed $C$ is a continuous function of depth only. Caustic points are defined as those values of depth $Z$, range $R$, and phase velocity $C_m$ for which $Q(R, Z, C_m) = 0$, where $Q = \partial R(Z, C_m)/\partial C_m$. The ray angle, $\theta_\circ = \cos^{-1}(C/C_m)$. The analysis extends that of (1).

OLD AND NEW METHODS FOR DETERMINING CAUSTIC POINTS

The old method searches through rays with depth fixed. For specified $Z$, $C_m$ and $R$ are determined by Newton's method with iteration on $C_m$. The phase velocity correction, $\Delta C_m = -Q/Q_2$, where $Q_2 = 3^2 R/\partial C_m^2$. Each iteration requires a time-consuming complete re-determination of the complicated functions $Q$ and $Q_2$ which depend on the complete history of the ray path over each profile layer traversed. Examples are given in (1-3), where $Q$ and $Q_2$ are designated as $dR/dC_m$ and $d^2R/dC_m^2$ respectively. The method is complicated by multiple caustic points and fails at a cusp where $Q$ and $Q_2$ vanish simultaneously.

The new method searches in depth along a fixed ray. For specified $C_m$, $Z$ and $R$ are determined with iteration on $Z$. The depth correction, $\Delta Z = -Q/Q_1 - Q^2 Q_{11}/2Q_1^3$, where $Q_1 = \partial Q/\partial Z = -\csc^2 \theta \cot C_m^{-1}$ and $Q_{11} = 3^2 Q/\partial Z^2 = -\csc^2 \theta (1 + 3 \cot^2 \theta) C_m^{-2} dC/\partial Z$. The use of both terms of this expression for $\Delta Z$ represents a second-order method while use of the first term only represents a first-order (Newton's) method. Each iteration requires an evaluation of the simple functions $Q_1$ and $Q_{11}$ at the last interface only. Re-determination of $Q$ is not required but only an updating by adding in the contribution over the last layer of thickness $\Delta Z$. In this method there is precisely one caustic point. Cusps pose no problem and $Q_1$ never vanishes.

RESULTS

Computation times have been compared for an ocean profile based on curvilinear segments (4). The new second-order method is about 2% faster than the new first-order method. (Fewer iterations for the second-order method are almost offset by the longer computation time per iteration.) The time for the old method is proportional to the number of profile layers. For a 22-layer profile the new second-order method is about 25 times faster than the old method. The new method then is superior for mapping caustics for a multi-layer profile.

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EIGHTH INTERNATIONAL CONGRESS
ON ACOUSTICS, LONDON 1974

PRODUCTION ACOUSTIQUE MARINE PAR PETIT FOND

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INTRODUCTION

Afin d’étudier la propagation acoustique basse fréquence en régime entretenu par petit fond, on a transposé le problème dans la gamme ultrasonore sur un modèle réduit. Le rapport d’échelle est de l’ordre de 1/2000° et la fréquence utilisée est de 124 kHz.

RESULTATS EXPERIMENTAUX

Le modèle choisi, constitué d’une forte couche de sable fin surmontée d’une lame d’eau de quelques centimètres dans une cuve de 5 m de long permet l’étude du champ acoustique jusqu’à dix modes présents.

Un fond en pente nous montre le comportement des différents modes lorsque la profondeur d’eau varie avec la distance à la source.

La réalisation d’un gradient de température (8°C pour une hauteur d’eau de 45 mm) nous permet de voir son influence sur une propagation en présence de quatre modes.

INTERPRETATION DES RESULTATS

On montre que la théorie de Pekeris (1) modifiée pour tenir compte de la qualité du matériau sous-jacent (sable à la place d’un fluide) peut être vérifiée dans de bonnes conditions jusqu’à dix modes présents.

Le champ acoustique s’exprime sous la forme :

\[ \phi = \frac{2 \pi}{\text{H}} \sqrt{\frac{2}{\pi r}} \sum_{n} e^{-i(k_n r + \pi/4)} e^{-\gamma_n r} G(k_n) \sin(\beta_n h) \sin(\beta_n z) \]

où l’on relie le coefficient d’atténuation \( \gamma_n \) de chaque mode au coefficient de réflexion du fond par :

\[ \gamma_n = - \frac{L_n(R_n(\theta_n))}{2H \tan \theta_n} \]

Lorsque la profondeur d’eau croît avec la distance à la source il n’y a pas création de modes supplémentaires. Lorsqu’elle décroît les modes disparaissent pour des profondeurs d’eau correspondant aux hauteurs de coupure de ces modes.

La perturbation apportée par le gradient de température est faible lorsqu’il y a quatre modes et correspond aux calculs effectués à partir des travaux de A.O. Williams (2).

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EIGHTH INTERNATIONAL CONGRESS ON ACOUSTICS, LONDON 1974

CUMULATIVE EFFECTS ON AN ACOUSTIC WAVE BY WEAK INHOMOGENEITIES

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INTRODUCTION

Though inhomogeneities in the ocean's index of refraction "η(\vec{r},t)" are regularly weak, their cumulative effect on the signal becomes significant at an extended path length. For improved applicability relative to ray theory, Rayleigh-Born approximation (R-B-A) or the method of Rytov, variants on the renormalization methods have been adopted (1). Most such formulations, however, have postulated homogeneity for "η" in its deterministic part, "η₀", and in the correlation function of its random part "η₁(\vec{r},t)". Here, an extension is presented to include both a slowly varying "η₀(\vec{r})" and correlation function of "η₁(\vec{r},t)". Such an extension offers a closer description of a variety of transmission channels in the ocean.

ANALYSIS

The interaction problem between wave, exp \(i(\omega t - \vec{k} \cdot \vec{r})\), and medium is solved directly in terms of the scattered amplitude, "I(\vec{r})", that reaches the receiver (2). The governing equations are coupled through the coherent component, "I₀", and the incoherent component, "I₁", of the field "I". They are decoupled up to a second order iteration in the incoherent subspace of "I(\vec{r})", that is:

\[
I₁(\vec{r},t) = η₁(\vec{r} - \vec{k},t) + \epsilon G(\vec{p}) η₁(\vec{r} - \vec{p},t) I₀(\vec{p}) + \epsilon G(\vec{p}) [η(\vec{r} - \vec{p},t) - P η₁(\vec{r} - \vec{p},t)] ×
\]

\[
[η₁(\vec{p} - \vec{k},t) + \epsilon G(\vec{p}) η₁(\vec{p} - \vec{p},t) I₀(\vec{p})]
\]

\(\epsilon\) is a fixed parameter; G(\vec{p}) is the Green function in transform space. \(P\) is the statistical ensemble operator. The value of "I₁" is substituted in the non-convolution type integral equation for "I₀". This equation already includes the radiation and boundary conditions constraining the field. Its solution is performed using the strongly peaked local properties of "η" in the transform space (3); there results:

\[
I₀(\vec{r},r) = \frac{η₀(\vec{r} - \vec{k}) + \epsilon A(\vec{r} - \vec{k}) G(\vec{p}) G(\vec{p}) \phi / 2 - \vec{p},r) [1 + \epsilon G(\vec{p}) η₀(\vec{p} - \vec{p})]}{[1 - \epsilon G(\vec{p}) η₀(\vec{r} - \vec{k}) - \epsilon G(\vec{p}) (\vec{p} - \vec{p},r) G(\vec{p}) A(\vec{r} - \vec{p})] [1 + \epsilon G(\vec{p}) η₀(\vec{p} - \vec{p})]}
\]

(2)

where \(A(\vec{p} + \vec{p}_2) \phi (\vec{p}_1 - \vec{p}_2,2,τ)\) is the inhomogeneous correlation function of "η₁(\vec{p},t)"; 
τ = τ₁ - τ₂. With the expressions in (1) and (2), the incoherent power, the doppler spread and the correlation function may now be determined for "I₁".

All these statistical measures of "I" are particularly relevant to the design of coherent communication and remote sensing systems in the ocean; they have been calculated respectively for slab, sphere or cylinder geometries. Some, results on such calculations will be discussed and compared to those obtained by R-B-A; their similarity may be inferred from the expressions in (1) and (2) for a small expansion parameter "\epsilon".

REFERENCES

An analysis is being made of the fluctuations of sound waves propagating through a medium with small random variations in the refractive index superposed upon a smoothly-varying refractive index. Let $\rho$ be the horizontal distance and $z$ the vertical distance in the medium; we assume that the zero-order solution $p_0(\rho, z)$ is known for a point source in a medium with $z$-dependent refractive index, $n_0(z)$. Since $p_0$ is the Green's function for this medium, we may formulate an expression for the case of a medium with refractive index (squared) of the form $n_0^2(z) + \alpha n(r)$, where $\alpha$ is the r.m.s. refractive index variation and $n(r)$ is a random function of space. For small $\alpha$, a single-scattering approximation can be made, using $p_0$. The analysis is proceeding using the usual Fourier-Bessel representation of the field variables for the zero-order solution. Comparison will be made with the known results for the homogeneous zero-order medium ($n_0=1$); and the constant-gradient (Pekeris) problem will also be discussed.
INTRODUCTION

Although stochastic wave theory does not require perturbation techniques in principle to provide solutions (1&2), in practice, ensemble expectations are usually resolved by perturbing the random medium (3&4). It will be shown that first forming a uniform wave functional $F[u(x)]$ of a medium $u(x)$ of randomness $a$ and then perturbing the resultant expectation $\langle F(x) \rangle$, via $a$, will produce nonsecular relations in agreement with experimentally verifiable results for atmospheric and oceanic sound propagation.

DEVELOPMENT

A two-variable expansion (5) for the solution of the 1-D stochastic Helmholtz equation produces a uniform relation for the acoustic pressure wave $p(x)$. Its expectation

$$\langle p(x) \rangle = r_0 \exp \left(-a^2 k_0^2 L x + ik_0 x\right), \quad 0 \leq a \ll 1, \quad (1)$$

is nonsecular and depends on the wave-number $k_0$, $a$, range $x$ and integral scale $L$ as expected. In contrast, if $p(x)$ is expressed (conveniently but indiscriminately) as an outer expansion in $a$, the consequent ordered set (finite) of differential equations produce secular expression for $\langle p(x) \rangle$.

In 3-D, a uniform two-variable expansion produces the eikonal and transport equations, to order $k_0^2$, which can be solved for $\langle p(x) \rangle$ nonsecular. Two techniques are given: (1) an Eulerian-Lagrangian method, for statistically isotropic media, which is analogous to a Wiener integral approach and (2) a stochastic perturbation of ray theory which permits a macrovaration of the mean sound speed. The coefficient of intensity fluctuation saturates with range as found in ocean experiments.

CONCLUSION

Distinguishing between uniformly expanded wave functionals and the randomly perturbed medium over which they propagate, avoids the cumulative phase effects that yield secular expectations.

REFERENCES

Ray theory calculations done for a multiple caustic convergence zone generated by an arbitrary constant gradient profile do not provide an adequate description of the sound field. Due in large part to ray theory deficiencies near caustics, ray computations show poor agreement when compared with normal mode calculations done using a finite difference approach (1). When results from modified ray theory (2) are used near caustics instead of ray theory, agreement with the normal mode results improves considerably. The figure shown demonstrates this agreement in the vicinity of the first caustic in the convergence zone resulting from a typical deep water sound velocity profile. In other comparisons, the oscillations present in the typical propagation loss versus range curve for a convergence zone are shown to be not only due to double arrivals to the right of a particular depth minimum caustic, but also due to interference to the left of the caustic between its shadow zone contribution and real rays associated with the preceding caustic. Supported in part by ONR and DNA (Subtask V99QAXNA003-04).

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DEVELOPMENT OF A THEORETICAL MODEL FOR SOUND PROPAGATION IN A DEEP, VARIABLE OCEAN

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INTRODUCTION

The development to be described attempts to steer a middle course between that of searching for an exact solution on the one hand and that of accepting a crude numerical approximation on the other. The result is admittedly an approximate theory, but one whose level of approximation seems well suited to the imperfect state of present-day knowledge of oceanic variability. It has the advantage of dealing in a practical way with source and receiver directivities, with large-scale sound-speed variations, with small-scale fluctuations in sound speed due to internal waves and turbulent motion, and with the rough sea surface.

THE MODEL

The basis of the approach is the representation of fields as an angular spectrum of plane waves. A directive source emits an angular spectrum which is proportional to its far-field pattern. The directive sensitivity of a receiver is also specified in the similarly practical terms of its angular plane-wave response. The large-scale irregularities in sound speed refract the individual plane waves according to the laws of geometrical acoustics, and hence the well-known techniques of ray tracing can be incorporated in the model (1).

The rough sea surface, according to the Kirchhoff approximation, induces a random spatial and temporal phase modulation in each plane-wave component incident upon it, the phase being proportional to the instantaneous surface height (2). A similar effect occurs when the acoustic wave field is transmitted across a pronounced thermocline whose depth is randomly perturbed by internal waves (R.H. Clarke, to be published). The resulting reflected or transmitted field for each plane-wave component of the incident field is itself an angular spectrum of plane waves, which eventually, after refraction by the large-scale irregularities in sound speed of the medium, will be incident on the receiver. Hence, by integrating these plane waves over all directions, the received signal and its statistics can be estimated and related to oceanographic measurements.

The above picture may be modified by the presence of turbulent fluctuations in sound speed within the body of the ocean, which have the effect of cumulatively degrading the coherence of each propagating plane wave. But again this effect can be related to oceanographically measurable quantities in the final result.

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A FOURIER PERTURBATION MODEL FOR THE PROPAGATION OF
SOUND IN THE DEEP OCEAN SOUND CHANNEL

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INTRODUCTION

The propagation of sound in the deep ocean sound channel has been studied by means of idealized models as well as arithmetic solutions for realistic sound speed profiles. This effort seeks solutions for realistic sound speed profiles, including microstructure, which permit study of the effect of changes in the sound speed profile on the propagation of sound.

FORMULATION OF THE MODEL

The model is based on the parabolic sound speed profile

\[ C(Z) = C_0 (1 - \alpha^2 Z^2)^{-1/2} \]  \(1\)

where \(Z\) is the depth coordinate, \(C_0\) is the minimum speed of sound, and \(\alpha\) is a constant which determines the "width" of the deep sound channel. Substitution of eq. (1) into the scalar wave equation with sinusoidal point source results in a normal mode series solution (1,2). This model is extended to more general sound speed profiles by writing a Fourier series expansion of the "non-parabolic" part of the sound speed profile, i.e.

\[ C(Z) = C_0 \left[ (1 - \alpha^2 Z^2) + \sum_j \left( A_j \sin \omega_j Z + B_j \cos \omega_j Z \right) \right]^{-1/2} \]  \(2\)

SOLUTION OF THE WAVE EQUATION

A previous paper (3) details the extension of the parabolic sound speed profile to more general cases by means of a polynomial correction to \(C(Z)\) and perturbation theory. This paper, using sinusoidal correction terms to \(C(Z)\), results in a perturbation correction series in terms of generalized Laguerre polynomials. The perturbation correction series converges uniformly and therefore the Fourier parameters of \(C(Z)\) can be varied to study the effect of changes in the shape of the sound speed profile on deep ocean propagation loss.

REFERENCES

EIGHTH INTERNATIONAL CONGRESS ON ACOUSTICS, LONDON 1974

COMPUTER SIMULATION OF LONG-RANGE OCEAN ACOUSTIC PROPAGATION USING THE PARABOLIC EQUATION METHOD

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INTRODUCTION

A new theoretical-numerical method has been developed which has enabled us to accurately and efficiently compute entire two-dimensional (depth and range) low-frequency acoustic fields in realistic ocean environments.

METHODS

The theoretical development is based on the parabolic equation approximation to the reduced wave equation (1):

\[ i \frac{\partial \psi}{\partial r} + A \psi + B \psi = 0, \quad A = \frac{1}{2k_0} \frac{\partial^2}{\partial z^2}, \quad B = \frac{k_0}{2} (n^2 - 1), \quad (1) \]

where \( k_0 = \omega/c_0 \), \( n^2 = c_0^2/c^2(z,r) + iv(z,r) \). This parabolic wave equation automatically includes diffraction and all other full-wave effects as well as depth and range dependence of the sound speed \( c \) and volume loss \( v \) and variable bathymetry.

Eqn. (1) is solved numerically by marching forward in range \( r \) using the split-step Fourier algorithm (2):

\[ \psi(z,r+\Delta r) = e^{iA\Delta r/2} e^{iB\Delta r} e^{iA\Delta r/2} \psi(z,r). \quad (2) \]

The exponential operators are evaluated by means of the fast Fourier transform (denoted by \( F \)) in the depth variable \( z \):

\[ e^{iA\Delta r} \psi(z) = F^{-1} \left\{ e^{-ik^2\Delta r/2k_0} [F\psi(z)] \right\}. \quad (3) \]

This algorithm has exponential accuracy in \( z \), second order accuracy in \( r \), and is unconditionally stable. Further, it is fast, efficient, and is readily implemented on computers.

RESULTS

Operational computer programs based on the above methods have been used to reliably predict long-range underwater propagation effects in areas of practical interest. Representative results are presented in the form of detailed computer-drawn dB contours which illustrate SOFAR, BSR, surface duct, bottom bounce and other modes of propagation in the presence of horizontal thermoclines, sea-mounts and sea-ridges, migrating sound channel axes, multiple sound channels, and so forth.

REFERENCES

SOUND FIELD IN CONVERGENCE ZONE IN OCEAN

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The sound field distribution in the convergence zone in a deep region of the ocean was investigated. The depth of the underwater channel axis was about 1000-1200 m; the transmitter and the receiver were placed at the depth 150 m. Tonal pulses with carrier frequencies 1.2 kcps and 5 kcps and the pulse duration 1 sec were repeated with the II sec period. The distance between the transducers varied continuously while received signals were recorded. An example of the records for the frequency 1.2 kcps is shown in Fig.1 (levels in db). Each dot corresponds to one pulse. Two variability scales are seen in the figure. It appears that the small scale represents fluctuations of transmitted signals and the large scale represents the regular interference pattern. In order to verify this assertion the smoothing of the records was made by averaging groups of 10 points (the thick line in Fig.1). The resulting curve shows the same variability scale as the computed interference curve (Fig.2). Similar results were obtained for 5 kcps signals.

![Fig.1](image1)

![Fig.2](image2)

References

NORMAL-MODE CALCULATION OF UNDERWATER SOUND PROPAGATION FROM DIRECTIONAL SOURCES

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We have obtained the normal-mode solution for the problem of sound propagation in an under-ocean sound channel with velocity profile $c(z)$, for the case that emission from the sound source $\rho(\mathbf{r})$ occurs in a directional fashion. The wave equation

$$ [v^2 - (1/c^2(z))(\partial^2/\partial t^2)] p(\mathbf{r}) = \rho(\mathbf{r}) $$

(1)

is separated in cylindrical coordinates assuming harmonic time dependence. The normal-mode depth functions satisfy

$$ (d^2 u_n(z)/dz^2) + [(\omega^2/c^2(z)) - k_n^2] u_n(z) = 0 $$

(2)

and the Green's function solution of Eq. (1) is found to be

$$ p(\mathbf{r}) = (1/4i) \sum_{n,m} \int_{-\infty}^{\infty} \int_{-\infty}^{\infty} \int_{-\infty}^{\infty} \frac{\hat{u}_n^m(k_n^m r') u_n(z') e^{i\mathbf{r} \cdot \mathbf{r}'}}{k_n^m r'} \, d^3 r' \, d^3 r $$

(3)

The integral is readily expanded in multipoles:

$$ \int d^3 r' \hat{u}_n^m(k_n^m r') u_n(z') e^{-i\mathbf{r} \cdot \mathbf{r}'} = \tilde{K}_{n,m}(r) $$

(4)

For a dipole source, one has, e.g., (in the long-wavelength limit):

$$ q_{10}(n) = \tilde{u}_n^m(z_o) n_z $$

$$ q_{11}(n) = (1/2) k_n^m u_n(z_o) (n_x + i n_y) $$

(5)

where

$$ n_j = f x_j \rho(r') d^3 r' $$

(6)

are the dipole moments of the source. In a similar fashion, the results for sources with higher multipole moments have been obtained by us. They will be shown graphically for the example of a parabolic velocity profile where analytic solutions are known (1), as well as for an arbitrary velocity profile with numerical depth functions (2).

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SOUND ABSORPTION BY BORIC ACID IN SEAWATER

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Measurements below 10 kHz with a 200 liter spherical resonator are used to determine the magnitude of the acoustic absorption due to boric acid which has been identified in earlier work as the cause of the 1 kHz acoustic relaxation observed in sea water. The temperature jump technique is used to determine the interaction of boric acid with other ionic species in sea water. The results of these measurements will be discussed.
EIGHTH INTERNATIONAL CONGRESS
ON ACOUSTICS, LONDON 1974

NORMAL-MODE THEORY IN AN OCEAN STRATIFIED IN RANGE
AND DEPTH

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INTRODUCTION

In both deep and shallow water the stratification of the medium quite
often varies significantly with range as well as depth. A computer-
oriented procedure based on normal mode theory has been developed
for problems of this type. Arbitrary changes with range in velocity
profile, water depth, and bottom composition can be accommodated.
The effect of mode conversion is included in the analysis.

THEORY

The sound field is determined by the numerical integration of the wave
equation. This is accomplished by using difference formulas in which
\( K_N \), the horizontal wave number, is varied systematically until a solu-
tion is obtained which satisfies the boundary conditions. The effect of
range-dependent stratification is determined by using equivalent verti-
cal line sources derived from the orthogonality properties of the modes.

EXPERIMENTAL RESULTS

To test the validity of the model, experimental data were considered in
a case where the stratification changed significantly with range as well
as depth. In the explosive data, the slope of propagation loss was
roughly 10 dB per 100 miles out to 300 miles. Beyond 300 miles, the
more familiar cylindrical spreading occurred. Consideration of range-
dependent stratification led to the prediction of this change in slope.
Allowance for the time history of the source was necessary to predict
the order of magnitude of the interference pattern with range.
EIGHTH INTERNATIONAL CONGRESS ON ACOUSTICS, LONDON 1974

ZUR EINSCHÄTZUNG DER VERÄNDERLICHKEIT DER SCHALLAUSBREITUNGSBEDINGUNGEN IM OZEAN

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EINLEITUNG


EXPERIMENT UND BESPRECHUNG DER ERGEBNISSE

Die Messungen wurden im tiefen Ozean in der Nähe der Schallkanalachse vorgenommen. Es wurde eine vertikale Kette von Wasserschallempfängern angewandt.

Während der Drift ändern sich die Schallausbreitungsbedingungen in einer Zeit von der Größenordnung einer Minute nur wenig. Die Signale nach flachen(\( \sim 10^\circ \)) Strahlen fluktuieren stärker, aber ihre Fluktuationen sind längs der Trasse in größeren Abständen als quer zu ihr korreliert. Die schnelle R-Abnahme längs der Trasse nach steilen(\( \sim 18^\circ \)) Strahlen hängt mit sich ändernden Mittelwerten der Ankunftszeiten zusammen. Die Längsdrift muss die Messergebnisse längs den flachen Strahlen nicht unbedingt verzerren.

\[ \begin{array}{c|c}
\hline
R & 240-340 Hz \\
\hline
0 & 5 \times 10^2 \\
1 & \times 10^1 \\
0,5 & \times 10^0 \\
\hline
\end{array} \]

\[ \begin{array}{c|c}
\hline
R & 240-340 Hz \\
\hline
0 & 5 \times 10^2 \\
1 & \times 10^1 \\
0,5 & \times 10^0 \\
\hline
\end{array} \]

Fig.1 Korrelation von auseinanderfallenden Explosionen in einem Abstand von 736 km nach flachen(a) und steilen(b) Strahlen während der Drift(*) und der Bewegung der Signalquelle längs(=) und quer(+) zur Trasse.

SCHRIFTTUM

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EIGHTH INTERNATIONAL CONGRESS ON ACOUSTICS, LONDON 1974

UNDERWATER IMPACT MEASUREMENTS

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INTRODUCTION

Underwater shock waves generated by small (0.8g) detonators were measured by two different types of Brüel & Kjaer production hydrophones. The influence of the transducer bandwidth on the results was investigated. The application of the results to underwater echo ranging as well as to analysis of marine animals echolocation signals are also considered. The repeatability of the experimental data was found to be good.

EXPERIMENTAL

The experiments were carried out in a small watertank with dimensions of 2 x 3 x 2m. The detonators employed consisted of a primary charge (0.25g) and a base charge (0.55g). The transducers used were two Brüel & Kjaer production hydrophones with sensitivities of -205 dB re 1V/µPa and -212 dB re 1V/µPa respectively. The transducer output signal was recorded directly on an oscilloscope to ensure that the transducers were the only components in the set-up that limited the frequency response. A typical recording of a shock wave measured at a range of 2m is shown in fig.1 (sweep 20 µs/div, 5V/div).

THEORETICAL CONSIDERATIONS

When using a piezoelectric transducer for shock measurements two parameters are especially important: (a) transducer size, (b) the LLF of the measuring set-up. In the first case the result of integration will introduce errors in the recording of high frequency waves. In the second case there will be an incorrect representation of the low frequency waves. According to Cole 3), the investigated shock wave can be represented by the following expression:  \[ P = P_m e^{-t/\tau}, \quad t \geq 0, \]  where \( P_m \) is the peak pressure and \( \tau \) is the time constant. \( P_m = K_1 R^{-\alpha} \) where \( R = \text{distance and } \alpha = -1.13. \)

ACKNOWLEDGEMENT

The author wishes to acknowledge the valuable assistance given by Prof. L. Bjørnø of the Department of Fluid Mechanics, the Technical University of Denmark, in the discussion and experimental work.

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SENSITIVITY AND RESONANCE FREQUENCY OF HYDROPHONES

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Sensitive Hydrophones are generally large. Therefore their frequency of mechanical resonance is low. In the design of a hydrophone the requirement for a high resonance frequency conflicts with the desirability for a high sensitivity.

Piezoelectric hydrophones can be designed as hollow spherical or cylindrical elements, the latter without or with endcaps, or with only exposed pistons, as depicted in the figures below:

The active elements are made of a modern piezoelectric ceramic material (lead-zirconate-titanate compositions). With the known mechanical and piezoelectrical constants of this material it is possible to calculate the resonance frequency and the pressure-sensitivity at lower frequencies. The product of these values ranges from 5 to 10 V/Pa.s for spherical and end-capped cylindrical elements, up to 100 V/Pa.s for elements with pistons.

Hence a spherical or cylindrical hydrophone with a required sensitivity of $10^{-3}$ V/Pa should have a resonance frequency lower than 10 kHz, for which the necessary diameter would be of the order of 20 cm. A piston-hydrophone with the same sensitivity may be designed with a resonance frequency of up to 100 kHz, with a diameter of only 1 cm.

It will be shown that a sensitivity of $10^{-3}$ V/Pa is needed in order to observe the ambient noise in the sea during extreme calm weather.

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EIGHTH INTERNATIONAL CONGRESS ON ACOUSTICS, LONDON 1974

APPLICATION OF A SLOW WAVEGUIDE TO A FREE-FLOODED RING TRANSDUCER

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Free-flooded ring transducers simplify design of deep submergence sonar arrays. However, the directional response, omnidirectional in the plane of the ring, is not ideal for a directional source. If a slow waveguide (a cylinder of silicone rubber) is placed coaxially to and at one side of the ring, the radiation from the inner ring surface is delayed so that it is approximately in phase with the radiation from the outer ring surface at the far end of the waveguide. Thus, the free-flooded ring with a coaxial slow waveguide produces a unidirectional source, as shown in the left half of Fig. 1, in contrast to the radiation from the ring by itself (right half of Fig. 1).

The slow waveguide free-flooded ring combination is mathematically modeled using coupled surface Helmholtz integral equations. One for inside the waveguide, using the density, sound speed and wave number for the waveguide material, relates the pressure on the interior surface of the waveguide to the normal velocity of its surface. The other for the external medium, using the density, sound speed and wave number for this medium, relates the surface pressure and normal velocity on the composite surface consisting of the surface of the ring and the exterior surface of the waveguide. These two integral equations are coupled by requiring that pressure and normal velocity be continuous across the surface of the waveguide. Given the normal velocity of the ring, the system is completely determined. The coupled integral equations are solved numerically by a finite element technique using computer algorithms developed at NRL which are suitable for this axis symmetric geometry (1).

Measurements have shown that the application of a slow waveguide to the free-flooded magnetostrictive ring transducer increases the maximum source level by 4 dB at 6 kHz, 7-1/2 dB at 7 and 8 kHz without changing efficiency.

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Fig. 1 Directional response of free-flooded ring (right) and free-flooded ring with slow waveguide (left).
LOW-NOISE HYDROPHONE FOR ALL UNDERWATER OCEAN ENVIRONMENTS

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INTRODUCTION

The accuracy of underwater acoustic measurements depends on many factors including the stability and acoustic characteristics of the measuring hydrophone. A hydrophone has been developed for use as a primary or secondary reference standard under widely varying environmental conditions. The free-field voltage sensitivity changes by less than ±0.2 dB in the frequency range 10 Hz to 30 kHz when the hydrophone is exposed to hydrostatic pressure of 69 MPa (6900 m depth) and to temperatures between 3° and 30°C. The sensitivity is constant within ±1.5 dB to frequencies as high as 100 kHz. The size of the piezoelectric sensor has been selected to provide an optimum sensitivity, directivity, and source impedance for many calibration and acoustic measurement requirements.

DESIGN PARAMETERS

Lithium sulfate piezoelectric crystal was selected for the sensor element because of its stable characteristics and sensitivity as a volume expander. The receiving sensitivity has been computed to a high degree of accuracy, especially in the frequency range where the sensor element is much smaller than a wavelength. Unlike ferroelectric ceramics, lithium sulfate does not undergo an aging effect to degrade its electromechanical characteristics. An acoustic window of low water permeability maintains the high resistance across the crystal electrodes to ensure a long RC time constant and low cutoff frequency.

Self noise and preamplifier characteristics have been determined by computer using a noise model of the sensor and amplifier. Comparison of computed and measured data verifies the model and permits selection of values to optimize the desired characteristics. Typical equivalent noise pressure levels have been compared to the level of minimum noise as given by Wenz and to Knudsen's sea state zero.

The preamplifier with an output impedance of 5 ohms will drive a 7000-m cable and has a dynamic range adequate for sound pressure levels as high as 190 dB re 1 µPa. All metal components are isolated from the water by an elastomeric covering that prevents corrosion and reduces the problem of electrical ground loops.

CONCLUSION

A wide frequency band hydrophone that is omnidirectional over a portion of this band has been designed with sensitivity and noise characteristics suitable for measurements at sea under most environmental conditions and noise levels that will be encountered.
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A DISTRIBUTED PRESSURE TRANSUDER WITH SPATIALLY STEERABLE RESPONSE OBTAINED BY ELECTROMAGNETIC SAMPLING

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INTRODUCTION

A novel acoustic line pressure transducer concept is presented that possesses uniform spatial sensitivity and surface impedance in conjunction with steerable spatial response. It is envisioned as an alternative to line arrays, offering inherently superior flow noise rejection as well as simplified signal transmission requirements.

DESCRIPTION AND THEORY OF OPERATION

The transducer schematically depicted in Fig. 1 has the configuration of a terminated strip transmission line consisting of a rigid and a flexible conductor. Local fluctuation of the characteristic impedance are induced by the impinging acoustic pressure field described in general form as a function of wave vector \( k \) and angular frequency \( \omega \) as

\[
P(k, \omega) = \sum_n P_n \exp(i(k \cdot x - \omega t))
\]

An electromagnetic pulse (approximated by a delta function \( \delta(t_O) \)) applied to the terminals travels along this transmission line sampling the local fluctuations of the characteristic impedance and hence, the acoustic pressure field on the surface of the transducer. A return signal is generated by scattering, of the form

\[
E(t-t_O) = (d/\rho c^2) \sum_n P_n (k_n \cdot x/k_n x) \exp(\frac{i}{2 \pi c} k_n \cdot x/ck_n x - 1) t
\]

where \( c \) is the propagation speed of the acoustic field, \( c' \) that of the electromagnetic pulse in the transmission line, \( \rho \) the density of the medium and \( d \) represents the sensitivity. This signal contains the wave vectors and frequencies of the acoustic pressure field in its high and low frequency components respectively. Steering of the transducer response can be accomplished by repeated application of the sampling pulse and frequency-domain filtering.

EXPERIMENTS

A transducer based on this concept was realized for operation in air, and was evaluated by means of a time-domain reflectometer. The results confirm the feasibility of the concept. These results indicate sensitivities comparable to those of electrostatic microphones.
Prestressed Sandwich Transducer Analysis by Admittance Techniques

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Introductory Analysis

When prestress is applied to ceramic transducers their mechanical, dielectric and electromechanical properties change. These changes are also observed when the prestress is removed. The ultimate values of the physical parameters of the transducer are difficult to know. The prestressing effect is in fact manifested by an aging process. Some of these effects have been discussed in several papers (1,2).

We discuss here the time dependency of several transducer properties characterizing the equivalent circuit of the transducer. The data was obtained by means of the admittance circle diagram, and the transducers, of sandwich type, were designed following a previous theoretical model (3,4,5).

EXPERIMENTAL RESULTS

Fig. 1 shows the aging process after removing the prestress on one of the many experimented transducers. It can be seen that the inverse of the circle diagram clearly diminish, showing the tendency to reach the $R_m$ value showed before the prestress was applied. The $Q$ factor also increases with time to the value before prestressing. Lastly in Fig. 1, the clamped capacitance $C_o$ is shown to increase greatly and after several minutes tends to remain constant with a final light descent. Among other parameters investigated were also the series resonance capacitance $C_m$, and the coupling coefficient $k_t$.

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EIGHTH INTERNATIONAL CONGRESS ON ACOUSTICS, LONDON 1974

THEORY OF AN ACOUSTIC, SPHERICAL, COMPLIANT TUBE, LUNEBOURG LENS

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A Luneburg lens (1) is a nonhomogeneous, spherically symmetric, refracting structure which focuses a parallel beam of rays exactly at a point on the opposite surface from which the rays enter. One method of achieving a variable index of refraction is to use a mixture of compliant tubes and water (2). A compliant tube is a hollow, flattened tube made of aluminum or plastic with a cross-section that is small compared to a wavelength and a compressibility greater than that of water. By varying the spacing between tubes, the index can be made to vary throughout the lens. If the maximum tube spacing is small compared to a wavelength, a continuously variable index is approximated extremely well. Since a mixture of compliant tubes and water is dispersive, i.e., the index of refraction is frequency dependent, the compliant tube lens can be designed to be a Luneburg lens only at a single frequency called the design frequency. When operated above the design frequency the compliant tube lens is a perfect focusing Gutman lens (2) with an interior focal point dependent on the frequency. When operated below design frequency, the lens is not perfect focusing and the frequency dependent focal region is exterior to the lens. However, the spherical aberration is so slight that it results in negligible degradation.

Using the frequency dependent index of refraction the wave equation for the acoustic pressure was solved for plane wave incidence. This solution corresponds to measurements made with an omni hydrophone on the lens. The theory was further developed to include solutions corresponding to measurements made by a dipole and cardioid hydrophone. For these three types of hydrophones the on-axis pressure gain, directivity index and beam patterns were studied as a function of frequency and distance from the center of the lens.

Numerical calculations were made for a 10 foot diameter compliant tube lens with a design frequency of 5000 Hz. This corresponds to a lens built by the Automatics Division of Rockwell International. Fig. 1 shows a comparison of theoretical and measured results.

REFERENCES

LOW SIDELobe RESPONSE BY COMBINING HYDROPHONES ON AN ACOUSTIC LUNEBUG LENS

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INTRODUCTION

Reduced sidelobe performance of an acoustic Luneburg Lens can be achieved by employing directive hydrophones on the lens' focal surface. This paper describes a technique for further reducing side-lobes by combining the outputs from several appropriately weighted elements.

DISCUSSION

In a multiple beam lens system, hydrophones are normally spaced on the focal surface so that adjacent beams cross over at or near their half power (-3 dB) points, i.e., approximately λ/2 at the highest operating frequency, f_h (λ = wavelength). There are many ways for arranging the hydrophones on the lens. The chosen geometry is formed by placing hydrophones at the vertices of a hexagon with a seventh hydrophone at the center. This pattern is repeated over the surface of the lens so that every hydrophone is surrounded by a hexagonal 'ring' of hydrophones. (This geometry can only be approximated on a spherical surface). The 6 elements surrounding a central element are summed, through an appropriate weighting resistor, to the central element; thus, the output, for each beam position, is the weighted sum of the 7 elements. This configuration produces a symmetric displacement of low sidelobes and adjacent beams.

Broadband operation may be obtained in a number of ways. In this study, rings of elements successively displaced from a central element by mλ_h/2, (m = 1, 2, 4, 8 ...) are used over an octave bandwidth, (λ_h = wavelength at f_h). The summing is done in an operational amplifier, thus the outputs are decoupled from the inputs. The weighting factor chosen (.35 relative to unity for the central element) was a compromise between main beam broadening and reduced sidelobes. Calculated patterns for the weighted sum of 7 cardioid hydrophones on a 10' diameter lens are shown in Fig. 1, along with corresponding patterns for a single cardioid. (Unpublished calculations by C. A. Boyles, 1972). These results, which are in excellent agreement with measurements, illustrate the dramatic reduction in sidelobes that can be achieved using this technique.
The paper presents a new method of the measurement of Doppler frequency shift by means of the digital processing technique, applied to the sonar signals.

The concept of the method lies in the simultaneous measurement and comparison of the high frequency time period both for sounding pulse: $T_s = 1/f_s$ and echo: $T_e = 1/f_e$.

There are two pulse formers generating the time gates controlled by the corresponding time periods: $T_s$ and $T_e$.

The two pulse counters, controlled by these gates, count the number of the clock pulses for both signals.

The difference of the output states of the counters so obtained, represents the difference of the time periods for both signals, which is evidently proportional to the Doppler frequency shift of echo signal.

The decoder translates Doppler frequency shift to the velocity units and outputs it to the digital velocity indicator.

The advantages and the shortcomings of the proposed technique are also discussed in comparing with the existing solutions on the state of the art.

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EIGHTH INTERNATIONAL CONGRESS ON ACOUSTICS, LONDON 1974

LONG-RANGE SONAR STUDIES - A FIVE-DAY RECORD OF AN EXTENSIVE CONCENTRATION OF FISH

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The purpose here is first to introduce some long-range sonar studies of reverberation structure, and second to illustrate them by some remarkable records for a particular five-day period. The early work is described in (1). By choosing a low carrier frequency, where sound propagation is good, it is possible to achieve long ranges. By using a fixed system, with transducers laid on the sea-bed in the shallow waters of the Bristol Channel, it is possible to monitor the structure in a given receiver beam or beams over very long periods.

For the period 15-20 May 1967 long FM pulses of mean frequency 2 kHz and bandwidth 100 Hz were projected into a 15° sector, and the echoes received in a 2° beam and displayed after correlation processing. The usual features were seen, the most obvious being the meandering discrete tracks due to shoals of pilchard, though these disappear at night. There are strong echoes at various fixed ranges, due to bottom roughnesses. There are also patterns due to the interference of the various normal modes.

The outstanding and new feature in this period is the strong return that starts at a range varying with time from 30 to 46 km, so strong that the action of the automatic gain control removes all traces of the pilchard shoals. The fine patterning of the echo region suggests returns from a large number of discrete scatterers, such as shoals of fish, and there is also a coarse patterning due to propagation effects as well as variations in concentration. The most surprising aspect of the return is its variation in range, which is diurnal rather than tidal. Thus for some unknown reason the leading edge of the return closes range near dusk, with a velocity that can reach 1 m/s or 2 knots. The five-day record is certainly unusual, and in monitoring periods just before and just after the present one there are similarities to it but only a suggestion of its grandeur. In June 1967 in work in cooperation with the Lowestoft Fisheries Laboratory some diffuse echo traces were identified as due to sprats, and 1 m/s is about the cruising speed of sprats. The present strong returns are certainly due to fish, and very likely to large numbers of medium size sprat shoals.

The five-day period also shows at least three other special features. The first is a lack of returns beyond 7 km near midday on 19 May, which is due to 20 m/s winds. The second is a rapid change in the interference pattern, best seen near dawn on 16 May, which comes about when the fish change depth and alter the phase of the pattern. Third it is possible to follow the breaking up of the pilchard shoals near dusk - they grow progressively fainter and wider, in a process akin to diffusion, till at the point of disappearance they approach 1 km in width.

REFERENCE
EIGHTH INTERNATIONAL CONGRESS ON ACOUSTICS, LONDON 1974

A STUDY OF THE PROPERTIES OF THE OUTPUTS OF A MULTI-ELEMENT SONAR RECEIVING ARRAY OPERATING IN A REVERBERANT ENVIRONMENT

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INTRODUCTION

An experimental and theoretical study was made of the properties of the outputs of a six element linear receiving array. The emphasis of the study was on the properties of the covariance matrix of the receiving array; i.e., on the combined spatial-temporal properties of the received reverberation signals.

EXPERIMENTAL

The experimental data were measurements of the outputs of the six receiving elements resulting from the scattering of 100 µsec (8 cycles at 80 kHz) CW pulses from the wind-driven surface of a fresh water lake. Measurements from a total of 100 different reverberation events were obtained. The receiver outputs were tested for independence, homogeneity, and normality at 900 different times (after transmission) throughout the reverberation process. Also, the sample estimate of the mean of each receiver output was tested to determine if the mean output was different from zero. The receiver outputs resulting from the surface reverberation appeared to have been drawn from a zero mean Gaussian distribution.

Sample estimates of the covariance matrix of the receiver outputs were obtained using the 100 element sample ensemble. The correlation time of the covariance matrix was found to be approximately 125 µsec. Although the autocovariance functions and the crosscovariance functions had the same correlation time, the detailed structure of the different covariance functions was not identical. The correlation distance was about 7λ. The decay of the spatial correlation coefficient with separation distance was monotonic rather than oscillatory. Over the observed time interval of the reverberation process the maximum interelement coherence was approximately constant. However, this maximum did not generally occur for t=0. And, further, the detailed structure of the crosscovariance functions was dependent on the time (after transmission) that the receiver outputs were observed. Finally, it was found that increasing horizontal directivity while maintaining constant separation increased interelement coherence.

THEORETICAL

A theoretical estimate of the covariance matrix of the receiver outputs was obtained using the discrete scatterer approach (1). Quantitative results for the predictions of this model were obtained using a digital computer. Where comparisons were made, the corresponding sample and theoretical estimates were in generally good agreement.

REFERENCES

OSCILLATIONS OF NONSPHERICAL BUBBLES

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ABSTRACT

The oscillation of nonspherical bubbles is studied based on the general variational formulation in terms of both Eulerian and Lagrangian coordinates. (1)(2) The variational formulation of the general dynamical problem of two fluids separated by an interface in terms of Eulerian coordinates is first presented. From this formulation, a set of approximate wave equations are derived and the interactions between the various surface wave modes and the spherical mode are discussed. Then the general dynamic problem and the variational formulation is presented in terms of Lagrangian coordinates. The Lagrangian formulation is more suitable to deal with the breaking of the bubble. For small nonspherical oscillation, the results from the analyses by spherical harmonic expansion are the same for both the Eulerian and Lagrangian formulations. However, the extrapolations to finite amplitude oscillation from the results of small oscillation differ for these two formulations. The application of the variational formulation in terms of Lagrangian coordinates also lead to interesting mathematical problems in the calculus of variation.

REFERENCES


INTERACTIONS BETWEEN GAS BUBBLES IN A SOUND FIELD

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In typical cavitation fields many bubbles are present and the behavior of any one bubble is influenced by the presence of others. There is some evidence that interactions are important in sonic processes. For example, Howkins (1) described a cavitation "ghost" structure which is important for sonic erosion of solids while Temple et al (2) found a cavitation "shuttlecock" to be involved in sonoluminescence. Both the ghost and the shuttlecock appear to be clouds of rapidly moving bubbles of varying sizes.

Insight on the altered response of a bubble when in the presence of others is gained by investigating a pair of bubbles, A and B. Linear theory is used in which each member of the pair is assumed driven by a sound field consisting of (1) an ambient field of angular frequency \( \omega \) and (2) a field scattered by the other member. Calculations based on this theory have been carried out for the volume amplitude of each bubble as a function of frequency for various choices of parameters. For some conditions a given bubble A exhibits two maxima in its volume amplitude. If it is appreciably smaller than B and if the A-B separation is large the primary maximum for A is in the vicinity of its own (noninteracting) resonance frequency \( \omega \), and the other maximum, much reduced in amplitude, is approximately at the noninteracting resonance frequency \( \omega' \) of B. As the separation decreases the response curve may be greatly altered. For example, if the bubbles are identical, each with noninteracting resonance frequency \( \omega \), there is only one maximum. This maximum occurs at a frequency which falls below \( \omega \) by an amount which increases with decreasing separation.

Details will be presented on theoretical response curves for pairs and, in addition, on related experimental findings at low frequencies. Results will also be discussed for attractive and repulsive ("Bjerknes") forces between bubbles. Examples of interaction phenomena will be shown in a film.

REFERENCES

A two-dimensional fish stock distribution can be obtained by measuring hydroacoustically /e.g. by an echo integration technique/ the average fish density q along the ship's track. In a previous paper /1/ we have proposed "the thin layers method" allowing to obtain an overall spatial distribution of fish stock by determining depthwise fish density profiles.

THE SYSTEM DESCRIPTION

The authors developed the RYPOL computerized system which utilizes the thin layers method, and now is operating on board R.V. "Profesor Siedlecki". The system incorporates the following basic units: the SIMRAD BK 38 echo sounder, the SIMRAD QM echo integrator, and an ELLIOTT 905 digital computer.

The computer controls on-line sampling of the integrator output at 1 ms intervals for every echo sounder transmission. In such a way thin layers are formed. Increments of integrated echo signal, obtained in the computer by subtracting values of adjacent samples, are proportional to the volume back scattering strength /v.b.s.s./ of fish within the layers. After processing the sampled data in real time, the depthwise fish density profiles /in terms of v.b.s.s. profiles/ are obtained, and output to a digital plotter every 0.2 Nm of ship's track.

RESULTS

An example of fish density profiles obtained from the system on board the vessel, during its survey cruise on the SW African fishing grounds, is shown on Fig.1.

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EIGHTH INTERNATIONAL CONGRESS
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BEHAVIOUR OF SINGLE CAVITATION CENTRES IN WATER AT 1 MHz

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INTRODUCTION

Photographic studies by Willard (1) of cavitation in a focused 2.5 MHz field in water showed that there were two fundamental processes involved when cavitation occurred at the focus. The "initiation phase" lasted for hundreds of microseconds and was followed by a "catastrophic phase" of millisecond duration. During the latter phase the cavitation events appeared as microbubble clouds of a size which varied cyclically at a frequency of about one thousandth of the fundamental frequency. Recently we have detected similar events acoustically in a 1 MHz focussed field (2). In a study of sonochemical activity sources of chemical action were observed both at the focus and at other points in the field. The non-focal chemical activity was observed coming from fixed points in the sound field. These origins of sonochemical activity were stable for as long as 20 sec (3).

EXPERIMENTAL

Measurements of sound emission (subharmonic and harmonic) and white noise by a hydrophone placed in the cavitating field under experimental conditions where focal events are rare and the position of non-focal chemically active cavitation can be located by its visible chemical products will be reported.

DISCUSSION

Previously it has been shown that focal and non-focal cavitation centres have broadly similar sonochemical yields (3), lyse similar volumes of amoebae and erythrocytes per unit time (4, 3), and modulate the envelope of the voltage applied to the focused bowl at similar frequencies (2). These points suggest that the focal and non focal events may be similar in nature. The relevance of the above measurements to the understanding of non-focal cavitation will be discussed. The stabilization of the non-focal cavitation origins in space will be examined in the light of recent work on the behaviour of bubbles in standing wave fields at lower frequencies (5).

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HOLOGRAPHIC CINEMATOGRAPHY OF CAVITATION BUBBLE FIELDS

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In cavitation research the mutual influence of individual bubbles within a large ensemble of cavitation bubbles is of much interest. High-speed investigations are confronted with several problems: the bubbles are quite small, they are distributed over a volume much larger than their size, and they are subject to fast motions.

EXPERIMENTAL METHOD

High-speed holography is employed to study fields of cavitation bubbles. To investigate the dynamic behaviour of the field a series of holograms is recorded one shortly after the other. A Q-switched ruby laser is driven to produce up to four light pulses of some 100 nsec separation and 20 nsec duration. They are used to expose the holograms one upon the other on a photographic plate. For identification of bubbles the reconstructed images from the holograms can be separated since the reference light beam is switched in its direction between the exposures. This is achieved by an appropriate arrangement of acousto-optic deflection elements. Information on the three-dimensional configuration of bubbles at the times of the exposures can be drawn from the reconstructed images which may contain many hundred bubbles.

RESULTS

Prior experiments on acoustically produced cavitation fields with two light pulses only (1) have shown that the field is highly dynamic. The extended investigations give information on movement distributions in the field, quasi-stable overall bubble configurations (streamers), bubble annihilation, and production of cavitation nuclei. Theoretical considerations concerning the forces governing the observed processes are stimulated. Studies of more simple bubble fields are encouraged, consisting of less bubbles which are, for example, produced by focussing laser light into the fluid (2).

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EIGHTH INTERNATIONAL CONGRESS
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ACOUSTICAL AND OPTICAL CAVITATION THRESHOLD MEASUREMENTS

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EXPERIMENTAL SETUP

An apparatus has been built allowing the simultaneous measurement of acoustic cavitation thresholds in water by acoustical and optical means. The strong sound field is obtained in a hollow water-filled piezoelectric cylinder driven at its fundamental mode (about 16 kHz, depending on the additional container). The pressure in the cylindrical volume has been measured as a function of the applied voltage with a calibrated microphone up to about 1 bar, where cavitation starts at the surface of the microphone. A strongly linear relationship has been found over the three decades measured. Thus absolute pressure values for the cavitation thresholds can be given.

The detection of the first subharmonic (of the driving frequency) in the spectrum of the cavitation noise is taken as the acoustical indicator of the onset of cavitation. Some other spectral lines and frequency bands are also tried. The light of a He-Ne-laser scattered by the bubbles formed when cavitation sets in is used as the optical indicator. The advantage of the optical method is that the frequency around the driving frequency need not be excluded. But despite this fact the onset is not as sharp as the onset of the first subharmonic.

RESULTS

First results indicate that both methods lead to about the same threshold values. The following results were obtained with the first subharmonic as indicator. The threshold (incipient and desinent) of a strongly degassed water sample has been measured in dependence on the static pressure between 0 and 1 bar when the acoustic pressure is changed linearly with time. The cylinder has not been cleaned by rinsing for this measurement so that nuclei with diameters > 1 μm certainly were present. As the incipient threshold is much more repeatable than the desinent one and moreover shows a linear dependence on the external static pressure, it has been taken for further measurements with added impurities (spherical particles with diameters of 0.35 μm and 1.2 μm). These investigations show that gas content of the water and solid particles act in a combined manner. At high gas content the nuclei size seems to be of no importance whereas at low gas content (100 mbar partial gas pressure) the particle size has a strong influence on the threshold.
EIGHTH INTERNATIONAL CONGRESS ON ACOUSTICS, LONDON 1974

THE ANGULAR RESOLUTION AND PROPERTIES OF A PHASE-SENSITIVE SONAR SYSTEM

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GENERAL PRINCIPLE

Phase-sensitive systems are widely used in radionavigation and known for their accuracy.

A similar idea may be adopted for measurement of signal direction within a directive beam. The phase of a received signal changes at $2\pi$ between the minima of the beam. The actual direction of the received signal may be thus determined. The system is characterized by its angular sensitivity i.e. the number of electrical degrees per one degree of arc. A good angular accuracy may be obtained even if the accuracy of the phase measurement is not very high.

SIGNAL PROCESSING

For a B-scope display an angular time base must be generated. An example of the processing circuit is shown on Fig. 1. At the input terminals the signals received by a pair of transducers are applied and in balanced modulation circuit the phase difference $\phi$ is transferred to the modulation envelope:

$$U_i = U \cos(\omega t + \frac{\phi}{2}) \sin(\omega t + \frac{\phi}{2})$$

where $\omega$ and $\rho$ are signal and modulating angular frequencies, respectively \cite{1}.

ADVANTAGES AND SHORTCOMINGS

The system provides a monopulse observation of a required sector with a transducer array considerably smaller than that needed in amplitude, narrow beam scanning.

The system is therefore suitable for low frequency long range sonar. It is convenient and accurate for detection of single targets. When the target is space distributed as e.g. a fish shoal, the resultant phase only is measured and it indicates a kind of "acoustic centre of gravity". Strong reverberation deteriorates the accuracy.

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EIGHTH INTERNATIONAL CONGRESS ON ACOUSTICS, LONDON 1974

ACOUSTIC VOLUME SCATTERING MEASUREMENTS IN THE CHUKCHI SEA

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INTRODUCTION

Measurements of acoustic volume scattering strengths (AVSS) in the marginal ice zone of the Chukchi Sea were made during the summers of 1972 and 1973. Some results of the 1972 measurements and a description of the experimental arrangement have been reported previously (1).

EXPERIMENTAL

In 1972 AVSS were measured at 105 kHz and 38 kHz with pulsed, downward-looking echo sounders from the icebreaker USCGC STATEN ISLAND. Helicopters were used for some of the 105 kHz measurements. Biological samples were taken simultaneously in an effort to identify the scatterers. In 1973 similar acoustic measurements were made from the icebreaker USCGC BURTON ISLAND, and measurements at 60 kHz were made by helicopter, either from a hover or by landing on an ice floe.

System self-noise was such that AVSS above -85 dB ref. 1 m⁻¹ could be detected. The echoes (heterodyned down to 5 or 10 kHz) were recorded on magnetic tape and on echogram charts. The magnetic tapes were subsequently analyzed by averaging over many pulses to yield the average AVSS in up to ten depth intervals which were selected in each case by inspection of the echogram for the specific location and frequency. There are 179 such data files from 1972 and 126 from 1973.

RESULTS

Numerous scattering layers and individuals were observed at all depths, but local variations were great. AVSS as high as -40 dB were observed at 105 kHz. Typical layers were found to have AVSS of -50 to -70 dB. Measurements at 105 kHz and 38 kHz, taken at the same location, sometimes were very comparable but often were not. Small targets (in the Rayleigh scattering limit) are expected to have an AVSS about 18 dB larger at 105 kHz than at 38 kHz. Some of the data show approximately such a difference while some, presumably from larger targets, do not. Measurement results under heavy ice cover generally were lower than in open water. Especially high AVSS were observed in warm surface intrusions of Pacific water. Euphausiids typically have a backscattering cross section of 1.4·10⁻⁴ cm² at 102 kHz (2). A layer with an AVSS at 105 kHz of -50 dB ref. 1 m⁻¹ would, if composed mainly of euphausiids, imply a density of about 600 euphausiids per cubic meter.

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PROPRIÉTÉS STATISTIQUES DE LA RÉFLEXION SOUS-MARINE DE SURFACE

Gazanhes C
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INTRODUCTION

Afin d'étudier la diffusion d'une onde sonore par la surface de la mer, on a transposé le problème dans la gamme ultrasonore, sur un modèle réduit, en cuve acoustique. On s'intéresse aux fluctuations d'amplitude d'un signal de 150 kHz. (1)

RESULTATS EXPERIMENTAUX

Suivant la direction spéculaire, on analyse au moyen d'un ensemble de traitement statistique Didac 800, les fluctuations de l'enveloppe du signal diffusé. On obtient des densités de probabilité (d.d.p.) qui permettent le calcul de la valeur moyenne et de l'écart type des fluctuations. La fig.1 représente la variation de l'écart type en fonction du paramètre de Rayleigh.

Fig.1 Ecart type du signal diffusé

INTERPRETATION DES RESULTATS

On montre que le signal diffusé comprend un terme cohérent assimilé au signal utile et un terme incohérent assimilé à un bruit. L'évolution des d.d.p. du signal diffusé s'explique bien par la loi de Beckman (2).

Partant des résultats de C. Eckart (3), on écrit le terme incohérent :

\[ \sigma_d^2 = K^2 \iint J(x,y) \exp \left[ +c^2 \sigma^2 s^2 (C(x,y) - 1) \right] dx dy \]

Où J(x,y) traduit la répartition de l'énergie acoustique sur la surface et c²C(x,y) est la fonction de corrélation spatiale de la surface. Les résultats numériques et expérimentaux sont en bon accord.

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(3) C. Eckart, J. A. S. A. (1953) 25 3,566
Recent measurements of volume scattering (1) in widely separated Pacific and Indian Ocean areas show a striking similarity in their dependence on frequency in the range 1 kHz to 20 kHz. Typically, volume scattering strength increases at nearly 12 dB per octave in the 1-6 kHz range and is fairly constant thereafter. We believe this frequency pattern is due largely to the presence of various gasbladder fishes, members of the Myctophidae, Sturnophyidae, Gonostomatidae and Melamphaidae families. Bladder sizes for these mesopelagic fishes are relatively small; myctophids, the most numerous in species and number of individuals, have bladders in the range 0.5 to 5.0 mm. The large bladders make the chief contribution to the scattering at the lower frequencies where, below resonance, the increase in strength is relatively rapid. The fairly constant strength above 6 kHz is the combined effect of the scattering from bladders of decreasing size which are resonant at successively higher frequencies.

Calculations to test our hypothesis were made for a fictitious assemblage of fish of plausible size and depth distribution. Eqn. (1) gives the average volume scattering strength of fish in a unit volume over a depth interval (b-a), as a function of bladder radius R, frequency f, resonance frequency f₀, quality factor Q and n(z), the number of individuals per cubic meter at the depth z.

\[ S_v = 10 \log \left[ \frac{R^2}{(b-a)} \int_a^b \left[ \left( \frac{f}{f_0} \right)^2 - 1 \right]^2 + \frac{1}{Q^2} \right]^{-1} n(z) dz \]  

The integrands of Eqn. (1) were summed for each bladder size and frequency considered. The resulting volume scattering strengths produce the expected frequency pattern and thus support the hypothesis.

The results of a realistic test of the hypothesis are shown in Fig. 1. In this case the bladder sizes and their depth distribution were determined from net hauls made in a deep-water area off Baja California; the calculated results using these net-haul data are shown in the continuous curve labeled Constant Volume. The measured scattering strengths obtained acoustically at five discrete frequencies in this same deep-water area are also plotted in Fig. 1, with agreement and further support to our contentions.

INTRODUCTION

The nonlinear effects, taking place in the course of propagation, multiple reflection, and interaction of the pulses caused in layered half-space by a surface input are analysed with paying the main attention to the echo-pulses from the layers \( n = 1, n = 2 \) at \( X = 0 \) (see Fig. 1).

STATEMENT OF THE PROBLEM

Let the prime denote a derivative with respect to coordinate \( X \), the dot a derivative with respect to time \( t \) and \( H(t) \) the Heaviside function. The pulses propagation in layer \( n \) is described by the equation

\[
c_n^{\prime\prime\prime}(X,t) - q_n(U'U''(X,t)) = 0
\]

where \( c_n = \text{const} \), \( q_n(U') = 1 + a_n U' + b_n(U')^2 + \ldots \) Fig. 1 Layered half-space

Two types of input are considered

\[
U'(0,t) = 0 \quad \text{or} \quad U'(0,t) = \varepsilon \gamma(t) [H(t) - H(t-t_0)]
\]

where \( t_0 \ll c_n^{\prime\prime\prime} \) and \( \gamma(t) \) is an arbitrary continuous function which satisfies the following conditions

\[
\gamma(0) = 0, \quad \gamma'(0) = 0, \quad \gamma(t_0) = 0, \quad \gamma''(t_0) = 0
\]

Definitions:

\[
\varphi_n \approx \frac{\varepsilon}{2} c_n^{\prime\prime\prime} \max \gamma(t), \quad \varphi_n' \approx \frac{\varepsilon}{4} c_n^{\prime\prime\prime} \max \gamma(t) \int_0^{t_0} \gamma(z) dz
\]

RESULTS

By making use of the method suggested in (1), the analytical solution is obtained. From it follows: (a) neglecting the effect of interaction of two pulses in layer \( n \) leads to the asymptotic error \( \varphi_n' \), (b) if the propagation of a pulse through the layer \( n \) is considered then the asymptotic error of the linear solution is equal to \( \varphi_n' \), (c) the asymptotic error of the linear approximation of the first echo-pulse at \( X = 0 \) lies between the limits \( 2 \varphi_n' \) and \( \varphi_n' \) which consequently correspond to the special cases of rigid and free surfaces at \( X = h_0 \).

REFERENCE

CONSIDERATION ON SAMPLE SIZE FOR MEASURING ULTRASONIC REFLECTIVITY OF SAND SUSPENDED IN WATER

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INTRODUCTION

Ultrasound reflectivity of sand suspended in water is one of the most important data for designing such a system as an ultrasonic suspended sand density meter. However, a change of the distributive situation of suspended sand causes a large fluctuation in sound pressure level received by a receiver due to the interference of ultrasound. Therefore, when determining the reflectivity by the mean power method, the dependance of sample size on reliability is discussed here.

EXPERIMENT

Ultrasound is emitted by a disk transmitter driven at the frequency of 1 MHz, is reflected by a sample of suspended sand fixed with gelatin located at the center of a water bath, and is received by a receiver. The angle of the receiver from the transmitter at the center of the sample is held at 90°. The amplitude of the received signal is measured at every degree of the rotation of the sample. Fig.1 shows an example of the results.

THEORETICAL ANALYSIS

The reflective particles are regarded as an assembly of pairs of small volume elements located symmetrically to the center line of the sample. Statistical analysis leads to the result that the amplitude of received signal is approximately in Rayleigh's distribution. Theoretical and experimental standard deviation of rms values of received sound pressure in small sample sizes from one in a very large sample size is shown in Fig.2. It can be seen that ultrasound reflectivity from the sample size of 36 is within 10% error.

Fig.1 An example of measured reflection.

Fig.2 Standard deviation of rms values of sound pressure obtained by small sample sizes from that obtained by a very large.
THE UNDERWATER SOUND REFLECTION OFF A TRIPLANE

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INTRODUCTION
For training purposes and ranging with active sonar equipment triplanes are used besides echo repeaters and transponders. Already during World War II the most interesting characteristics of these reflectors have been measured, namely the target strength and its dependence on the orientation of the triplane to the axis projector – reflector (1). Modified construction principles and materials as well as varied frequency range induced new measurements.

DESIGN CONSIDERATIONS
Optimally smoothed out orientation dependence is wanted because of the unavoidable rotations of a triplane round the axis marker buoy – ground anchor. We found best geometry by shifting the lower half of the reflector against the upper one by an angle of $45^\circ$ (Fig. 1). The principle of pressure release has been used for reflection either by an airgap included between metal sheets or by hard foam plastic planes.

EXPERIMENTAL
A projector transmitted pulses of $16/f \text{ ms}$ length ($f$ in kHz), the distance to the reflector being 15 or 30 m at a depth of 5 to 6 m below surface. One calibrated hydrophone was located at the projector, measuring EL, a second one at the triplane, measuring (SL – TL). For evaluation

$$TS = EL - (SL - TL) + TL \quad \text{in dB}$$

was used. Notations see (2). Spherical spreading was present. The results were:

<table>
<thead>
<tr>
<th>Material</th>
<th>Diameter</th>
<th>TS in dB</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>$m$</td>
<td>$8$</td>
</tr>
<tr>
<td>hard foam plastic</td>
<td>1.6</td>
<td>-</td>
</tr>
<tr>
<td>airgap between iron</td>
<td>1.1</td>
<td>+8</td>
</tr>
<tr>
<td>metal sheets aluminium</td>
<td>1.1</td>
<td>-</td>
</tr>
<tr>
<td></td>
<td></td>
<td>+4</td>
</tr>
</tbody>
</table>

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A REVERBERATION TANK FOR HYDROACOUSTIC MEASUREMENTS

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The reverberation chamber methods of air-borne acoustics for the measurement of absorption coefficients and of the total power radiated are applicable to water-borne sound, too. An early construction of a suitable tank has been described in (1). An irregularly shaped container with a capacity of about 2 m³ was lined with plates of polystyrene foam, acting as a sound-soft reflector. The reflectivity and temporal stability of this lining has been essentially improved by inserting a thin polyethylene foil between foam and water (D. Guicking and Th. Wetterling, to be published in Acustica (1974) Vol. 30). Fig. 1 shows the reverberation time of the tank as a function of frequency for the present and the former construction.

A necessary assumption for all reverberation room measurements is a diffuse sound field. The diffusivity of the tank is measured by the method of Furduiev (2) with a directional hydrophone. The influence of additional diffusers (e.g. aluminium plates) is shown for some special cases.

The evaluation of absorption measurements is often complicated by diffraction from the edges of the absorber sample by which the apparent area of absorption is changed. The influence of this “edge effect” is more pronounced for absorber samples on sound-soft than on rigid walls (3). This problem is therefore treated theoretically and experimentally for rectangular samples.

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EIGHTH INTERNATIONAL CONGRESS
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SEA SURFACE SCATTERING MEASUREMENTS USING A PARAMETRIC
SOURCE AND DIRECTIONAL SEA SENSING SYSTEM

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INTRODUCTION

Acoustic signals scattered from the sea surface are modulated in both
amplitude and phase. The doppler spectrum shift of the reverberation
is dependent on the directionality of the sea surface waves. Doppler
spectrum measurements are not practical from mobile platforms due to
source/receiver motion; nor are doppler measurements practical in
shallow water due to multipath interference. These limitations have
been overcome in a shallow water experiment by using a parametric
source for projecting a narrow beam of acoustic energy at the sea
surface with no minor lobe interference. The sea surface directional
wave spectrum is measured so that an adequate prediction of the
reverberation characteristics can be obtained.

EXPERIMENTAL PROCEDURE

A vertical array of 25 hydrophones is supported from a barge which is
implanted in a water depth of 100
feet. A parametric source that is
mounted to the barge is tilttable in
angle of elevation ±90 degrees
about normal incidence to the sea
surface. (See Figure 1.) Beam
patterns and source levels have
been obtained for the parametric source over the frequency range 3-10
kHz. The bistatic reverberation level from the sea surface is received
on the vertical array and recorded on magnetic tape for later digital
processing. Data are obtained as a function of angle of incidence,
angle of scatter, and sea state. To supplement the surface reverbera-
tion measurements, an estimate of the directional wave spectrum of the
sea surface waves is obtained by measuring the wave height, as a
function of time, at five points on the sea surface. (The acoustic sea
sensing technique was described in a paper presented at the 84th
Meeting of the Acoustical Society of America.) Meteorological data are
recorded throughout the experiment.

EXPERIMENTAL RESULTS

The bistatic reverberation level, doppler spectrum, temporal and
spatial coherence are discussed in terms of the acoustic frequency,
geometry and sea state.

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ON VERTICAL CORRELATION OF SOUND SIGNALS SCATTERED
BY THE OCEAN BOTTOM

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Sechkin VA Volovov V I

INTRODUCTION
Vertical and horizontal spatial correlation functions of scattered signal amplitude fluctuations have been simultaneously measured over a frequency range from 3 to 30 kcps at normal incidence of sound on the ocean bottom.

EXPERIMENTAL
Measurements of spatial correlation were carried out from a drifting vessel by means of a vertical array of omnidirectional hydrophones. Fig.1 gives a typical dependence of vertical (I) and horizontal (II) correlation function on hydrophone separation $s$ at sound frequency 30 kcps.

![Spatial correlation functions](image)

It can be seen that the radius of horizontal correlation $s_h$ is much less than the radius of vertical correlation $s_v$. In accordance with the approximate theory their ratio is

$$\frac{s_h}{s_v} = \sqrt{2} t_0 \delta,$$

where $t_0 \delta$ is mean-root-square slope of roughnesses on the ocean bottom. Using eqn(1) it is possible to determine magnitude of $\delta$. For this case we obtain $\delta \approx 2.6^\circ$.

The experimental data on vertical correlation obtained simultaneously at two different frequencies and at particular hydrophone separation permit to determine mean-root-square effective height and correlation radius of roughnesses on the ocean bottom.

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SUR L'IMPÉDANCE MOTIONNELLE D'UN HAUT-PARLEUR CHARGÉ
PAR UN TUYAU SONORE

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INTRODUCTION

Le système étudié est un haut-parleur chargé par un tuyau sonore fermé de section égale à la surface du diaphragme. Les importantes variations de l'impédance acoustique du tuyau sonore au niveau du diaphragme du haut-parleur et la connaissance d'une expression mathématique correcte pour cette impédance permettent le calcul des paramètres mécaniques et acoustiques du dispositif.

ÉTUDE THÉORIQUE

Des expressions de l'impédance d'entrée d'un tuyau sonore ont déjà été proposées. Il en est de simples (1) que nous n'avons pas pu retenir car elles ne tiennent pas suffisamment compte de la dispersion. D'autres plus élaborées (2), obtenues en traitant le tuyau comme une ligne électrique, présentent l'inconvénient d'introduire quatre paramètres. Nous avons montré que deux paramètres, déduits de la théorie de Kirchhoff (3), suffisent pour décrire les propriétés de notre tuyau dans un large domaine de fréquence. En outre le facteur de couplage bobine-tuyau défini par Fay et White (4) est ici exprimé en fonction de trois paramètres complexes.

ÉTUDE EXPÉRIMENTALE

L'étude expérimentale est basée sur la mesure de l'impédance motionnelle du haut-parleur par la méthode du pont déséquilibré (5). L'allure de l'admittance motionnelle (figure ci-contre) fait clairement apparaître l'influence de l'impédance acoustique d'entrée du tuyau sonore. Le tracé de la partie réelle de cette admittance en fonction de sa partie imaginaire pour une fréquence donnée (la longueur du tuyau étant variable) donne des points qui peuvent être approximativement distribués sur des cercles, ce qui permet d'obtenir les valeurs des paramètres par itération.

REFERENCES

EIGHTH INTERNATIONAL CONGRESS ON ACOUSTICS, LONDON 1974

THE EFFECT OF JET TEMPERATURE GRADIENTS ON THE DIRECTIVITY OF SOUND RADIATION

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INTRODUCTION

Temperature gradient effects on the directivity of spherical modes in a subsonic jet are investigated. Results are obtained which show good agreement with a previous theoretical prediction(1) and experimental results(2).

ANALYSIS

The convected wave equation governing the propagation of sound in an axisymmetric hot jet is derived. By assuming a solution similar in form to the classical solution, the equation for the directivity of the radiated sound may be obtained as

$$\frac{d^2 F}{d\theta^2} + A(r, \theta) \frac{dF}{d\theta} + B(r, \theta) F = 0 \quad (1)$$

The solution to Eq. (1) is obtained by a seminumerical technique using a Runge-Kutta method at each radial position.

DISCUSSION

The computed directivity solutions show that the refraction "valleys" which exist on the axis of an ambient jet are enhanced in the hot jet. In a cold jet the refraction due to the temperature gradient is in opposition to the flow effects. The present solution for the directivity of sound radiated from a point source in a hot subsonic jet is compared with that of Schubert(1) and with the experimental data of Grande(2) in Fig. 1. Refraction effects within the jet (10° half-angle spread) are predicted to within 5dB of the experimental results while outside the jet the present solution approaches the experimental results asymptotically.

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SOUND SOURCES OF HIGH DIRECTIVITY

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INTRODUCTION

The realisation of circular piston with Gauss' distribution of sound velocity amplitude (1,2) prompted us to the theoretical researches after the possibility of sound radiation only in the definite cone.

THEORETICAL ANALYSIS

The directivity pattern is easily expressed in terms of a Hankel's transform if the amplitude distribution diminishes so rapidly with the radius \( r \) that the upper limit of integration \( a \) (the radius of a transducer) may be replaced by infinity (2,3):

\[
R = \frac{2 \pi u_0(r) r^2}{Q} J_0(ksin\gamma)(ksin\gamma)^2 dr
\]

where \( Q \) is the total source output calculated in an usual manner. In that case:

\[
R = \frac{2 \pi}{Q(ksin\gamma)^2} F_0(ksin\gamma)
\]

where \( F_0(ksin\gamma) \) is the Hankel's transform of zero order.

Looking for the inverse transform of \( F_0(ksin\gamma) \), being constant in a certain angle \( \gamma_{um} \) and zero for \( \gamma > \gamma_{um} \) we can find \( u_0(r) \) for a piston which radiates sound only if \( \gamma < \gamma_{um} \).

The most important example is the distribution:

\[
u_0(r) = J_1(ar) \]

where \( J_1(ar) \) is the first order Bessel function.

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    miai Kiado (1971) 537; (2,3) R.Wyrzykowski, Cz.Sołtys, Archiw.
    Akustyki (1972) 2 327; (1973) 2 31.
UNTERSUCHUNG SYNTHETISCHER SCHALLFELDER MITTELS COMPUTER
BEI VERWENDUNG EINER GRADIENTENEMPÄNGER IN 1-TEN ORDNUNG

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PROBLEMSTELLUNG

Die meisten Arbeiten die sich mit dem Problem des synthetischen /z.B. stereo-, quadro-, u.s.w./ Schallfelder /F/ befassen, suchen die subjektive Verhältnisse, und die physikalischen Beziehungen werden selten behandelt. Die Berechnungen werden durch die Abtastung des F mittels einer Gradientenempfänger in 1-ten 0. in der Empfängerabstand 0,21 m auf die Druckdifferenzen durchgeführt.

THEORIE

In $\mathcal{E}(1,n)$ Quellen werden im freien $F$ errichtet. Der Schalldruck einer punktförmigen Quelle ist auf der Abb. 1.

$F=\{k_j, r_j, \varphi_j\}$. In der Praxis sind noch Quellen-D($\theta_j, \varphi_j, \psi_j$), und Empfängerrichtfaktoren $K_{nj}(k_j, \alpha_j, r_j, \varphi_j, \beta_j)$ vorhanden. Die Werte $A_n, D_n, K_n$ können in je eine Spaltenmatrix mit $n$ Elementen geordnet werden. Der Schalldruck ist an jedem Empfänger $n$

$\rho_n = A_n D_n K_n$

(1)
die Druckdifferenz

$\Delta p = D(K, A_1 - K_1 A_2)$

(2)

Die Berechnungen beschränken sich auf $k=1$ und in Spezialfällen auf $D=1$. Die Modifikationen der $F$-parameter können im Form des totalen Differentials von $\Delta p$ geprüft werden $c(\Delta p) = \frac{\partial \Delta p}{\partial k} = \frac{\partial \Delta p}{\partial l}$

(3)

DIE RECHNERERGEBNISSE

Abb. 2. Die Computerergebnisse zeigen eine sich stark ändernde Feinstruktur der synthetischen $F$, auch bei grossem Quellenabstand. Die Modifikation der $F$-parameter erzeugt bei einer $1\%$ Änderung der Variablen Änderungen, die grösser sind, als die Grenze der subjektiv wahrnehmbaren Pegeländerungen. Als Beispiel werden hier eine Stereosanordnung und die Rechnerergebnisse von $\Delta p$ auf Abb. 1 und 2 dargestellt.

LITERATURSTELLEN

EIGHTH INTERNATIONAL CONGRESS
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A THEORETICAL FORMULATION OF THE INFLUENCE OF A BOUNDARY
LAYER GROWTH IN AN INLET FLOW DUCT WITH TEMPERATURE
GRADIENTS ON SOUND PROPAGATION FROM AN ARBITRARY SOUND
DISTRIBUTION

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INTRODUCTION

The accurate evaluation of the noise radiated from an inlet of an air-
craft engine necessarily involves a study of sound propagation in such
inlet ducts. In the inlet, apart from the fact
that noise must propagate upstream, the flow is
such that the inherent boundary layer growth
in the initial part of the inlet results in an
accelerated axial flow gradient. The velocity
is uniform at the entrance and grows with dis-
tance from the entrance until flow becomes
fully developed, if the duct is long enough.

Fig 1 Sound propagation in
in an Inlet Duct

THEORETICAL ANALYSIS

The problem of influence of a boundary layer growth in an inlet duct
with flow and temperature gradients on sound propagation from an arbi-
trary source distribution has been theoretically formulated. Two me-
thods are suggested to solve a set of partial differential equations
subject to certain boundary conditions. The first method is the
travelling wave method. It concerns matching iterative solutions of
the travelling wave type, using successive approximation of Runge-
Kutta. The second method is the finite element method of the type
considered by Skiba (1). Each of the linearised Navier-Stokes, con-
tinuity and entropy equations is replaced by an assembled Galkerkin-
finite-element equation. By the application of boundary conditions,
finally one gets from the above set of equations, an equation linking
unknown p values to those of the specified values at the source plane.
Here p denotes acoustic pressure.

REFERENCES

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EIGHTH INTERNATIONAL CONGRESS ON ACOUSTICS, LONDON 1974
THE TRANSMISSION OF SOUND IN A NONUNIFORM DUCT WITH NO MEAN FLOW

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INTRODUCTION
The transmission of sound in a nonuniform two dimensional duct without flow is investigated by a method of weighted residuals (MWR) which leads to a set of coupled "generalized telegraphist's equations". Results for several duct configurations are compared with a variational method (1), a stepped duct approximation (2), and an eigenfunction expansion method based on linearly tapered duct segments.

METHOD
The MWR (3),(4) is employed because no restrictive assumptions on duct geometry are required and because it appears to be extendable to ducts with flow. Solutions for the acoustic pressure, axial particle velocity, and transverse particle velocity are assumed in the form

\[ p = \xi p_n(x) \cos k_n y \]
\[ u = \xi u_n(x) \cos k_n y \]
\[ v = \xi v_n(x) \sin k_n y \]

where \( k_n \tan k_n b = ik \tilde{A} \), \( k_b \) is the reduced frequency based on the local duct height, and \( \tilde{A} \) is the local lining specific admittance. A set of coupled equations of the form of eqn. (1) results.

\[
\begin{pmatrix}
\frac{d}{dx} u_n \\
\frac{d}{dx} p_n
\end{pmatrix} = [C]
\begin{pmatrix}
u_n \\
p_n
\end{pmatrix}
\]

These are integrated from the beginning to the end of the nonuniformity to form a transfer matrix relating \( u_n \) and \( p_n \) at these points. By relating the \( u_n \) and \( p_n \) at each end to the amplitudes of right and left running waves on the source side (\( a^+_n, a^-_n \)) and right running waves on the transmission side (\( b^+_n \)), reflection and transmission coefficient matrices for the nonuniformity are computed as in eqns.(2)

\[
\begin{pmatrix} a^-_n \end{pmatrix} = [\text{REF}]{a^+_n} \quad \begin{pmatrix} b^+_n \end{pmatrix} = [\text{TRAN}]{a^+_n}
\]

The method appears to be valid for any smoothly varying nonuniformity in duct height or lining admittance.

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EIGHTH INTERNATIONAL CONGRESS ON ACOUSTICS, LONDON 1974

ON THE THEORY OF SOUND RADIATION BY PLATES AND SHELLS

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INTRODUCTION

The problem of sound radiation by a thin elastic vibrating plate due to external pressure and shear stresses fluctuations is considered. The solution is used for analyses of influence of shear stresses fluctuations at sound field.

STATEMENT OF THE PROBLEM

It is required to solve the equation

$$\Delta = \sum \frac{\partial^2}{\partial x_2^2}$$

under boundary conditions

$$\rho \frac{\partial \Delta W}{\partial x_2} = \frac{\partial^2 \rho}{\partial t^2} |_{x_2=0}$$

$$m \frac{\partial^2 W}{\partial t^2} + q \frac{\partial^2 W}{\partial x_2^2} = p^o - \rho + h \frac{\partial L_i}{\partial x_1} + h \frac{\partial L_s}{\partial x_2} |_{x_2=0}$$

and $p$ is regular at infinity, when $p^o$, and $L_i$ - pressure and stresses fluctuations, $m$, $h$ - mass and thickness of the plate.

RESULTS

The solution is written immediately in integral form by using reciprocity theorem [1]. Spectral and stresses fluctuations [2] are used for calculations. The more common problem-cylindrical shell sound radiation is considered.

REFERENCES

ÉQUATION DE HELMHOLTZ À COEFFICIENT VARIABLE ET PROPAGATION DANS LES PAVILIONS

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La propagation du son dans les tuyaux de section variable obéit à l'équation de Webster :

\[ a_{xx} p(x) + \frac{1}{S(x)} a_{x} S(x) a_{x} p(x) + k_o^2 p(x) = F(x). \]

On ne connaît pas la solution générale de cette équation, mais un changement de fonction (1) permet de l'écrire :

\[ a_{xx} \psi(x) + \left[ k_o^2 - A(x) \right] \psi(x) = F^0(x). \]

Il y a donc analogie entre la propagation du son dans un tuyau de section variable et dans un milieu inhomogène à une dimension.

L'équation (2) présente sur l'équation (1) l'intérêt d'être du type de Helmholtz, à coefficients constants :

\[ a_{xx} \psi(x) + k_o^2 \psi(x) = F^0(x) + A(x) \psi(x) \]

si l'on considère \( A(x) \psi(x) \) comme un terme source. Par conséquent l'expression formelle de \( \psi(x) \) s'en déduit immédiatement :

\[ \psi(x) = \int_{x}^{x'} F^0(x') G(x,x') \, dx' + \int_{x}^{x'} A(x') G(x,x') \psi(x') \, dx' \]

avec \( G(x,x') = \frac{e^{-jk|x-x'|}}{2jk} \).

Cette formulation intégrale permet donc d'aboutir, au moins numériquement, à la solution de l'équation de Webster quelle que soit la fonction \( S(x) \). Elle devient même particulièrement avantageuse dans le cas d'un connecteur, c'est-à-dire d'un pavillon de longueur finie raccordant deux tuyaux cylindriques de sections différentes.

A titre d'exemple nous traitons le cas du connecteur conique en calculant notamment son impédance d'entrée. Nous comparons nos résultats à ceux d'une méthode plus classique (2) ; ils concordent avec une très bonne précision.

REFERENCES

INTRODUCTION
As an example we determine the near-field of a circular piston set in an infinite baffle. The special results of Barkhausen and Stenzel are well known. The sound field in an arbitrary point of the space was calculated by Stenzel with help of a dual series. In this paper the author will give the sound field in an arbitrary point by an infinite series whose zero order term gives the near-field in a closed mathematical form.

THEORETICAL ANALYSIS
As a ground for our calculating process we choose the well-known integral representation of the sound field on the baffle:

\[ P_{200} = jk \rho c \nu_0 R \int J_0(kr) J_1(kr) \left( \frac{kr}{2} \right)^{\frac{3}{2}} \, dr \]

where: \( R \) is the piston's diameter; \( \nu_0 \) is the piston's velocity; \( k \) is the wavenumber; \( \rho c \) is the specific acoustical resistance; \( \frac{kr}{2} \) is the radial coordinate; \( J_0 \) is the first kind Bessel function. After using of the summation theorem and of some integral formulas we get the result in form of a Neumann series:

\[ P_{z=0} = jk \rho c \nu_0 R \sum_{n=0}^{\infty} \frac{(kr)^{2n+1}}{2^{2n+1} n!} \, \left( \frac{kr}{2} \right)^{\frac{3}{2}} \]  

This series converges more rapidly than the Stenzel's series and gives the sound field on the piston, as well as outside the piston area on the baffle. \( H \) is the Hankel function. The sound field in an arbitrary point of the space is determined by solving a special Dirichlet problem. The Green function will be:

\[ G = \frac{1}{4\pi} e^{-r/k} \cos(m\phi) \sum_{n=0}^{\infty} \frac{(kr)^{2n+1}}{2^{2n+1} n!} \, \left( \frac{kr}{2} \right)^{\frac{3}{2}} \]  

where: \( r \) is the radial coordinate; \( z \) is the axial coordinate; \( \phi \) is the axial coordinate of the point source generating the Green function. Consequently the Dirichlet problem results in following formula:

\[ p = \int_{0}^{2\pi} \left[ \frac{\partial}{\partial z} G \right] \cdot g \, dp \, d\phi \]

After integrating, the zero order term which gives the near-field in an arbitrary point will be:

\[ p(0) = \rho c \nu_0 \left( 1 - \exp(-kr) \right) \left( \frac{k}{2} \right)^{\frac{3}{2}} \left( \frac{k}{2} \right)^{\frac{3}{2}} \left( \frac{k}{2} \right)^{\frac{3}{2}} \left( \frac{k}{2} \right)^{\frac{3}{2}} \]

where: \( H \) is the zero order Struve function. The above calculating method is able also in the case of a vibrating circle, or in the case of the finite length cylinder radiator.

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MOTION-INDUCED CHANGES IN A CLASS OF RADIATION PATTERNS

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INTRODUCTION

Exact analytic expressions, valid for arbitrary Mach numbers, are presented for the motion-induced changes in the radiation patterns of a class of point acoustic sources. The radiation patterns of the sources are surfaces of revolution where the axis of revolution is parallel to the forward velocity vector of the source. The total energy radiated is assumed to be independent of the source velocity.

The directivity induced on an otherwise omnidirectional stationary point source by ocean currents has been considered in previous papers (1,2); in the ocean, this directivity is bounded and shown to be negligible. For aircraft operations, however, such motion-induced changes are not negligible.

RESULTS

The directivity changes are the result of two causes. First, energy that a stationary observer "sees" leave a stationary source in direction A will appear to leave in direction B when the source is moving (3,4). The angles A (stationary) and B (moving) are measured from the velocity direction to the line between the observer and source at the instant the energy reaches the observer. The angles A and B are related by

\[
\cos A = \cos B \left( 1 - M^2 \sin^2 B \right)^{1/2} - M \sin^2 B,
\]

where the Mach number \( M \) is the ratio of the velocity magnitude to the constant sound speed. This angular sweepback (3) is symmetrical around the velocity direction.

The second cause is less direct. A radiation pattern can be considered as a measurement of the acoustic energy that passes through small standard areas in a spherical surface centered on a stationary source. Motion-induced changes in the areal density and, consequently, in the radiation pattern, are found by examining motion-induced changes in the above supporting standard areas. The result is

\[
f(B) = \left( 1 - M^2 \sin^2 B \right)^{1/2} \left[ 1 + M \cos B \left( 1 - M^2 \sin^2 B \right)^{-1/2} \right]^2.
\]

For a moving source, the changes due to this second cause range from -3 dB ahead to +2 dB behind when \( M = 0.3 \) and from -20 dB to +5.5 dB when \( M = 0.9 \).

The total change is given by the sweepback from A to B and the change in the areal density in the direction B.

REFERENCES

A METHOD FOR MEASURING THE ACOUSTIC RESISTANCE OF POROUS MATERIALS

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A device has been designed to determine the flow resistance of closed cells porous materials, a datum that cannot be measured with the normal method of the continuous flow. The measurement has to be carried at very low frequencies, in order to approximate the conditions of a continuous flow.

Fig. 1 reports the schema of the device and fig. 2 its equivalent circuit when distances a and b are small with respect to the wavelength (in our device a = 2b): l is the thickness of the material, represented in the schema as a symmetrical four terminal network.

Two measurements are made, releasing the amplitude ratio and the phase difference between the microphone outputs of \( M_1 \) and \( M_2 \) when the sound sources \( S_1 \) and \( S_2 \) (both having a very high mechanical impedance) are successively supplied. These data allow to calculate easily the values of the characteristic impedance and the propagation constant of the material under test. In fact the flow resistance \( r_s \) (specific per unit of length) is given by:

\[
r_s = \frac{\frac{N_1}{N_2}}{\frac{a}{b}} \cdot \frac{\omega l}{c}
\]

where \( N_1 e^{i\psi} \) and \( N_2 e^{i\psi'} \) are respectively the amplitude ratio and the phase difference.

The device is also used for measurements on materials under controlled atmosphere: some results will be furtherly produced.

Fig. 1 - Measuring apparatus Fig. 2 - Equivalent circuit of the measuring device
THE FREQUENCY SPECTRUM OF UNLOADED PLANAR RECTANGULAR PIEZOCERAMIC RESONATORS

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The identification of the vibrating modes is necessary for measurement of constants of materials used for resonators and transducers. The frequency spectrum of thin, mechanically free, fully electroded, rectangular piezoceramic plates made from PbSr(Zr,Ti)O₃, was examined.

Three fundamental groups of the vibrating modes were identified. (See Fig.1. Frequency spectrum vs length l to width b ratio.)

The A group - dilation modes - was studied theoretically by Holland (1). Our experimental results shown on Fig. 2 and 3-frequency spectrum and piezoelectricity vs 1/b ratio respectively, were in good agreement with theoretical ones presented in (1). The resistance in resonance and its correlation with piezoelectricity was studied too.

The B group - vibrating modes in thickness resonance region. The frequency spectra of resonators square, oblong, trapezium and circular shapes were compared. It was found certain regularity, which made possible the identification of thickness resonance necessary to k₄ and c₃₃ determination.

The third group - vibrating modes that depend on the width of plate only (mode C) or on both width and thickness (mode D, E, F).

Used measuring method and its accuracy is discussed. The comparison with early results (2) is shown as well.

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SOUND GENERATION BY IMPACTING SOLID SPHERES

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INTRODUCTION

A simple theory of the sound produced by the collinear collision of two solid steel spheres is presented. This theory is in reasonable agreement with experimental sound pressure time histories.

THEORY

The leading face of a moving sphere is represented by a fluid source and its trailing face by a fluid sink. For simplicity these are considered to be a point source and sink located at the centres of the leading and trailing faces. Providing the sphere moves with a constant velocity, the sources and the acoustic velocity potential which they generate remain invariant, therefore giving rise to no acoustic overpressure. Overpressure is however generated when the sphere undergoes acceleration in the collision process.

In the collision process each sphere behaves as an acoustic dipole giving rise to a highly directional sound field. The actual over-pressure experienced is due to the time delay between the signal from the front of the sphere and a signal of opposite polarity from its rear face.

As the over-pressure was the difference of two time shifted time histories, these time histories had to be calculated with the maximum degree of precision. The Hertz contact law, for the force between the spheres, is used to numerically calculate the acceleration time history.

EXPERIMENTS

Two ball bearings were given collinear bifilar pendulum suspensions in the manner of Newton's cradle. One pendulum was released from a known height and the sound pressure history arising from the first collision with the sphere on the second pendulum was recorded. The monitoring system involved a condenser microphone and digital event recorder.

Results such as those shown in Figure 1 are presented for various microphone positions for collisions of equal and unequal sized spheres.
EIGHTH INTERNATIONAL CONGRESS ON ACOUSTICS, LONDON 1974

RECIPROCITY MEASUREMENTS OF SOUND TRANSFER IN MECHANOCOUSTICAL SYSTEMS FOR MULTIDIRECTIONAL EXCITATION

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Recent publications (1,2) have pointed out that, when multidirectional excitation is to be considered, the measurement of sound transmission through resilient mountings, cables etc. becomes practicable by means of reciprocity experiments.

For instance, suppose that on board a ship the six transfer functions describing the sound transfer from an engine foot (position 1) to a position in the accommodation or in the water (position 2) should be measured. It would give serious practical problems to excite the three translational and the three rotational vibration components independently.

These difficulties are circumvented by performing the equivalent reciprocity experiments.

\[
\begin{align*}
\begin{array}{ccc}
E & \rightarrow & v_{x1}' \ (m/s) \\
\downarrow & & \\
p_2' \ (Pa) & \rightarrow & \begin{array}{c}
F_{x1}'' \ (N) \\
U_2'' \ (m^3/s) \end{array}
\end{array}
\end{align*}
\]

direct experiment

reciprocity experiment

The equivalent direct and reciprocity experiments follow from the reciprocity relation given in eqn. (1). The symbols are illustrated in the figure.

\[
\frac{p_2'}{v_{x1}'} = \frac{F_{x1}'' \ blocked}{U_2''}
\]

From this equation it can be deduced that the determination of the ratio of the volume velocity \(U_2''\) of an acoustical point source and the blocked force on the top of the mounting is equivalent to the determination of the ratio of a velocity component of the engine foot and the sound pressure generated by it.

In the reciprocity experiment the mounting is loaded ("blocked") by a rigid block, the centre of gravity of which being accessible to and in solid contact with the top of the mounting. By measuring the translational velocities of the gravity point and the rotational velocities of the block, the forces and couples are calculated with Newton's law. The measuring technique will be explained and illustrated by some results.


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DUAL FORMULATIONS FOR ACOUSTO-STRUCTURAL VIBRATIONS

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INTRODUCTION

Acousto-structural vibration problems arise in many engineering systems - notably as panel and duct oscillations in aerospace vehicles. For design purposes it is essential to develop procedures for calculating the dynamic response of the gas-structure system. In recent years considerable progress has been made in the development of finite element procedures for analysis of large scale structural and acoustic vibration problems, e.g. (1, 3). The elements developed have invariably employed displacements for the structure and the pressure for the gas. Use of these elements for the combined gas-structure problem encounters difficulties at the interface where neither equilibrium nor the compatibility requirements can be strictly satisfied.

METHOD OF ANALYSIS

In the present paper it is shown that for a consistent formulation one may start from Hamilton's principle and develop finite element models for both the gas and the structure in terms of kinematic variables. Then the compatibility requirements can be satisfied exactly and \( \text{a priori} \), for the entire system. The equilibrium equations will tend to be satisfied through the process of extremisation, i.e. \( \text{a posteriori} \). Conversely one may employ the complementary variational principle and develop the gas-structure system equations in terms of force variables. In that case the equilibrium equations can be satisfied exactly and \( \text{a priori} \) while the compatibility equations tend to be satisfied \( \text{a posteriori} \). The two procedures are discussed in parallel and the advantages of each are pointed out. The paper includes some illustrative examples the results of which are compared with experimental results and theoretical results of other investigators.

REFERENCES

EIGHTH INTERNATIONAL CONGRESS ON ACOUSTICS, LONDON 1974

SOUND FIELDS NEAR PLANE, ELASTIC AND INHOMOGENEOUS WALLS

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PROBLEM

We investigate boundary conditions for a sound field in a fluid bounded by an elastic, plane wall of uniform thickness h, but with non-uniform elastic properties. In a previous paper (1) the problem was solved analytically for uniform properties.

PROCEDURE

Let \( U(z) \) be the set of those quantities (displacements, forces, fluxes), which are continuous at the interface. The equations of motion, in the form \( \frac{dU}{dz} = HU \), continue them along \( z \) normally into the wall. The matrix operator \( H \) differentiates the quantities \( U \) with respect to time \( t \) and coordinates \( x \) parallel to the wall. At \( z = h \) the solution \( U(h) \) meets given boundary conditions, which imply those for \( U(0) \).

UNIFORM WALL

The formal solution is \( U = e^{H\xi} U(0) \). Fourier components of \( U(0) \) with phase factors \( \exp(iK \cdot x - i\omega t) \) are subject to independent boundary conditions, stated in (1) in terms of the tensor of susceptibility \( S(k,\omega) \), which relates displacements to forces.

NON-UNIFORM WALL

\( \frac{dU}{dz} = (H + V)U \), where \( V \) is a non-uniform perturbation of the uniform operator \( H \). \( V \) is supposed to be a periodic function of \( x \) and a function of \( z \). The solution \( U(z) \) can be approximated by an expansion with respect to the perturbation operator \( V \):

\[
U(z) = e^{H\xi} (1 + \int_0^z e^{-H\xi} V e^{H\xi} + ...) U(0)
\]

The required conditions for \( U(0) \) can be obtained by first applying the operator in brackets, followed by the boundary conditions for the uniform wall. Of course, the Fourier components of \( U(0) \) are no longer independent, because the non-uniform operator \( V \) combines different wave numbers \( k \).

REFERENCE

(1) H. U. Vogel, Fluid Dynamics Transactions 6, 11; Warsaw 1972
INTRODUCTION

This study investigates the large amplitude two dimensional acoustic waves resulting from the resonant excitation of a periodically supported infinitely long viscoelastic plate. The analysis fully treats the nonlinear terms arising from the equations of motion for the fluid and the plate, and from the boundary condition describing their interface. The resulting solution for the nonlinear acoustic waves shows several strong differences from the predictions of linear theory.

ANALYSIS

The solution uses Lighthill's method (1) to express the independent variables, as well as the dependent ones, in asymptotic series. Uniformly valid expressions for the fluid potential, the fluid particle velocity, and the plate displacement are obtained.

In the case of radiating waves the velocity potential \( \phi \) is found to be

\[
\phi = -\frac{\epsilon}{k} \cos(\tau - k\eta) \sin(j\pi\xi) + O(\epsilon^2)
\]  

(1)

while for decaying waves

\[
\phi = -\frac{\epsilon}{k} \exp(-k\eta) \sin\tau \sin(j\pi\xi) + O(\epsilon^2)
\]  

(2)

where \( \epsilon \) is the non-dimensional amplitude of the plate displacement and \( \tau \) is the non-dimensional time. The variables \( \xi \) and \( \eta \) are related to the spatial coordinates \( x \) (distance parallel to the plate) and \( z \) (distance perpendicular to the plate) by

\[
x = \xi + \epsilon X(1)(\xi,\eta,\tau) + O(\epsilon^2)
\]  

(3)

\[
z = \eta + \epsilon Z(1)(\xi,\eta,\tau) + O(\epsilon^2)
\]

The functions \( X(1) \) and \( Z(1) \) depend upon the type of wave. The results of linear theory (2) are obtained by letting \( X(1) = Z(1) = 0 \).

It is found that radiating waves develop a shock at a critical distance from the surface of the plate, as illustrated for a typical case in Fig. 1.

![Fig.1](image)

**Fig.1** x-component of fluid velocity, \( x=1/6, \tau=0, \epsilon=0.0025 \)

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NUMERICAL PROCEDURE FOR CALCULATING FINITE-AMPLITUDE DISTORTION IN A CLOSED TUBE

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INTRODUCTION

A numerical procedure is developed to calculate the one-dimensional, finite-amplitude, standing wave produced between a source vibrating with a given particle-velocity time profile and a boundary of known impedance.

DESCRIPTION OF THE MODEL

The procedure begins with the launching of a progressive wave with the particle-velocity time profile of the source. The propagation of this wave under the combined effects of nonlinearity and absorption (with possible dispersion) is calculated stepwise using a computer model similar to the one described in (1) but modified to include the Fast Fourier Transform and complex absorption coefficients. Reflection of the wave from the boundary is treated linearly by assuming that each frequency component reflects independently (2). Propagation of the reflected wave back to the source is calculated under the assumption that oppositely-directed waves do not interact (3). At the source, the returning wave is rigidly reflected and added with the proper phase to the new wave being launched from the source. The procedure is repeated until convergence is obtained, i.e., until the time waveform at each point during a given round trip does not differ appreciably from the corresponding time waveform during the previous round trip. The final time waveform at each point is the sum of the two oppositely-directed components. At present, the computer program is limited to preshock conditions. However, the use of weak shock theory to describe the propagation of the wave when shocks exist (4) may extend the program's range.

TEST RESULTS AND CONCLUSION

A comparison test of this procedure was made with a completely different procedure developed by Coppens and Sanders (5) which accurately predicts the preshock behavior near resonance in a gas-filled, rigid-walled, closed tube. Results were in near-perfect agreement. Thus it is expected that the present model can be applied successfully to predict preshock behavior in other one-dimensional, standing wave systems such as an open-ended tube.

REFERENCES

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INFRASONIC NOISE PROBLEMS

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INTRODUCTION

The investigations of effect of noise on man in different industries have shown that there occurred a greater deal of hearing damage, serious illnesses as well as psychological disturbances in some factories more than in the other ones, although there has existed almost the same sound level from 90-96 dB and a similar frequency spectrum in the audible range. From the obtained measurement results in sub-audible range in those factories where sufferings have been worse it can be seen that there has been intense infrasonic noise there.

EXPERIMENTAL

An audiological analysis of fatigue has been carried out by means of tone decay at the threshold of hearing (Carhart's test). The ear has been loaded by various kinds of noise, i.e. by the industrial noise recorded on a tape recorder, by a 1/3-octave band of industrial noise (f₀=2000 Hz) and by the industrial noise to which is added an infrasound component (8 Hz, 120 dB re 20 μPa) during ten minutes. The sound level of the noise in the audible range for all three cases was 96 dB. The investigation results have shown that the ear loaded by two noise components - audible and infrasonic - simultaneously shows a rise in auditory threshold even for 45 dB.

The measurements as well as the analysis of noise of different sources - factories, shipyards, ships, road vehicles, locomotives, aeroplanes (1) - have shown that there exists an infrasound level from 90-140 dB in the range from 0.5-20 Hz with prominent components in the middle of the range.

CONCLUSION

The phenomenon of the intensive infrasound components in noise as well as their harmful effect on man (2) imposes the need of noise measurements not only in the audible range but also in the infrasonic range too. A sound level meter is recommended for both audio and infrasonic range. The basic characteristics of such an instrument in the audible range are weighting curves A, B, C, and D. The frequency range of the infrasound range has two principal characteristics: the first one from 0.5 Hz - 20 kHz and the second one from 0.5-20 Hz. Due to measurements taken at these ranges there is a relation achieved between the audible and infrasonic components in the investigated noise. On behalf of varying the upper limiting frequency (e.g. down to 10 Hz) and the lower one (e.g. to 1.2.5 Hz) in the infrasound range it is possible to obtain nearer data on infrasound content in noise.

REFERENCES

SOUND ABSORPTION IN AIR

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Investigations concerning the propagation of sound and the prediction of environmental noise levels require a detailed knowledge of the attenuation coefficient of sound in air and its dependence on frequency, humidity and temperature. Whilst the existing data may be adequate for some purposes, there remain uncertainties, arising out of discrepancies in the published data, which become important in such applications as scale-model studies which are carried out at very high frequencies and the determinations of aircraft noise levels which involve large propagation path-distances.

In an earlier paper (1) absorption data were presented for the frequency range 1 - 12.5 kHz. These data are now given in a normalised form, allowing a more direct comparison with later work, notably that of Harris (2). The relaxation frequency derived from the measurements, which is used as one of the normalising parameters and varies with the water vapour content of the air, is also shown in comparison with data from other sources. In particular, there is seen to be good agreement with the recent work of Evans et al (3).

The normalisation is based on an expression for the $O_2 - H_2O$ relaxation process of the form

$$m/M = 2\left(\frac{f}{F} + \frac{f}{F_r}\right)^{-1}$$

where $m$ is the molecular attenuation coefficient and $M$ its maximum value, $f$ is the frequency and $F$ is the relaxation frequency. The influence of relaxation effect in nitrogen is discussed.

These audio frequency data are used to predict attenuation values at frequencies up to 100 kHz.

It is difficult to resolve the discrepancies in the published data by means of direct measurements over large distances outdoors because these are subject to uncertain meteorological conditions along the sound path and to other practical difficulties such as the effects of ground absorption. However, a limited number of laboratory measurements over distances of a few metres have been made in the frequency range 25-100 kHz at ambient humidities and temperatures, and are shown in comparison with the predicted values.

REFERENCES

THE ROLE OF RESONANCE IN INFRASOUND EFFECTS

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Omitting overtones is an essential point of the hearing tests and of the investigations of physiological effects of infrasound. For this purpose experiments were tried in Helmholtz resonators. Good resonators could be made by use of convenient size of the neck even in big chambers, and infrasonic resonance can come into being both in practical life and in nature /1/. The big volume \( V \) makes it possible to place experimental animals or even men inside the resonator. To avoid the outside noise, experiments were made of closed "double Helmholtz resonators", too. The advantages of the use of resonators for experiments are as follows:

1st: by the resonance the level of the overtones decreases rapidly relative to the fundamental tone;
2nd: less energy is needed to excite a desired sound level.

In the course of the investigations it was aimed to achieve resonances as sharp as possible by small balloons \( \text{max. 20 litres}/ \). It was found, that in the case of the same neck, \( V \) is a less determinant parameter at low pressure levels for reaching a small logarithmic decrement \( \Lambda \), but at higher levels \( \Lambda \) increases rapidly with \( V \). Important parameters are the size and the number of the necks /2/ and - if the sound level is higher - also the form of the end of the neck and the method of the excitation.

The resonance frequency of both the Helmholtz resonators and the \( \Lambda/4 \)-tube ones, could be calculated with the same improved formula /3/ but the \( \Lambda \) values as a function of \( V \) show a break point at a conversion of one type of resonators into the another one /see figure/.

\[ \text{Log. dec.} (-) \]

\[ \begin{align*}
0.14 & \quad 120 \text{ dB} \\
0.12 & \quad 140 \text{ dB}
\end{align*} \]

Helmholtz resonator Tube


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ON THE ATTENUATION COEFFICIENT OF AN N-WAVE PROPAGATING INTO A CIRCULAR PIPE

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INTRODUCTION

It is known that attenuation coefficient of $\delta/L$ for N-wave propagating in air is proportional to $\delta/L$, where $\delta$ is excess pressure ratio, $L$ is pulse width. The proportional constant is determined by measurements of wave from at various distances from source in circular pipe.

EXPERIMENTAL ANALYSIS

The attenuation constant of $\delta/L$ for N-wave is (1)

$$\alpha = \frac{1}{\delta} \frac{d}{dx} \left( \frac{\delta}{L} \right) = \beta \left( \frac{\delta}{L} \right) . \quad (1)$$

From eqn. (1), one obtain

$$\frac{\delta}{L} = \frac{1}{\beta} \frac{1}{x + a_0} , \quad (2)$$

where $a_0$ is arbitrary constant. Eqn.(2) is compared with experiment. Fig.1 shows measured waveforms at various distances. One can obtain the accurate waveform from trace on X-Y recorder by sampling of 1 $\mu$-sec. One has to correct the attenuation due to the wall effect from observed value. In principle, the correction should be made on amplitude of each Fourier component. However, the wall effect depends upon $\sqrt{\omega}$, and its magnitude is less than about 1/10 of the attenuation of $\delta/L$. Thus, one can correct $\delta/L$ with good approximation by using the wall loss for frequency corresponded to maximum amplitude among the Fourier component of N-wave. Results obtained is shown in Fig.2. Solid line is for $\beta = (\nu + 1)/\nu$, and dotted line for $(\nu + 1)/\nu^2$. Roughly speaking, the difference between $1/\nu$ and $1/\nu^2$ is the order of the wall loss in the pipe of radius, 3 cm.

REFERENCES

HOLOPHONIE

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INTRODUCTION

L'holophonie est à l'acoustique ce que l'holographie est à l'optique : la restitution exacte d'un champ dans un domaine spatio-temporel étendu. En fait l'essentiel est la reproduction fidèle de l'évolution temporelle du son, alors que la configuration spatiale du champ sonore admet, heureusement, des approximations moins fines.

THEORIE

L'outil théorique de l'holophonie est le principe de Huygens, dont la traduction mathématique traditionnelle consiste dans les formules intégrales de Kirchhoff. L'usage des opérateurs de découpage conduit toutefois (1) à une analyse plus rigoureuse et plus commode.

Soient, dans l'auditorium A, les sources primaires $K^o$ et les sources virtuelles $K^v$ de réflexion et de diffusion. On se propose de restituer dans un voisinage R des auditeurs P (Fig.1) le champ émis par les sources $K^o$ et $K^v$.

On introduit une fonction $s$ égale à 1 dans R et égale à zéro hors de R + S, S étant la zone où seront placées les sources holophoniques $K_h$. Ces dernières sont de deux sortes : des sources de débit $q_h$ et des sources d'impulsion $F_h$. On les obtient aisément à partir du champ acoustique $(p,v)$ régnant dans A :

$$(1) \quad q_h = v \cdot \text{grad} s, \quad F_h = p \cdot \text{grad} s.$$ 

L'énergie acoustique débitée par l'ensemble de ces deux densités de sources est :

$$E_h = s(p \cdot q_h + v \cdot F_h) = pv \cdot s \cdot \text{grad} s$$

où apparaît l'intensité acoustique pv traversant la zone S.

CONSIDERATIONS PRATIQUES

Les formules (1) et (2) décrivent des densités sur S. Le diagramme cardiodie des sources (1) paraît moins essentiel que le respect de la répartition des flux (2). La discrétisation s'effectue en faisant des moyennes sur les formules (1) et (2).

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EIGHTH INTERNATIONAL CONGRESS
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SUR LA MISE SOUS FORME D'ÉQUATIONS INTÉGRALES DE CERTAINS PROBLÈMES D'ACOUSTIQUE NON LINÉAIRE

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INTRODUCTION

La théorie du bruit émis par des fluides en mouvements turbulents (jets) repose sur la donnée implicite de sources acoustiques fictives localisées dans la zone de mélange du jet et reliées directement au tenseur de Lighthill. Nous avons écrit un système d'équations intégrales dont les solutions sont les éléments de ce tenseur.

ANALYSE THÉORIQUE

Nous partons des équations de conservation :

\[ \frac{\partial \rho}{\partial t} + \nabla \cdot (\rho \mathbf{u}) = 0 \]

\[ \frac{\partial \mathbf{u}}{\partial t} + (\mathbf{u} \cdot \nabla) \mathbf{u} = -\nabla p + \frac{1}{\rho} \nabla \cdot \mathbf{T} + \mathbf{f} \]

\[ \rho = \rho_0 + \rho_s ; \quad \mathbf{u} = \mathbf{u}_0 + \mathbf{u}_s \quad \mathbf{T} = \mathbf{T}_0 + \mathbf{T}_s \]

où \( \rho \) est la densité, \( \mathbf{u} \) la quantité de mouvement ; \( p \) et \( f \) sources réelles. Nous posons (1) :

\[ \rho = \rho_0 + \rho_s ; \quad \mathbf{u} = \mathbf{u}_0 + \mathbf{u}_s \quad \mathbf{T} = \mathbf{T}_0 + \mathbf{T}_s \]

On aboutit facilement à l'équation de Lighthill (définie sur la totalité de l'espace E) :

\[ \nabla \times \rho = c^2 \nabla \times \nabla \cdot \mathbf{T} + c^2 \left[ \nabla \times \mathbf{Q} - \text{div} \mathbf{F} \right] = \mathbf{N} ; \quad \nabla \cdot \mathbf{Q} = \frac{2}{3} \nabla \cdot \mathbf{u} \cdot \mathbf{u} = \frac{1}{3} \nabla \cdot \mathbf{u} \quad \nabla \times \mathbf{F} \]

On peut à partir des équations (1) et (2) aboutir aussi à une relation vectorielle (2) :

\[ \nabla \mathbf{F} = \text{div} \mathbf{F} + c^2 \mathbf{F} - \frac{1}{2 \rho} \nabla \cdot \mathbf{Q} + \frac{1}{4 \rho} \mathbf{F} \]

 où \( \mathbf{F} \) vector autre arbitraire fonction uniquement des variables spatiales. Les solutions du système d'équations (3) (4) s'écrivent dans E. (3) :

\[ \mathbf{u} = \frac{\mathbf{u}_0 \cdot \mathbf{r}}{4 \pi r} ; \quad \rho = \rho_0 + \rho_s \quad \mathbf{F} = \mathbf{F}_0 \]

En supposant le gaz parfait et non visqueux on aboutit à une relation intégrale non linéaire en \( \mathbf{T} \) à résoudre dans le volume turbulent :

\[ \mathbf{T} = \frac{1}{\rho_0} \left[ \left( \frac{\mathbf{F}_0}{2 \pi r} \right)^n \sum_{r} \left[ \left( \frac{\mathbf{N}}{\rho_0} \right)^n \right] \right] \]

REFERENCES

RADIATION FROM FLUID-LOADED SEMI-INFINITE PANELS

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This paper discusses the response to a distributed sinusoidal excitation of a semi-infinite taut membrane whose plane is extended by a rigid baffle. The membrane and baffle are in contact with a compressible fluid of infinite extent. Particular attention is paid to the associated acoustic radiation when the radiation is generated in the vicinity of the edge of the membrane, that is, to edge mode radiation, and to the effect of fluid-loading on this type of radiation.

The responses of the membrane and the fluid can be described by a mixed boundary value problem for the acoustic pressure in the fluid. The different boundary conditions apply on two halves of an infinite plane; the problem thus may be solved by using the Wiener-Hopf technique. An explicit solution is obtained.

Asymptotic forms of the integrals describing the acoustic field at large distances from the edge of the membrane are obtained for light and dense fluid loading, and for large and small values of the ratio (in vacuo membrane wavespeed/ acoustic wavespeed). An interesting conclusion is that the often used approximation of neglecting fluid-loading in estimating the membrane response and subsequently estimating the radiation is accurate even for quite dense fluid loading.
ANALYTIC APPROACH TO VARIOUS SURFACE WAVE MODES ON ELASTIC CYLINDERS

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The surface wavenumbers for a solid elastic cylinder imbedded in a liquid are found from the roots of a $3 \times 3$ determinant in the complex wavenumber plane. When the problem of scattering by an elastic cylinder is treated using the Sommerfeld-Watson transformation, the zeros of this determinant give pole-type contributions to the scattered field. There are several classes of zeros involved in this problem (1). The "Whispering Gallery" type combine to form the longitudinal and transverse lateral waves in the limit of infinite cylinder radius. The single Rayleigh type pole goes over to the Rayleigh surface wave in the zero curvature case. In addition, there are modes that are close to the liquid wavenumber. One of these, the Stoneley pole, tends toward the Stoneley pole for the flat elastic half-space in the limit of infinite cylinder radius. The other modes, the Franz type modes, tend toward the liquid wavenumber in the limit of infinite radius. The behavior of all of these modes for large cylinders has been determined analytically by substituting the appropriate Debye- and Airy-type asymptotic expansions into the Bessel and Hankel functions in the determinant. Analytical expressions for the mode trajectories and dispersion curves were then calculated using an iterative scheme. The results of this calculation for a Stoneley wave on a solid aluminum cylinder in water are shown in the figure ($k$ is the liquid wavenumber, $a$ is the cylinder radius, and $k_{s R}$ and $k_{s I}$ are the real and imaginary parts, respectively, of the Stoneley wavenumber). Finally, the residue sums corresponding to these surface waves can be shown to tend toward the surface wave solutions for the flat surface in the limit of infinite cylinder radius. Thus, the relationship between the surface wave modes in the cylindrical and flat cases has been established.

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A METHOD OF IDENTIFICATION OF SPHERICAL AND CYLINDRICAL SHELLS BY ECHO SIGNALS

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INTRODUCTION

Let the incident pulse be sinusoidal and of finite length. The echo signal from a thin spherical or cylindrical shell mainly consists of a reflected pulse and of some series of radiated pulses generated by circumferential and creeping wave modes (1-3). All these pulses have approximately the same form as the incident pulse. There exists (1,4,5) a range of frequency at which the amplitudes of the radiated pulses and the reflected pulse are of the same order.

THEORETICAL ANALYSIS

The method is based on the analysis of theoretically calculated echo signals.

Let the frequency \( \omega_k \) of the incident pulse be of such a value that the first radiated pulse of bending mode has a maximum amplitude in the source point. This maximum amplitude being greater than the amplitude of the reflected pulse specifies the case of a spherical shell and being smaller - the case of a cylindrical shell.

The radius of the shell and the distance to the center/axis of the shell may be determined analogically to the case of a similar rigid object. This is possible due to the fact that the reflected pulse from a thin shell has practically the same arrival time and the same amplitude, as compared with those of a similar rigid object.

The thickness \( h \) of the shell and the dilatational wave velocity \( c \), may be determined by empirical formulae

\[
h = k c \omega_k^{-1}, \quad c = 6.20 R (t_j^{j+1} - t_j^{j}) - 0.05 R c^{-1}\]

where \( R \) is the radius of the shell, \( c \) is the sound velocity in the liquid, \( t_j \) is the arrival time of \( j \)-th radiated pulse of the membrane mode, \( k \) is the coefficient depending on the parameters of the shell material and geometry. An orientated incident pulse may be used for better determining of the arrival times \( t_j^j, t_j^{j+1} \).

REFERENCES

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ACOUSTIC SCATTERING FROM SALT FINGERS

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INTRODUCTION

In the tropical ocean, one often encounters step-function discontinuities of temperature and salinity as a function of depth. One mechanism for the vertical transport of heat and salt between the layers is a salt-finger interface, in which small columns of warm salty water travel downward while columns of cool, fresher water move upward. In the laboratory, it is possible to simulate oceanic salt fingers by placing a sugar solution over a denser salt solution. A finger interface of increasing thickness forms between the well mixed sugar and salt layers.

In a previous paper (1), we reported on the increased attenuation of 1-40 MHz sound propagating horizontally in the finger layer and speculated that this must be due to scattering. In the present paper we report on the observation of the scattered signal.

EXPERIMENTAL

By using an incident sound beam of 3.4 MHz and a separate receiver constrained to move about at a fixed radius from the center of the salt-finger region, the angular behavior of backscattered sound has been investigated. The salt-finger region was constrained to a horizontal diameter of about ten centimeters by a light plastic bag submerged in a tank of pure water. Both the transmitting and receiving transducers were located outside the finger region. Back-scattered sound was observed both from the salt-finger region and from the turbulent convection region immediately above and below the salt-finger interface. While the turbulence-produced scattering was quite random in amplitude and angular dependence, the scattered sound from the finger region appeared at specific angles with respect to the incident sound beam. A diffusive interface (without salt fingers) did not cause any perceptible scattering of the sound.

REFERENCES

The scalar scattering problem of many bodies is considered. Plain wave scattering by the system of many small bodies of an arbitrary form is investigated. It is supposed that the summary field vanishes on the scattered surfaces. The distance between the bodies can be more or less than wavelength or even be of the same order as the wavelength. The interaction of the scattering waves is considered. The formulas for the scattering amplitude are obtained by means of integral equations in all abovementioned cases.

Scattering by the small body with an impedance boundary is considered and formula for the scattering amplitude is obtained. Scattering by a lot of small bodies, which are randomly distributed in space is investigated. Differential equation of Shrödinger type is obtained for the averaged field. A potential in this equation is closely connected with the scattering properties of an elementary volume of random medium, which is formed by randomly distributed small bodies.

Another approach to the electromagnetic wave scattering had been presented in (1–6).

REFERENCES

The Sommerfeld-Watson transformation is applied to the normal mode series describing the scattering of a plane acoustic wave from an elastic cylinder (1). The nature of asymptotic solutions to the secular equation is discussed. The numerical solution of the secular equation is briefly outlined and a display of the zeros of the secular equation in the complex index plane is presented. Here the groups of zeros corresponding to Rayleigh and Stoneley wave and to whispering gallery modes and Franz waves can be segregated and the behavior of these groups can be visualized as certain limits are taken (e.g., the limit to rigid, soft, or fluid cylinders, or the limit of large cylinder radius or high frequency).

Dispersion curves similar to the one shown here give the qualitative results for an elastic cylinder in a liquid. In the limit of high $ka$ (wave number, cylinder radius product) the various modes approach phase velocities characteristic of that mode. The Rayleigh and Stoneley modes (R and S) approach the Rayleigh and Stoneley wave velocities ($C_R$, $C_S$) for a flat surface, and there are an infinite number of whispering gallery (WG) and Franz (F) modes which approach the transverse bulk wave velocity ($C_T$) and the liquid borne wave velocity ($C_1$), respectively.

REFERENCES

EIGHTH INTERNATIONAL CONGRESS ON ACOUSTICS, LONDON 1974

BERECHNUNG DES SCHALLDURCHTRITS DURCH EIN IDEALISIERTES MODELL EINES WALDSTREIFENS MIT HILFE EINES MONTE-CARLO-VERFAHRENS

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EINLEITUNG

Die Untersuchung an einem stark vereinfachten Modell soll Aufschluß über die zeitliche und räumliche Ausbreitung von Schall in einem Streifen mit statistisch verteilten zylindrischen Hindernissen geben. Es ist möglich, durch Hinzufügen weiterer Bedingungen den Realfall besser anzunähern.

BESCHREIBUNG DES MODELLVERSUCHS

Das Problem wird nur in der x-y-Ebene betrachtet. In einem Streifen von \( y = 0 \) bis \( y = 1 \), der in x-Richtung unendlich ausgedehnt sei, sind Hindernisse so verteilt, daß man auf einem Fahrrad beliebiger Richtung mit der Länge 1 im Mittel \( p \) Hindernisse trifft. Die Wahrscheinlichkeit, beim Durchlaufen der normierten Strecke 1 kein Hindernis zu treffen, ist gegeben durch

\[
W(1,0) = \exp(-pl).
\]  

(1)

Es wird der Fall betrachtet, daß eine ebene Welle auf den Streifen trifft, die sich parallel zur x-y-Ebene ausbreitet, wobei beim Antreffen eines Hindernisses isotrope Streuung in der x-y-Ebene erfolgt, die Energie also in der Ebene verteilt.

Bei der Berechnung wird der Weg von "Schallteilchen", die unter einem bestimmten Winkel in den Streifen eintreten und an den Hindernissen Reflexionen erfahren, mit Hilfe eines Monte-Carlo-Verfahrens verfolgt (1,2).

ERGEBNISSE

Man erhält bei einer genügend hohen Zahl von Schallteilchen eine Aussage über die räumliche Energieverteilung, die Richtung und das zeitliche Verhalten (Fig.1) des am Waldstreifen reflektierten und des durchtretenden Schalls, wobei \( p \) über den Streuquerschnitt mit realen Hindernissen in Beziehung gebracht werden kann.

Diese Forschungen wurden mit Unterstützung des Landes NRW am Institut für Technische Akustik der RWTH Aachen durchgeführt.

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The expansion (E. Colton, 78th ASA Meeting, L.A., 1968)

$$\sum_{n=0}^{\infty} A_n \frac{d^n x_n}{dt^n} = F(t) = f_n = F_n e^{i\omega t}$$  \(1\)

can be generalized by introducing Coulomb's laws of electric charges and magnetic poles in the complex form

$$F_E = \frac{(1Q_1)(1Q_2)}{kr^2} \quad \text{and} \quad F_M = \frac{(jM_1)(jM_2)}{\mu r^2}.$$  \(2\)

This form also results in a proof of Maxwell's electromagnetic field equations.

A further generalization ensues with the introduction of differentials of non-integer order. These can be calculated by using the convolution theorem. Although a physical interpretation of such differentials is rather simple, it is suggested that their geometric explanation might require the introduction of lines of finite thicknesses. Such thicknesses would be a function of the 'fact' that two adjacent but not identical points on a line are necessarily a non-vanishing distance apart.

By expanding terms of eqn. (1) into series of their own it is possible to include all of the formulations which have thus far been made to describe nonlinear acoustic oscillations. However, finding solutions to the resulting equation can be rather difficult. If one approaches this solution problem in terms of Pfaffian expressions one obtains inequalities rather than equations. Hence, one uses the notions from probabilistic potential theory to obtain solutions. This requires that one considers the physical continuum variables in terms of particles. In this connection it is of interest that this author was able to derive Heisenberg's uncertainty principle from continuum considerations using the above approach (E. Colton, oral portion of paper 21H1, 7th ICA, Budapest, 1971). It is noteworthy that this approach not only interprets such inequalities in terms of accessibility of points but also implies the existence of an absolute time scale. One can conclude from this that the application to acoustics of mathematical methods used in quantum and in statistical mechanics is not just a fortuitous circumstance but results from the much deeper problem of independence of physical variables.

In order to cover all possible neighborhoods of a point it is also necessary to invert the inequality resulting from the above approach. Physically this implies either a loss of symmetry during some sort of transform or a change in interpretation.
NONLINEAR DISTORTION AND ITS ROLE IN AIRCRAFT NOISE PROPAGATION

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INTRODUCTION

In a previous paper (1) the results were given of a theoretical and experimental investigation into the harmonic components generated during the nonlinear propagation of sinusoidal and narrow-band noise signals in air. In this paper the r.m.s. spectrum pressure level and the distribution of peak amplitudes of the second harmonic component generated during the propagation of broad-band noise are examined.

SPECTRUM PRESSURE LEVEL OF SECOND-HARMONIC COMPONENTS

An analysis is developed for the second harmonic component generated when a moderately intense broad-band noise signal propagates over a considerable distance in a tube having an arbitrary relationship between attenuation and frequency. This is based on the method of determining the response of a nonlinear device with a quadratic characteristic to a noise signal, which makes use of a self-convolution operation on the power spectral density of the signal. Experimental data are presented supporting the validity of the method using bands of noise with centre frequencies in the range 500 Hz to 2 kHz with bandwidth exceeding two octaves (2). The application of the method to the propagation of aircraft noise is discussed.

AMPLITUDE DISTRIBUTION OF NONLINEAR NOISE SIGNALS

An analysis is developed which distinguishes between the peak-amplitude distributions of a noise signal generated by nonlinear distortion and a linear noise signal covering the same frequency range. Given that a narrow-band linear noise signal has a Rayleigh distribution of peak amplitudes \( P(x) \) it is shown that the corresponding distributions for harmonic components obtained as a direct second harmonic \( P(y) \) and as a cross-modulation product \( P(z) \) are given by

\[
\begin{align*}
P(x) &= xe^{-x^2/2} \quad 0 \leq x \\
P(y) &= e^{-y} \quad 0 \leq y \\
P(z) &= 2z K_0(z\sqrt{2}) \quad 0 \leq z
\end{align*}
\]

where \( x, y, z \) are amplitude of the three types of signal and \( K_0(z\sqrt{2}) \) is a modified Bessel function of imaginary argument.

Experimental data are shown to support the analysis. The possibility of detecting the products of nonlinear processes in the presence of linear signals is discussed in the context of aircraft noise.

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ULTRASONIC NONLINEARITY PARAMETERS

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A consistent definition of the nonlinearity parameters of gases, liquids, and solids can be made. By using an appropriate form of the nonlinear wave equation

\[ \rho_0 \frac{\partial^2 u}{\partial x^2} = K_2 \frac{\partial^2 u}{\partial x^2} + (3K_2 + K_3) \frac{\partial u}{\partial x} \frac{\partial^2 u}{\partial x^2}, \]

the nonlinearity parameter \( \kappa = -\frac{3K_2 + K_3}{K_2} \) can be recognized as the negative of the ratio of the coefficient of the nonlinear term to that of the linear term. The nonlinearity parameter has considerable significance for solids\(^1,2\) as well as for liquids and gases. For isotropic solids and the \([100]\) direction in single crystals, \( K_2 = C_{11} \) and \( K_3 = \frac{C_{111}}{C_{11}} \). Thus, \( \kappa = -\frac{3 + \frac{C_{111}}{C_{11}}}{K_2} \). For liquids \( \kappa = 2 + B/A \). For gases \( \kappa = 1 + C_p/C_v \).

For example, we calculate room temperature values: for air, \( \kappa = 2.4 \); for water \( \kappa = 7.0 \); for copper \( \kappa_{[100]} = 5.25 \); for germanium \( \kappa_{[100]} = 2.78 \). But for fused silica \( \kappa = -9.2 \). In our harmonic distortion experiments, the negative nonlinearity parameter implies that the waveform approaches a "backward sawtooth" as the wave progresses.

In addition, the magnitudes of \( \kappa \) for fused silica\(^3\) seem to be sensitive to small amounts of impurities. An OH content of 1200 ppm led to a measured \( \kappa = -9.64 \). The same fused silica with only 5 ppm OH had a measured \( \kappa = -9.25 \).

Figure 1 shows recently measured nonlinearity parameters of germanium\(^4\) and fused silica plotted as a function of temperature. Consistent with other single crystals measured to date\(^5\), germanium \( \kappa \)'s are almost independent of temperature. That for the fused silica sample is not. [Research supported in part by the U.S. Office of Naval Research.]

REFERENCES

The polarization effects of the shear elastic waves can be successfully used as the methods of solid state investigation. In particularly, the elliptical polarization of shear elastic wave, which propagate along the acoustic axis in crystal, arises under direct electrical field and can serve for the investigation of their nonlinear elastic and piezoelectric properties.

The external electrical field takes away the degeneration from the acoustical axis due to the action of the elastic and piezoelectric nonlinearity, that leads to the elliptical polarization of shear waves and changing of the main plane of polarization the wave on some proper distance.

Using the equation of motion in Lagrangian coordinates, boundary condition and condition of quasistationarity for the electrical field, connecting with the elastic field, one can find the nonlinear parameter \( \varepsilon \), responsible for the effect under question.

It was shown that nonlinear parameter determines of the linear and nonlinear elastic and piezoelectric coefficients in some combination. For shear wave velocity changing under external field in \( 3m \) crystals we have:

\[
\Delta \frac{d}{dT} E = \left[ (2C_{44} + C_{155} - C_{144}) d_{22} + C_{444} d_{45} + C_{144} d_{45} \right]/C_{44}
\]

The result of experimental investigation of elliptical polarization of shear elastic wave for crystals \( \text{LiNbO}_3 \) and \( \text{LiTaO}_3 \) on the frequencies 500 - 900 MHz/sec are reported. The dependence of shear wave amplitude from direct electrical field gives the possibility to evaluate the nonlinear parameter, responsible for the rotation of shear wave polarization. This coefficients are \( 1.4 \cdot 10^{-6} \) CGSE for \( \text{LiNbO}_3 \) and \( 3.7 \cdot 10^{-6} \) for \( \text{LiTaO}_3 \), respectively.

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ACOUSTIC EMISSION PRODUCED BY LARGE ALTERNATING STRAINS

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By applying a high alternating strain to single and polycrystalline metals and rocks, and picking up the generated acoustic emission by a wide band pickup device, three types of acoustic emissions have been observed. For a very thin metal layer - in this case lead tin solder - a single Frank-Read source is observed as shown by Fig.1. Calculations agree with Cottrell's (1) equations for a dislocation pile-up. For a thicker sample, odd harmonics are generated, as shown by Fig.2. When the strain becomes as high as $2 \times 10^{-3}$ or higher, a series of non-harmonic responses occur as shown by Fig.3. These are thought to be connected with the Frank mechanism (2) for producing dislocations by a dynamic process. The harmonic generation appears to be the origin of the continuous emission observed whereas the non-harmonic generation is the origin of burst emission.

The harmonic generation has been analyzed over the entire amplitude range and has been shown to be connected with the breakaway of dislocations from pinning points in the lower amplitude range, and with plastic flow in the upper strain range. In both ranges the acoustic emission is proportional to the internal friction $Q^{-1}$ with the constant of proportionality being larger for breakaway.

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INTERACTION OF TURBULENT BOUNDARY LAYERS AND COMPLIANT WALLS

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INTRODUCTION

Interaction of turbulent boundary layers and compliant walls is investigated assuming small compliance. This small compliance can be treated as causing a small perturbation to a basic flow, the turbulent flow along the rigid wall, regarded as known. There then results a linear problem for the flow perturbation, which though linear cannot be solved exactly in general because of the basic flow appearing in the linearised convection terms of the Navier-Stokes equations.

APPROXIMATE SOLUTION

Knowledge of the flow perturbation in the vicinity of the wall is of most interest. However an expansion of the flow perturbation with respect to the wall distance coordinate would require the problem for the flow perturbation to be an initial value problem with velocities and stresses given at the wall instead of a boundary value problem with part of the boundary conditions, the velocities, given at the wall and part of them, the decay conditions, given at infinity. However the wall stresses are the principal unknowns of the problem. Knowledge of these stresses would enable estimation of the change in drag due to the compliance of the wall or of the change in turbulence level and thus of the sound radiated.

Here a procedure for arriving at an approximate expression for the wall stresses is given and the result discussed. The differential equation is integrated with the help of a Green's Function which though not the exact Green's Function of the problem takes into account the essential features of the problem especially the behaviour of the solution at infinity. This integration results directly in a relationship between the wall stresses and wall velocities of the flow perturbation and an integral over the whole flow regime, which is therefore unknown. This integral can however be approximated by a known expression containing again the wall velocities and wall stresses. The result is conveniently stated in terms of Fourier transforms with respect to wall coordinates and time. The final expression will be discussed for a special case of compliant wall.
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FINITE-AMPLITUDE WAVE PROPAGATION IN WATER WITH AIR
BUBBLES

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INTRODUCTION

In water with air bubbles there is a strong dispersion
influential on the form of the finite amplitude sound waves.

FORMULATION OF PROBLEM

Using the expression for the wave number in water with
air bubbles concentration \(n\), it may be found two eqns.
for the particles velocity \(u(x,t)\) for cases \(\omega < \omega_0\) (1)
and \(\omega > \omega_0\) (1’). \(\omega\) - sound frequency, \(\omega_0\) - bubble resonant
frequency,

\[
\begin{align*}
  u_x - auu_t - bu_{tt} &= \begin{cases} 
    du_{ttt} \\
    w^j u dt 
  \end{cases} 
\end{align*}
\]

where \(c_0\) - sound velocity at \(n=0\), \(t=t-x/c_{1,2}\), \(c_1 = c_0/(1 +
2\pi n R c_0^2/\omega^2)\), \(c_2 = c_0\), \(a=(\gamma+1)/2c_{1,2}\), \(w=2\pi n R c_0\), \(d=2\pi n R c_0/\omega^4\),
\(b = \left(4/3\eta + \pi (1/c_v - 1/c_p)^2/2 \rho c_0^2 \right) / \left(2 \rho_0 c_0^2 + 2\pi n R c_0\mu/\omega^4\right)\), \(\mu\) - coeff.

attenuation for bubbles oscillation, \(R_0\) - radius of a
bubble. The approximate solutions of eqns. (1) and (1’)
with boundary condition \(u(0,t)=u_0 \sin t\) for \(q^2 b x < 1, q^2 d x < 1, w x / q < 1\) are (1):

\[
  u(x,t) = \sum_{m=1} B_m(r) \exp(-m^2 r/Re) \sin(mqt + \frac{m^3 r}{D})
\]

where \(B_m\) - Bessel-Fubini-Blackstock coefficients (2),
\(r=au_0 q^2, Re=au_0/bq, D=au_0/dq^2, S=au_0 q^2/w\).

PRINCIPAL RESULTS

Numerical solutions of eqns. (2) and (2’) with parameters
\(r=2.0, m=50, Re_1=1000, D=2000, S=5.0, Re_2 = 50.0\) are shown in
fig.1. The curve 1 shows generation of
solitons, curve 2 corresponds to the
wave form observed in (3).

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Fig. 1 Waveforms of finite amplitude sound
INTRODUCTION

In Ref. (1) the transmission of sound waves in a circular subsonic jet was discussed. The axisymmetric mode was considered and the eigenmodes were determined from the Helmholtz instability of the cylindrical vortex sheet which separates the uniform jet from the surrounding medium. In this paper, results for the first azimuthal mode are presented.

ANALYSIS

From the convective wave equation, the perturbation potential in the jet, \( \phi \sim \exp \left[ i \left( \beta_1 a - M_k a \right) (1 - M) \right] \).

Here \( k_1 \) is the wave number; \( M \) and \( a \) are the Mach number and radius of the jet respectively; \( \beta_1 \) is a complex constant.

The dispersion equation obtained from the kinematic and dynamic boundary conditions on the vortex sheet is solved in the complex \( \beta_1 a \) plane for different values of \( M \) and Strouhal number \( St \) using a numerical technique.

In Fig. 1, the imaginary part of \( \beta_1 a (1 - M^{-1})^{-1} \) is plotted against \( St \) at \( M = 0.2 \) for the first azimuthal mode. The label Mode I denotes the usual mode while for \( 0.7 \leq St \leq 2.8 \) a higher mode (Mode II) exists. Since the solution of the dispersion equation allows for positive and negative values of \( I_m [\beta_1 a (1 - M^{-1})^{-1}] \) to exist, the sound waves decay or grow depending on the sign used. Waves of Mode II type have much larger phase velocities and decay (or growth) rates than Mode I waves.

The characteristics of these two modes are discussed in terms of the radial pressure and velocity distributions in the jet and comparisons with the axisymmetric waves are also given.

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ACOUSTIC DAMPING IN A GAS MIXTURE WITH SUSPENDED, SUBMICROSCOPIC DROPLETS

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INTRODUCTION

The attenuation of acoustic waves in dilute suspensions is theoretically investigated for submicroscopic droplets having radii of about $2 \cdot 10^{-8}$ m. Measured values for the attenuation of shock waves in transonic jets with such droplets formed during rapid nozzle expansion are reported in (1) and (2).

THEORETICAL ANALYSIS

Expressions have been derived for the exchange of mass, momentum and energy between the phases. They include those used by (3) and (4) for large droplets (droplet radii greater than the mean free path of the molecules). For small droplet mass fraction $q_c \ll 1$ and low angular frequencies $\omega \ll [q_c/\tau_M \tau_Q]^{-1}$ an approximation of the attenuation coefficient $\beta$ is then obtained in terms of $\tau_M$ and $\tau_Q$ (relaxation times of the mass and energy exchange processes, respectively) as given by

$$\frac{\beta a}{\omega} \approx \text{const.} \frac{\omega \tau_M}{q_t} \frac{1}{1 + \left(\frac{\omega \tau_M}{q_t}\right)^2 + \left(\frac{\omega \tau_Q}{q_t}\right)^2}^{-1} \quad (1)$$

where $a$ is the velocity of sound in the gas and const. is in our case about 0.16.

CONCLUSION

Fig. 1 compares theoretical and experimental attenuation coefficient profiles. It is seen that the latter can be explained exclusively by mass and energy exchange processes.

REFERENCES

Fig. 1 Attenuation of acoustic energy
AUDIO FREQUENCY ABSORPTION BY FOAMS

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INTRODUCTION

Absorption and phase velocity of sound have been measured in detergent stabilised aqueous foams at frequencies between 600 and 1600 Hz. The effects of variation of cell diameter and viscosity of the aqueous phase have been studied.

EXPERIMENTAL

The foam was produced by blowing a jet of air into water containing a stabilising agent. Glycerol was added to control the viscosity. The foam passed to a tube through which a plane progressive sound wave was passed. A probe microphone measured attenuation and phase change with distance enabling absorption and phase velocity to be calculated. Foam density was measured.

RESULTS

The velocity of sound was in every case greater than the value calculated from the observed density and adiabatic compressibility. (Typical values: 102 ms⁻¹ observed, 85 ms⁻¹ calculated). The absorption rose from 50 to 200 dB m⁻¹ as the liquid viscosity increased from 1 to 10 cp. Changing cell diameter from 1.4 to 1.8 mm had little effect on the absorption but raised the velocity from 102 to 125 ms⁻¹.

THEORETICAL ANALYSIS

The increased phase velocity suggested a relaxation process. Using the simple formulae for a single relaxation, the relaxation time was calculated. The expected absorption was then worked out and found to agree to better than 9% with the measured values in nearly every case.

The cause of the relaxation is sought in the viscous flow of liquid in the cell walls. When the foam is compressed, the incompressible liquid phase has to be redistributed. A simplified cell model is proposed involving the propagation of a Lamb-type wave in the bounding liquid plates consequent upon the injection into these plates of liquid displaced by the foam compression. Analysis shows that this wave is rapidly damped, leading to a localisation of viscous shear near the cell corners where the liquid motion is further modified by the presence of the Plateau borders. Examination of wave behaviour in this cell model leads to propagation parameters of a similar order to those observed.
ÉTUDE DES ONDES DE FROTTEMENT

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INTRODUCTION
Dans un précédent travail (1) nous avons étudié le domaine d’existence des vibrations de frottement. Cette étude nous a amenés à étudier les ondes de contact découvertes par Schallamach lors d’un glissement.

EXPERIENCE
Si on fait frotter une sphère sur une piste en caoutchouc ou inversement, on s’aperçoit qu’au point de contact il y a parution d’ondes qui défilent à une vitesse plus grande que la vitesse de glissement. Un montage expérimental approprié permet de mesurer cette vitesse et de contrôler si nous avons bien affaire à des ondes de propagation.

THEORIE
Pour expliquer ce phénomène, on a analysé le régime transitoire qui fait passer de la configuration de Cattaneo (cylindre au contact le long d’une génératrice avec couple appliqué et sans déplacement relatif) à celle de Carter (même problème que précédemment mais avec déplacement relatif). La théorie montre qu’une zone d’adhérence isolée se déplace de l’arrière vers l'avant de la zone de contact et inversement pour une zone de glissement ; ce qui correspond aux observations expérimentales.

REFERENCES

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ACOUSTICS OF SNOW AND ICE

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INTRODUCTION

The results of studies of mechanical and acoustic properties of freshwater ice, sea ice, glaciers and snow cover are discussed. Acoustic methods permitted us to make extensive measurements of elastic properties of different kinds of ice and snow (elastic and shear moduli, Poisson's ratio, viscosity coefficients, etc) as a function of temperature (0-30°C), density (0.2-0.92 g/cm³), salinity (0-7‰) and static pressure (0-50 atm). The measurements made it possible to predict average values of mechanical properties of sea ice cover using the data on air temperature and salinity of ice. Methods and results of study of attenuation and scattering of sound energy in the ice are given below.

EXPERIMENTAL

The empirical equation for the dependence of the coefficient of sound attenuation in pack ice (α in dB/m) on the frequency (f in MHz) in the frequency range from 200 KHz to 1,000 KHz has been obtained. α = c₁f + c₂f² where

\[ c₁ = 4.8 \times 10^{-2} \text{ dB/MHz} \quad \text{and} \quad c₂ = 4 \times 10^{-5} \text{ dB/MHz}^2 \]  (1)

Dynamic viscosity coefficients (η) of polycrystalline freshwater ice at -7°C were estimated from measurements of logarithm decrement of bend waves (2) over frequency range from 5 Hz to 400 Hz (η = 2 \times 10^5 ÷ 3 \times 10^7 poise).

The longitudinal wave velocity in the snow as a function of density has a specific feature, characterized by the minimum, on the left side of which the velocity increases with the density decrease, and on the right a reverse dependence is observed (3).

THEORETICAL ANALYSIS

Experimental data obtained are in good agreement with Zwicker and Koster's theory of sound propagation in porosity granulated media.

REFERENCES


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I.- Dans le présent exposé on compare les solutions fondamentales en espace libre $G_S(\Sigma)$ et $\Gamma_S(\Sigma)$ de deux opérateurs $L$ et $\Lambda$ ; $L$ et $\Lambda$ sont des opérateurs décrivant le même phénomène physique dans deux milieux différents, et une condition de Sommerfeld convenable assure l'unicité de $G_S(\Sigma)$ et $\Gamma_S(\Sigma)$. Utilisant une expression stationnaire et une méthode de Ritz-Galerkin, la différence $G_S(\Sigma) - \Gamma_S(\Sigma)$ est développée en une série dont chaque terme s'exprime par une fonctionnelle soit de $G_S(\Sigma)$ soit de $\Gamma_S(\Sigma)$ ; quelle que soit la différence des opérateurs $L$ et $\Lambda$, la convergence est toujours assurée. Ce résultat est proche, mais d'une application plus générale de celui de S. Bergman.

II.- A titre d'exemple, considérons l'équation de Helmholtz, et définissons $L$ et $\Lambda$ par :

$$ L = \Delta + k^2 [1 + p(x)] \quad p(x) = \text{fonction à support borné} \omega ; \quad \Lambda = \Delta + k^2. $$

Soit $\{\varphi_n(x)\}$ une base quelconque de $L^2(\omega)$ à laquelle on associe les suites de fonctions :

$$ \hat{\psi}_n(x) = k^2 \int_{\omega} p(y) \varphi_n(y) \Gamma(x,y) \, dy , \quad \psi_n(x) = \varphi_n(x) + \hat{\psi}_n(x). $$

La différence $G(S,\Sigma) - \Gamma(S,\Sigma)$ est alors représentée par la série :

$$ (\ast) \quad G(S,\Sigma) - \Gamma(S,\Sigma) = \frac{\hat{\varphi}_o(S) \hat{\varphi}_o(\Sigma)}{\int_{\omega} k^2 p(x) \varphi_o(x) \varphi_0(x) \, dx} \sum_{n=1}^{\infty} \frac{\hat{S}_n[\hat{\psi}_i(S)]}{\hat{D}_n-1} \hat{D}_n $$

expression dans laquelle :

$$ \hat{D}_n = \text{déterminant} [k^2 \int_{\omega} p(x) \varphi_j(x) \varphi_i(x) \, dx ; i=0...n, j=0...n] . $$

$\hat{S}_n[\hat{\psi}_i(S)]$ est le déterminant obtenu en remplaçant la dernière colonne de $\hat{D}_n$ par $\{\hat{\psi}_i(S)\}$. (Pour plus de détails voir § Référence).

III.- Du fait que la base $\{\varphi_n\}$ est arbitraire, il devient alors loisible de la choisir de façon à maximiser le taux de convergence de la série représentative (\ast). Sur des exemples unidimensionnels on montre qu'un choix judicieux de $\varphi_o$ permet d'obtenir une bonne approximation de la solution en réduisant la série à son seul premier terme.

Le résultat, que nous présentons ici, est un cas particulier d'une théorie générale, valable pour toute équation scalaire de la physique mathématique, en particulier l'équation des plaques minces d'épaisseur et de densité variables libres ou couplées avec un fluide.

REFERENCE

INTRODUCTION

In addition to the turbomachinery noise generated due to high speed unsteady flows interacting with blades and vanes, high intensity noise can also be generated by the passage of velocity and temperature "eddies" through sufficiently large pressure gradients such as exist across turbine blade and vane rows. Physically, noise is generated because axial velocity fluctuations and variations of density due to temperature fluctuations convecting with the fluid, produce spatial and temporal fluctuations in momentum. As these fluctuations in momentum pass through large pressure gradients, differences are experienced in the rate of acceleration or deceleration of fluid elements relative to the main stream that give rise to acoustic and vorticity waves. This noise generating mechanism may contribute to low frequency core engine noise observed from high bypass ratio turbofan engines. Core noise may be the dominant noise source of future turbofan engines, as fan noise is further reduced with increased inlet and aft duct treatment.

THEORETICAL ANALYSIS

The analysis involves the solution of the three-dimensional linearized equations of continuity and momentum in conjunction with the satisfaction of appropriate steady and unsteady jump conditions across a vane or blade row which is modeled as an actuator disc. For given upstream vorticity (turbulence) and entropy (temperature fluctuations) waves convecting with the fluid and interacting with the actuator disc, the resulting upstream and downstream propagating acoustic waves are determined along with the downstream shed vorticity and entropy waves. The inputs required are the correlation volumes of the turbulence and temperature fluctuations and the steady fluid flow properties upstream and downstream of the actuator disc.

RESULTS

Application of the analysis shows that spanwise and circumferential variations in axial velocity and temperature interact with the actuator disc and give rise to vorticity waves that convect downstream of the blade row with the fluid. If axial variations are also included, they are found to cause upstream and downstream propagating acoustic waves at frequencies identical to that of the incoming upstream entropy and vorticity waves. The greater the degree of spanwise and circumferential coherence of these axial fluctuations, the greater the sound energy produced. The radiated sound intensity is also found to depend on the change in pressure across the rotor and the intensity of the temperature and axial velocity fluctuations. The results are compared with internal and far field low frequency core noise data obtained from a full scale JT8D engine tested at Pratt & Whitney Aircraft for the specific purpose of studying core engine noise.
A THEORETICAL EVALUATION OF THE ACOUSTIC RESPONSE OF A FREE TURBULENT JET AND SOME ASPECTS OF RELATIVE AMPLIFICATION AND FREQUENCY SELECTION

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INTRODUCTION

The transverse and axial stratifications inherent in a jet flow environment makes the propagation characteristics of the medium frequency selective. Some aspects of this are discussed.

ANALYSIS

The reduced form of the wave equation governing sound propagation in a homogeneous medium is of the form

\[ \frac{\partial^2 p}{\partial \eta^2} + \alpha \frac{\partial p}{\partial \eta} + \beta^2 p = 0 \]  

For one, two and three dimensional waves, \( \alpha = 0 \), \( 1/\eta \) and \( 2/\eta \) respectively. In the first case propagation takes place with no change of amplitude. In 2 and 3 dimensional cases, the amplitude of the sound pressure decays as \( 1/\eta \) and \( 1/\eta \) respectively. It is the coefficient \( \alpha \) that controls the rate of decay of the sound and the extent of the near field. In a jet flow environment, the governing wave equation in spherical coordinates is of the form

\[ f_1 \frac{\partial^2 p}{\partial r^2} + \frac{2}{r} \left( 1 + f_4 \right) \frac{\partial p}{\partial r} + \frac{\beta^2 p}{r^2 + \left( \cot \theta + f_3 \right)} \frac{\partial^2 p}{\partial \theta^2} + \frac{1}{r^2 \sin^2 \theta} \frac{\partial^2 p}{\partial \phi^2} + \frac{1}{c^2} \frac{\partial^2 p}{\partial \tau^2} + f_4 p = 0 \]

where \( f_1 \) are functions of the frequency, flow and flow gradients. \( f_3 \) and \( f_4 \) modify the directivity and the wavelength \( \beta \) modifies the rate of decay of the amplitude. Depending on the sign of \( f_2 \), the decay may be faster or slower than \( 1/r \). The solution may be written in the form

\[ p = A P(0)(1/r) \exp(-jkr) \]

where \( P \) and \( k \) are obtained by numerical integration. The imaginary part of \( k \) reflects the deviation of the solution from \( 1/r \) decay. A negative imaginary part will represent "relative amplification" but in fact represents a decay slower than \( 1/r \). Computed results show that this relative amplification is a function of \( (\omega D/c) \) and \( r/D \). Fig. 1 shows the variation of the imaginary part of radial wave vector \( k \), it is characterized by two peaks. For high frequencies the peak nearest the nozzle exit is larger. The peaks for the low frequencies occur further downstream.
INTRODUCTION

Recent experimental measurements of noise from heated subsonic jets have displayed, for the noise intensity perpendicular to the jet, a change from the $U^8$-power law for cold jets to a $U^4$-power law for heated ones, increasingly effective with an increasing temperature ratio and decreasing Mach number. This different power law must be attributed to additional sound sources within the flow. Earlier theoretical discussion of these sources, however, led to discrepant results and that to a $U^4$-law (1) and $U^6$-law (2), respectively.

ANALYSIS

The flow is discussed with the help of the method of matched asymptotic expansions separately for an inner region where hydrodynamical characteristics dominate and for an outer region where the sound field prevails. The actual calculations can then be confined to the inner region since the solution for the outer region depends only on the multipole-character of the inner solution. For a first inner approximation, compressibility effects are negligible and the flow field can approximately be described by an inhomogenous Laplace-equation or its corresponding integral form in analogy to the form deduced in (3).

Accounting for matching conditions, a $U^4$-power law is possible only if either monopole sources with a strength $\sim U^2$ or dipole sources with a strength $\sim U$ exist. The latter case could be realized only by surface tension effects between the cold and the heated gases. This of course disagrees with experience. Monopole sources exist if there is a mass flow between the heated and the cold gas due to the diffusion caused by the temperature gradient. This effect can be included by abandoning the usually applied assumption $dS/dt = 0$ (S entropy). This seems at first unrealistic since, from our knowledge of turbulent flow, such a mass flow should be negligible in a first approximation. However, realizing that noise generation by monopoles is, by a factor $M^{-4}$ (M Mach number), more effective than by quadrupoles, it then seems quite possible that diffusion effects, though hydrodynamically small, can be the major noise generation mechanism.

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In previous work (1,2) an open-ended formalism has been developed which permits unambiguous identification, in arbitrary fluid flows, of 'acoustic', 'turbulent' and 'thermal' components of the fluctuating motion. In this formalism, the linear momentum density is regarded as the principal dependent vector function to be determined, and its fluctuating part, for time-stationary fluctuations and an arbitrary mean flow, can be represented, without any further loss of generality, as the linear superposition of three components. These components, respectively, can be interpreted physically as the 'acoustic', 'turbulent' and 'thermal' components of the linear momentum density, by consideration of the forms to which the representation reduces in the respectively appropriate limiting cases from which these descriptive concepts are derived.

This identification of linearly superposed acoustic, turbulent and thermal components of the linear momentum density also leads, without any further assumptions, to representation of the total fluctuating enthalpy (energy) density flux as a corresponding linear superposition of acoustic, turbulent and thermal components.

The fluctuating energy density itself, in this formalism, appears as a superposition of separately identifiable acoustic, turbulent and thermal components together with three cross-components: acoustic-turbulent, turbulent-thermal and thermal-acoustic.

These representations provide a complete, exact and useful framework for identification and consequent understanding of both local and global mechanisms for phenomena, such as 'generation of sound by aerodynamic means', which involve acoustic/turbulent/thermal energy exchange in fluctuating flows of compressible fluids.

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SOME EXPERIMENTS ON THE INFLUENCE OF SOUND ON THE STRUCTURE OF LOW SPEED JETS

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INTRODUCTION

It has recently been suggested (1,2) that the structure of jet turbulence could be important in understanding jet noise. We have therefore carried out experiments on jets at Reynolds numbers in the range 1 - 2x10^4.

EXPERIMENTS

Firstly, hot wire anemometer measurements were made at four positions in the jet stream issuing from a 50mm diameter circular nozzle at speeds of 2.6 and 6.3m/s, while the jet was excited by sinusoidal sound waves from a loudspeaker placed on the jet axis upstream of the jet exit. At 6.3m/s the measured velocity fluctuations were much larger than those of the unforced system over a range of forcing frequency f from 20-300Hz. The maximum response, A, and the associated forcing frequency f both varied slowly with distance. The range of f over which the effect was observed increased rapidly with A and had a maximum value about 3 diameters downstream. Much more complex effects were observed at 2.6m/s in roughly the same Strouhal number range. Flow visualization was also performed using a smoke probe and a stroboscope. Stable vortex rings were observed whose speed was locked to f, whose spacing was roughly proportional to jet speed and whose definition decreased as A decreased and f increased.

Finally experiments were carried out in a water jet (not deliberately forced) using a hydrogen bubble visualization technique, which also showed a vortex ring structure. These rings showed several modes of instability, one of which involved a spinning motion. Cine films suggested that vortex ring formation was a result of a separate high growth rate instability of the shear layer.

CONCLUSIONS

The vortex structure of a low Reynolds number jet can be strongly influenced by sound, a conclusion which might also apply to high speed jets if the effect scales with a Reynolds number based on eddy viscosity.

REFERENCES

SOUND RADIATION FROM TURBULENT JETS AT LOW MACH NUMBER

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Over the past two years, the range and accuracy of basic jet noise measurements have both been significantly improved. The unexpected nature of some of the results has led to a re-examination of aerodynamic sound theory, with particular reference to the effects of non-uniform fluid density.

EXPERIMENTAL OBSERVATIONS

It is now clear that jet noise is increased by heating the jet above the temperature of the ambient fluid, provided the jet Mach number is below about 0.5. Measurements at NGTE and SNECMA [1], Southampton [2] and Lockheed-Georgia [3] have shown in detail how the radiated intensity at right angles to the jet (in third-octave bands) varies with jet exit velocity, for different exit temperatures. Whereas the unheated jets follow the expected 8th power dependence, the heated jets tend (at the lowest velocities) towards a 4th power dependence on jet velocity, for a given Strouhal number.

PRELIMINARY CONCLUSIONS

A 6th power dependence for heated jets would be easy to explain [4], as due to scattering of the pressure field in the mixing region by density variations. However, a genuine 4th power dependence does not appear possible on theoretical grounds, except through fluctuations in the rate at which differences in temperature are dissipated. The explanation is more likely one or more of the following: (a) Exit-plane velocity fluctuations. Such fluctuations could be caused by hot spots entering the nozzle from upstream, in view of the high contraction ratios used. (b) Acoustic diffraction by the hot jet, as described by Mani [5]. (c) Breakdown of similarity in the turbulent flow, at the rather low Reynolds numbers associated with the low-velocity data. This could be checked by tripping the nozzle boundary layer.

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ÜBER DEN EINFLUSS PULSIERENDER QUellen UND DIPole AUF
DIE BEWEGUNG EINER WIRBELSCHicht HINTER EINER
HALBUNENDLICHEN PLATTE

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Einleitung

Theorie
Für die Wellenbewegung einer unendlich dünnen Grenzschicht mit zur Grenzschicht symmetrischen Randbedingungen wird eine allgemeine Lösung angegeben. Ihre Ableitung ergibt sich wie üblich aus der Gleichheit der Auslenkungen der Ränder beider Strömungsgebiete und der Gleichheit des Wechseldruckes zu beiden Seiten der Wirbel schicht. Man hat für die Geschwindigkeit $v_0$ an der Wirbel schicht im Bereich des ruhenden Mediums:

$$v_0 = \text{Re} \left[ e^{-i\omega t} \cdot \frac{w}{u} \left( e^{\lambda_2 x} \int e^{-\lambda_2 x} v_{oq}(x) \, dx - e^{\lambda_1 x} \int e^{-\lambda_1 x} v_{oq}(x) \, dx \right) \right] \quad (1)$$

mit $\lambda_1 = \frac{w}{u} (i + 1)$; $\lambda_2 = \frac{w}{u} (i - 1)$

darin ist $v_{oq}$ die auf der x-Achse induzierte v-Geschwindigkeitskomponente einer äußeren Anregung ohne Vorhandensein der stationären Strömung $\bar{u}$; $w$ ist die Kreisfrequenz.

Für eine anregende Quelle oder einen Dipol in großer Entfernung $y_0 \gg x$ senkrecht über dem Ende der halbunendlichen Platte hat man z.B.:

$$v_0 = \text{Re} \left[ e^{-i\omega t} \cdot K \cdot \frac{w}{u} \sqrt{\frac{1}{\lambda_2}} e^{\lambda_2 x} \text{erf} \sqrt{\lambda_2 x} - \frac{1}{\lambda_1} e^{\lambda_1 x} \text{erf} \sqrt{\lambda_1 x} \right] \quad (2)$$

Die Konstante $K$ ist für eine Quelle mit der Stärke $Q$

$$K_Q = -\frac{Q}{4\pi} \sqrt{2/y_0} \quad (3)$$

und für einen Dipol mit der Stärke $D$ und dem Richtungswinkel $\varphi$ mit der waagerechten:

$$K_D = \frac{D}{4\pi} \frac{1}{y_0^{3/2}} \cos (\varphi - \pi/4) \quad (4)$$

Literatur
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PHENOMÈNE ACOUSTIQUE LIÉ À LA RENCONTRE D'UNE PERTURBATION ET D'UNE DISCONTINUITÉ

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INTRODUCTION

Le problème de l'interaction entre une onde de choc et une faible perturbation est traité analytiquement. Les résultats théoriques sont comparés aux résultats expérimentaux dans le cas d'un tourbillon (perturbation de vitesse rotationnelle). On détermine en particulier les champs de pression et de densité, et la déformation du choc, après l'interaction. On remarque surtout la création d'une onde de pression acoustique centrée sur le tourbillon.

ÉTUDE EXPERIMENTALE

L'étude est faite dans un tube à choc : une onde de choc réfléchie interfère avec un tourbillon créé par une aile dans un écoulement subsonique. Un film ralenti obtenu par cinématographie ultra-rapide (strioscopie et ombrographie) permet d'observer la déformation du choc et de déterminer les variations de densité (quatre zones de compression et de dépression) sur le front de l'onde acoustique.

ANALYSE THÉORIQUE

Toute faible perturbation incidente est décomposée suivant toutes les directions et tous les nombres d'onde, en ondes élémentaires planes. En particulier, le champ de vitesse rotationnelle du tourbillon détaillé :

\[ q(r,t) = \frac{\Gamma}{2\pi r} \left[ 1 - \exp\left(-\frac{r^2}{4\nu t}\right) \right] \]

est décomposé en ondes élémentaires planes de cisalement.

L'interaction du choc et de chacune des ondes planes élémentaires est analysée dans un système approprié d'axes de référence rendant le problème permanent (choc et ondulation élémentaire immobiles).

Les conditions aux limites (relations du choc oblique) et l'équation différentielle de l'écoulement perturbé tiennent compte de la déformation du choc inconnue. On peut ainsi exprimer toutes les grandeurs après interaction par des fonctions linéaires : soit d'une seule variable \( \omega / \bar{U} \) (onde de cisalement), soit de deux variables \( \omega / \bar{U} \), \( p_\tau / \bar{P}_\tau \). La résolution de l'équation différentielle conduit à un système d'équations linéaires derrière le choc.

L'intégration numérique suivant toutes les directions et tous les nombres d'ondes donne le champ total et la déformation finale du choc après interaction.

Cette méthode d'analyse peut être généralisée au cas de tous types de perturbation et de discontinuité.

REFERENCES

INTRODUCTION AND EXPERIMENT

Lip noise was measured using a 1-1/2 inch diameter model jet. For Mach numbers from 0.3 to 0.7 the lip noise intensity was measured at 90° to the jet axis, after filtering out the lower frequency free jet noise component. As is seen in Fig. 1 the intensity varies as the sixth power of the jet velocity. A 2mm probe tube was flush mounted inside the lip and pressures measured there were correlated with side-line sound pressures, yielding correlations over 10%. Cross-correlations between sound pressures measured on opposite sides of the jet show that the radiated sound is out of phase in opposite directions (as with dipoles). These sound field correlations show discrete peaks due to near and far side lip, dipole sources.

THEORETICAL ANALYSIS

To explain these correlation measurements, Curle's theory (1) was used, assuming that observation point is far away compared with D and the sound wavelength, and that the flow is subsonic. One finds for the cross-correlation of the far field sound, with \( p' \) and \( p'' \) the far-field pressures,

\[
\langle p' \cdot p'' \rangle = \frac{\gamma - 1}{\gamma + 1} \left( \frac{\rho}{\rho_0} \right) \frac{1}{r^2} \int_0^{2\pi} \cos \theta' \cos \theta'' \cos (\phi - \phi) \: P_{\theta'} \: e^{i \theta'} \: e^{i \theta''} \: d\theta
\]

with \( P \) the cross-correlation of the static lip pressure at two points \( \theta' \) and \( \theta'' \) and \( \phi \) is the azimuthal angle between the field points. We assume that the flow inside the nozzle is approximately frozen.

When we restrict to high frequencies, this expression correctly predicts the observed correlation peaks, their signs, and their amplitudes.

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PROPAGATION OF SOUND THROUGH A SPREADING JET

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INTRODUCTION

The sound propagation of harmonic disturbances through a real jet flow field, which contains mean flow gradients in pressure and velocities, is studied (1). A finite difference solution is obtained for an axisymmetric jet with Mach number $M_j = 0.62$ and nozzle diameter $d_0 = 0.062$ meter.

METHOD

The convected wave equation for pressure is derived. The mean flow field is obtained experimentally, Fig. 1. Equations are solved numerically for the phase and amplitude of the pressure between two concentric circles with origin at the source. The inner and outer boundaries have radii 1/4 and 100 jet diameters, respectively. The solution of the system of nonlinear difference equations is obtained by a Newton-type iterative scheme.

RESULTS AND CONCLUSIONS

The directivity, with the convection plus refraction effects, at various distances from the source for a point source location $2d_0$ downstream on the centerline is shown in Fig. 2. A comparison between the numerical result and available experiment (2) at 100$d_0$ is also shown in Fig. 2 and is qualitatively in general agreement. An apparent improvement from a previous calculation (3) is attributed to a fuller description of the flow field and its formulation. Scattering due to turbulence has been neglected; however, if included, it is believed that it will further improve the directivity at small angles from the jet axis.

Fig. 1. Flow Field of the Jet

Fig. 2. Noise Field Directivity

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EIGHTH INTERNATIONAL CONGRESS ON ACOUSTICS, LONDON 1974

EXPERIMENTS ON THE SOUND RADIATION FROM ORDERLY STRUCTURES IN TURBULENT JET FLOW

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Strong wave-like correlations of the velocity or the pressure in turbulent circular jet flow have been observed (1-3). Michalke (4) suggests that the main part of the sound radiated from the turbulent jet originates from these coherent structures due to instability waves in the jet. Cribb (5) calculates the "excess noise" caused by the interaction of instability waves with the nozzle.

It is our aim to measure the radiation efficiency of the axisymmetric instability mode to begin with and to compare it to the radiation efficiency of the total jet.

The nozzle (D = 2.0 cm) is situated in the wall of an anechoic chamber. A sinusoidal instability wave is excited by a loudspeaker in the settling chamber. Curve (a) in Fig. 1 shows the far field pressure of that loudspeaker radiating through the nozzle with superimposed jet flow. By some additive loudspeakers driven in antiphase and radiating through an annulus around the nozzle the sinusoidal part of the sound pressure is compensated at 60° (curve (b)), thus, the monopole-like radiation of the nozzle, but also a part of the sound from the instability wave is cancelled. Curve (c) shows the far field pressure of the undisturbed jet and curve (d), finally, is the expected "compensated" directivity pattern for air at rest (the curve really measured is more irregular). The comparison between the curves (d) and (b) shows that the "compensated" directivity pattern is changed completely by the flow. This can mainly be regarded as an effect of the instability wave and the refraction due to the flow field. The radiation efficiency, however, cannot yet be evaluated from these measurements, therefore, we intend to excite the instability wave by a longitudinally vibrating nozzle in future experiments.

REFERENCES


Fig. 1 Filtered far field pressure (1.65 kHz; 150 m/s)
EIGHTH INTERNATIONAL CONGRESS ON ACOUSTICS, LONDON 1974

NOISE GENERATION BY OPEN TURBULENT PRE-MIXED FLAMES

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ABSTRACT

Previous work (1) on noise generation by a small open burner stabili-
sed turbulent pre-mixed flame carried out elsewhere showed that such
a flame may be represented acoustically as an assembly of monopole
sound sources distributed throughout the reaction zone of the flame.
Over a limited frequency range a strong correlation was observed be-
tween the far field sound pressure and the first time derivative of
the heat release rate in the flame. The latter quantity was measured
by monitoring the emission intensity of free radicals, C2 or C H,
whose existence is confined almost solely to the reaction zone of the
ethylene/air flames studied in this work. Owing to the large amount
of electronic noise generated by the photomultiplier used to measure
the emission intensity, the correlation between the two quantities
could only be investigated over a restricted frequency range.

The present work was intended to confirm these observations for
a single ethylene/air flame and to extend the investigation to a
double flame system. The same techniques were applied but by using a
superior photomultiplier it was possible to investigate the corre-
lation over a wider bandwidth than previously.

Good qualitative correlation was observed over a limited frequen-
cy range for both the single and double flames. The frequencies bey-
ond which the correlation deteriorated were identified and for the double
flame system were found to depend on the spatial separation of the two
flames. Within the frequency range for which good correlation was
obtained, reasonable quantitative agreement was also obtained between
the measured sound pressure level and that calculated from the meas-
ured derivative of the emission intensity using a simple expression
based on monopole source theory.

The significance of the results obtained with the double flame
system is discussed in relation to the feasibility of a proposed
noise suppression technique.

REFERENCES

AN ANALYSIS OF JET TURBULENCE FROM MICROPHONE MEASUREMENTS

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Measurements of the rms values of turbulent pressure fluctuations \( \overline{\sigma} \) were made with small, streamlined microphone probes in a free subsonic jet. The preliminary tests confirmed Fuchs' (1) assertion that the \( \overline{\sigma} \)'s measured with these probes are, for practical purposes, those which exist in the flow independent of the presence of the probe.

Subsequently, the relationship between \( \overline{\sigma} \) and measured rms velocity \( \overline{u} \) (via hot wire) was studied at a distance of three nozzle diameters from the jet exit plane \( (x/D=3.0) \). Some typical results for the ratio \( \overline{\sigma}/\rho \overline{u} \overline{u} \) (\( \rho \) is the mean density and \( \overline{u} \), the mean jet nozzle discharge velocity) are plotted as a function of \( \overline{u} \) in Fig. 1. The ratio is independent of \( \overline{u} \) and is approximately the same both for unfiltered measurements and for those filtered at a Strouhal number of 0.45 (\( St = fD/\overline{u} \), where \( f \) is the nominal filtering frequency and \( D \) is the jet nozzle diameter). The values are greater on the jet axis \( (r/D=0) \) than in the middle of the mixing region \( (r/D=0.5) \).

The unique configuration of variables \( \overline{\sigma}/\rho \overline{u} \overline{u} \) stems from the following relationship for \( \overline{\sigma} \) and \( \overline{u} \) as derived for a frozen pattern of turbulence traveling at the convection velocity \( U_c \):

\[
\frac{\overline{\sigma}}{\rho \overline{u} \overline{u}} = 1 - \frac{U_c}{\overline{u}} + O(\overline{u}/U_c)
\]  

(1)

For \( \overline{\sigma}/\overline{u} \ll 1 \) the equation can be linearized; then \( U_c \) is calculated from it using our experimental data and plotted in Fig. 2 as a function of Strouhal number. \( U_c \) at \( r/D=0 \) agrees with direct measurements of Fisher and Davies (2) at \( r/D=0.5 \). However, our results for \( r/D=0.5 \) (which take some nonlinear kinetic energy terms into account) deviate from their measurements.

![Fig 1 Ratio of Pressure and Velocity Fluctuations](image1)

![Fig 2 Effect of Strouhal Number on Convection Velocity](image2)

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In jet flows the basic noise generation sources are considered to be turbulent eddies moving along with the mean flow at some reduced speed. The radiated field from these sources is influenced by two basic dominant effects, that of convection and that of refraction. It is of some considerable interest to be able to predict these effects throughout the full-scale jet engine operating range of jet velocities and temperatures.

Some basic analysis exists which has been verified to a limited extent (1) by means of observations of the radiation behavior of a noise source injected by a probe within the jet flow. However, the range of experimental conditions was rather limited and the work presented here is an effort to verify and extend the available basic data as a means of further understanding the parametric effects as well as improving the degree of confidence in currently available analysis.

The experimental procedure is simple, in that a high intensity noise source is injected into the flow of a 2\" diameter jet and the far-field directivity is measured using a boom mounted microphone, the entire system being within a new 10,000 cubic foot anechoic room. The position of the injected source has a profound effect on the directivity and is a significant parameter in this experimental study.

The range of experimental conditions covered are as follows; the frequency from 500 Hz to 6 KHz, with velocities up to Mach 1.7 and jet temperatures from ambient to 500°F.

As far as is possible the results of the experiments are correlated with theoretical developments with the major emphasis on subsonic effects.

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Eighth International Congress on Acoustics, London 1974

Untersuchung von Freistrahl Turbulenz im Hinblick auf die Schallzerzeugung

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Einleitung


Experimente

Nach ersten Ergebnissen axialer, radialer und azimuthaler Druckkorrelationen in (1) wurde in (2) ein neues Entwicklungschema vorgeschlagen, wonach die Turbulenz (und entsprechend das abgestrahlte Schallfeld) vollständig in azimuthale Komponenten der Ordnung m zerlegt werden. Experimentell erfolgt diese Zerlegung für die einzelnen Frequenzzentrale aus radialen und azimuthalen Raumkorrelationen in Ebenen senkrecht zur Symmetrieachse des Strahls. (H.V. Fuchs, soll erscheinen in AGARD CP 'Noise Mechanisms'). Man erhält so die Energiespektren der verschiedenen Azimuthalkomponenten z.B. für die Druckschwankungen in der Scherschicht. In Bild 1 sind zur axialsymmetrischen Komponente m = 0 (Ringstruktur) zusätzlich Spekten für m = 1, 2 und 3 superponiert. Im wichtigen Strouhalzahlbereich 0.1 bis 1.0 ist für m > 3 nur wenig Schwellungs-energie vorhanden. Diese starke Konzentration der Turbulenz in niedrigen Azimuthalkomponenten trifft zusammen mit der theoretischen Aussage in (2), daß solche Komponenten besonders wirkungsvolle Schallgeber sein können.

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(2) A. Michalke, Z.Flugwiss. (1972) 20,229

Bild 1: Analyse der turbulenten Druckschwankungen im Freistrahl
EIGHTH INTERNATIONAL CONGRESS ON ACOUSTICS, LONDON 1974

OPTISCHE KOMPENSATIONSMESSUNGEN ZUR ABFLUSSBEDINGUNG AN EINER DÜSENAUSTRITTSKANTE: EINE UNTERSUCHUNG IM ZUSAMMENHANG MIT DEM SCHALLDURCHGANG BEI DURCHSTRÖMTEN DÜSEN

Beichert D
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EINLEITUNG


EXPERIMENTELLE ANORDNUNG

Mit einer neuen optischen Kompensationstechnik wurde die Wellenbewegung einer akustisch gesteuerten freien Grenzschicht am Düsenaustritt untersucht, Bild 1. In den äußeren Rand der Grenzschicht wurde ein Rauchfaden eingeblasen, der durch einen fokussierten Laser-Lichtstrahl seitlich beleuchtet wurde. Am Kreuzungspunkt zwischen Rauchfaden und Lichtstrahl entstehen Helligkeitsschwankungen, die mit einem Elektronen-Viervielfach messen werden können. Wenn man den Lichtstrahl durch einen beweglichen Spiegel der Rauchfadenbewegung nachführt, kann man die Helligkeitsschwankungen am Kreuzungspunkt zu Null machen. Die Spiegelbewegung ist dann ein Maß für die Rauchfadenbewegung. Die erreichte Meßgenauigkeit liegt bei $1 \div 3 \mu m$.

ERGEBNISSE


LITERATUR

(3) BECHERT & PFIZENMAIER (1973) DLR-FL 73-93

Bild 1: Prinzip der optischen Kompensation

Bild 2: Hüllkurve der Grenzschichtbewegung

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INTRODUCTION

Noise of axisymmetric supersonic jets has been measured. Especially the influence of an outer subsonic jet with combustion to the noise production of an inner supersonic jet has been studied.

EXPERIMENT

The inner supersonic jet has a flow Mach-number of up to $M = 3$ and the gas at rest has room temperature. This inner jet is influenced by an outer subsonic jet consisting of mixed reacting gases (e.g. propan and oxygen). The noise measurements were carried out in air at atmospheric conditions at a radial distance of some hundred jet diameters. A typical example of these measurements is demonstrated in figures 1 and 2. Curve a in Fig. 1 shows the directivity pattern of the noise radiation of the supersonic inner jet measured in the plane of the jet axis. Curve b in Fig. 1 is taken with the same flow in the inner jet and in addition with subsonic flow and combustion in the outer jet. Fig. 2 shows third octave band frequency spectra of the same experiment taken under an angle of $30^\circ$ to the jet axis. As can be seen the addition of the outer subsonic jet leads to considerable changes in the noise production and to a reduction in the overall noise though the mass flow is increased.

Fig. 1 Angular distribution of overall sound pressure levels in dB(A) ($0^\circ$ corresponds to flow direction)

Fig. 2 Third octave band frequency spectra
INVESTIGATION OF SCREECHING JETS WITH THE LASER SCHLIEREN CROSSED BEAM METHOD

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INTRODUCTION
The generation of screech noise by choked jets is investigated acoustically and through optical measurements of fluctuations within the jet.

EXPERIMENTAL
Cold jets from a circular convergent nozzle were investigated at different pressure ratios of total pressure $p_1$ to ambient pressure $p_0$. Acoustical far field measurements include directivity patterns and frequency spectra. Fluctuations of the axial density gradient within the shear layer were measured with the Laser Schlieren Crossed Beam method (1, 2).

RESULTS
Far field spectra measured under different angles $\varphi$ from the jet axis show typical broadband jet noise for the sonic jet and screech tones for the higher pressure ratios, see Fig. 1. The correlations measured by the Crossed Beam system indicate random fluctuations of the density gradient in the case of the sonic jet; but for the choked jets strongly periodic fluctuations are observed, see Fig. 2, the frequencies corresponding to those of the screech tones as given in Fig. 1. The amplitude of the oscillations depends on the axial location $x/d$, see Fig. 3. A comparison with shadowgraphs showed that the maxima occur at the ends of the shock cells.

REFERENCES

Fig. 1

Fig. 2

Fig. 3
INTRODUCTION

Acoustical radiation characteristics, mean velocity field and fluctuating wall pressure field have been investigated experimentally for a subsonic air jet impinging on a simulated wing-flap configuration. The objective of the investigation is to identify the potential source mechanisms in the blown-flap noise generation.

EXPERIMENTAL

The subsonic air jet used was generated from a 2" diameter nozzle. The span of the wing-flap was 24" with a 45° flap angle. The flap chord was 7.5". The nozzle was placed 4" below the wing-flap and 11" from the flap surface. The geometrical impingement point was thus just downstream of the potential core of the free jet. The acoustical characteristics were obtained by conventional methods. The mean velocity was measured both in the jet flow and in the wall flow with a pitot-static tube. Fluctuating wall pressure spectrum and two point space-time cross-correlation of the fluctuating wall pressure were obtained by using flush mounted miniature condenser microphones.

RESULTS

The total acoustic power radiated was found to vary with the 5th power of the jet velocity. The acoustic power spectra were broadband and peaked at a Strouhal No. $fD/U_\text{j}=0.1$. The frequency dependence of the power spectrum were $f^2$ and $f^{-3}$ for low frequencies and high frequencies respectively. High level rms wall pressure, $Prms$, were found in two regions, both centered along the midspan; $Prms/q_0=11\%$ upstream of the impingement point, $Prms/q_0=7\%$ near the trailing edge of the flap, where $q_0$ is the dynamic pressure of the impinging jet. Space-time cross-correlations of the wall pressure indicated that the "wall pressure spot", while being convected downstream, was stretched by the spanwise spreading of the mean flow. Near the impingement point, the pressure spot was oriented along the chord. Close to the trailing edge the pressure spot was oriented parallel to the edge. The fluctuating wall pressure spectra were found to be similar within the correlation area of the fluctuating wall pressure spot. Qualitative deductions are made from the experimental findings regarding the physical nature of the source mechanisms in blown-flap noise due to jet impingement.
INTERACTION BETWEEN VISCOUS WAVES AND CLOSE-TO-THE-WALL-TURBULENCE IN NEWTONIAN AND NON-NEWTONIAN FLUIDS

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INTRODUCTION

In turbulent air-flow through pipes the sound attenuation is rapidly increased above a definite flow velocity (1/2). This increase is assumed to be caused by an interaction between the viscous wave -originated at the wall by the sound wave- and the turbulence very close to the wall.

EXPERIMENTAL

Measurements of such an interaction in turbulent flow of liquids had been done using a pipe which is built as a longitudinally vibrating resonator. When filled with liquid -flowing or at rest- the oscillations of the resonator are damped by losses due to viscosity. Thus, the oscillating wall shear stress $\tau_{os}$ can be calculated from the resonator halfwidth. In Fig.1 $\tau_{os}$ -normalized by the wall shear stress for the medium at rest- is plotted as a function of the nondimensional quantity $d^+_a=d_ao/u_T$ which compares two typical length scales: a) the thickness of the acoustic boundary layer $d_a=(2\nu/\omega)^{1/2}$ (\nu kinematic viscosity, \omega angular frequency) and b) a length $v/u_T$ of the turbulent flow close to the wall ($u_T$ shear stress velocity).

DISCUSSION

Resonator data for water -- and sound attenuation measurements (2) for air - coincide well. Thus, the acoustic-turbulent interaction below $d_a=7.5$ is a viscous effect. A minimum can be explained by a crude model of a viscous wave being reflected at a turbulent layer close to the wall --, the mixing-length theory fails ... For Newtonian fluids (water, air) the interaction starts at $d_a=7.5$ but -caused by a thickened sublayer- at $d_a=11.5$ for a 20 ppm aqueous solution of the drag-reducing Separan AP 30 - +-. 

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A PRECISE MEASUREMENT OF THE SOUND ATTENUATION IN AIR FLOWING THROUGH RIGID PIPES AND ITS APPLICATION IN TURBULENCE RESEARCH

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The sound attenuation in pipes containing turbulent air flow increases with increasing flow velocity (1). A precise measurement of this attenuation is desirable in order to reveal the responsible damping mechanism.

PRINCIPLE OF THE MEASUREMENT AND RESULTS

Two loudspeakers at both the ends of the pipe alternatively radiate a sinusoidal sound wave into the pipe. At ten positions along the pipe the magnitudes and phases of the resulting sound pressures are measured. By a regression analysis taking into account the reflections from the ends of the pipe and the loss of acoustic energy in the microphones the very small sound attenuation in a smooth and rigid pipe can be evaluated. For air at rest the errors are smaller than 2%.

Fig. 1 shows the sound attenuation in a smooth and rigid pipe with superimposed turbulent flow averaged for both the directions of propagation (•). The attenuation is normalized by the theoretical value for air at rest. The solid line corresponds to the increase of the attenuation due to the drag-induced static pressure gradient and to the convection of the sound field by the flow disregarding the turbulence. Therefore, the deviation of the measured points from the solid line should be caused by some interaction between sound and turbulence.

In fact, it turns out that this deviation can be expressed in terms of the thickness ratio of the acoustical viscous boundary layer and the viscous sublayer of the turbulent flow. Ahrens (2) has measured the oscillating component of the wall shear stress in a longitudinally vibrating pipe containing turbulent water flow. The open points (○) in Fig. 1 are calculated from his data. They agree very well with the solid points. Thus, an interaction between the acoustical viscous boundary layer and the turbulence near the wall is measured by the sound attenuation.

REFERENCES

(1) C. Ahrens et al., Acustica (1971) 25 150, (2) C. Ahrens, Dr. Thesis, Universität Göttingen (1973)
RADIATED ACOUSTIC FIELD OF A CIRCULAR JET IMPINGING UPON A CIRCULAR PLATE

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INTRODUCTION

Diffraction of sound by rigid objects and the sound field by the flow on a rigid surface (boundary layer noise) have been investigated. The introduction of the finite, rigid surfaces in a flow field influences the noise generation and propagation characteristics in three ways: (1) the reflection and diffraction of the sound by the solid boundaries, (2) the sound field generated by the turbulent flow on the solid surface, and (3) the sound generated when the momentum flux leaves the edges. Presented here are some experimental results of the radiated sound field of a subsonic jet impinging on a rigid plate.

EXPERIMENTAL RESULTS

The model consists of the air jet from a nozzle of 5.1 cm diameter impinging at the center and normal to a rigid, circular plate of 76.2 cm diameter as shown in the Figure. The noise was measured at 3.05m radius from the center of the nozzle exhaust, and at various angles from the inlet. The impingement distance and the jet Mach number were varied. The Figure shows the typical directivity pattern of the OASPL and one-third octave SPL with center frequencies of 0.5 and 1.25 KHZ, which were taken for a jet Mach number of 0.78 and the impingement distance of 28.0 cm. The various postulated noise sources are illustrated in the Figure.

CONCLUSIONS

These preliminary results indicate that the high-frequency noise (1.25 KHZ) and the OASPL below the plate are about 6 dB higher than that in the direction above the plate. Assuming that the wall jet noise is negligible, it is conjectured that the high-frequency noise is generated predominantly by the mixing and impingement of the jet. Most of the acoustic energy from these sources is propagated below the plate due to reflection by the plate. However, due to diffraction and refraction, a small amount of this energy is propagated in the direction above the plate. It is evident that the low-frequency noise (0.5 KHZ) is about the same order of magnitude above and below the plate. Thus, it appears that this acoustic energy is generated by the flow near the edge, and that the propagation is affected by the refraction through the flow and diffraction by the edge.
TURBULENCE ET SON PUR D'UN VENTILATEUR AXIAL

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Un ventilateur axial est placé dans un conduit cylindrique portant deux rangées de trous disposés en anneaux à une distance de l'ordre d'un quart de longueur d'onde. Un anneau de trous est au niveau du rotor et le second en amont. Ces trous sont reliés par un conduit (figure 1). Cette configuration permet d'introduire un débit d'air périphérique au niveau du rotor sans nuire aux qualités aérodynamiques du ventilateur.

On analyse l'effet de ces débits périphériques sur le niveau du son pur (fortement émergent par rapport au bruit dû à la turbulence de l'écoulement), à la fréquence liée à la vitesse de rotation et au nombre de pales du ventilateur, en procédant à l'analyse spectrale des fluctuations des vitesses $v'$ et de pressions statiques $p'$ dans l'écoulement, au niveau du rotor et en aval. Les figures 2 (a, b) montrent les spectres en (1) et (2) sans addition de débits périphériques. Les figures 2 (c, d) montrent l'évolution de ces spectres aprè addition de débits périphériques. Le niveau de la densité spectrale de puissance de la fluctuation de pression statique à la fréquence du son pur diminue de 6 à 7 dB au niveau du rotor et de plus de 20 dB dans l'écoulement en aval; on observe, pour la même fréquence, une faible variation des niveaux de fluctuations des vitesses. La mesure des corrélations entre les paramètres $v'$ et $p'$ a mis en évidence, en outre, la propagation de la composante $p'$ à la vitesse du son et de la composante $v'$ à la vitesse débitante du ventilateur.

Ces résultats montrent la présence d'un mécanisme d'interaction entre l'injection au niveau du rotor des débits périphériques aspirés en amont et le bruit rayonné à la fréquence du son pur du ventilateur qui pourrait procéder de l'absorption acoustique active.

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INVESTIGATIONS OF THE ACOUSTIC IMPEDANCE OF ORIFICES
AND PERFORATED PLATES WITH TURBULENT GRAZING FLOW

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INTRODUCTION

The acoustic impedance of an orifice changes in the presence of grazing flow. In the case of laminar flow it was found that for small flow velocities the impedance curve passes through a spiral in the complex plane [1]. For higher flow velocities the reactive part of the impedance decreases, whereas the resistive part increases linearly with the flow velocity.

EXPERIMENT

With turbulent grazing flow the measuring equipment described in [1,2] had to be improved. Now the sound pressure gradient within the cylindrical neck of the orifice is measured by four microphones located at various distances from the upper edge of the orifice. Thus it is possible to evaluate the impedance \( W \) in the plane of the upper edge of the orifice (for air at rest it equals the impedance \( W_0 \) of the so-called endcorrection).

RESULTS

The figure gives the normalized impedance \( W_n \) (\( W_n = W/W_0 \)) of an orifice with turbulent duct flow (diameter of the orifice 4 mm, frequency 315 Hz, flow velocity \( U \) in m/s). The characteristic shape of the impedance curve is the same as in the laminar case. But the slope of the real part \( \text{d}(\text{Re}(W_n))/\text{d}U \) is much smaller for turbulent than for laminar flow. Relations between the impedance and various parameters e.g. Strouhal number, ratio of boundary layer thickness to diameter etc. as shown in [1,2] will be checked for the case of turbulent flow.

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DRUCKMESSUNGEN AN DER ZUNGENKANTE EINES RADIAL-VENTILATORS

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EINLEITUNG

An frei gegen eine Kante laufenden Ventilatorrädern (1) und an denselben Rädern in einem Gehäuse (2) wurde früher der mittlere (rms) Schalldruck der Frequenzkomponenten des Drehklanges \( \tilde{p}_f \) als Funktion der Umfangsgeschwindigkeit \( U \), des Raddurchmessers \( D \), der betreffenden Frequenz \( f \) und des Messtorres \( x_1 \) im Fernfeld bei fester Schaufelzahl \( z \) und kinematischer Zähigkeit \( \nu \) untersucht. Für \( He \leq 1 \) wurden näherungsweise folgende Produktansätze bestätigt:

\[
\tilde{p}_f = \text{Re} \cdot \frac{\nu}{\nu_0} \cdot f \cdot \text{D} \cdot \text{e} \cdot \text{G} \cdot \text{He} \cdot x_1 \quad (1)
\]

\[
\text{Re} = \frac{U \cdot D}{\nu}, \quad \text{Ma} = \frac{U}{a_0}, \quad f = \frac{f \cdot D}{U}, \quad \text{He} = \text{Ma} \cdot f \cdot x_1 \cdot \frac{\nu}{D}
\]

THEORIE

Aus der Lighthill-Gleichung für ein Strömungsfeld im Volumen \( V \) mit Rand \( A \) ohne äußere Kräfte in isentropischer Näherung - z.B. (3), S. 28 - leitet man unter Vernachlässigung nichtlinearer Terme für kleine Machzahl \( Ma \) und große Reynoldszahl \( Re \) für den mittleren Druck \( \tilde{p}_f \) die folgende Integralgleichung ab:

\[
\tilde{p}_f (x', St, Ma, Re) = \frac{2}{\pi} \frac{2}{\tilde{p}_f} \int_A \frac{2}{\tau} \tilde{p}_f (x, St, Ma, Re) e^{2\pi i \cdot He \cdot \tau} \cdot dA
\]

(2)

Insbesondere gibt Gl. (2) einen Zusammenhang zwischen dem Schalldruck im Fernfeld und dem Druck auf dem Rand des Quellvolumens (Nahfeld). Ein Ansatz wie (1) für das Nahfeld überträgt sich unmittelbar auf das Fernfeld.

MESSUNGEN

Druckmessungen von Agnon an der Zungenkante eines Radialventilators zeigen, daß sich ein Produktansatz

\[
\tilde{p}_f (x', St, Ma) = \text{Ma} \cdot \text{F}_f (St) \cdot \text{F}_r (x', \text{He})
\]

näherungsweise verifizieren läßt. Im Unterschied zum Fernfeld scheint der Exponent für verschiedene Strouhalzahlen unterschiedlich zu sein.

Drehklangkomponenten im Fernfeld Druckkomponenten an der Zunge

LITERATUR


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BRUIT ÉMIS PAR UN PROFIL PLACé AU VOISINAGE D'UNE ZONE TURBULENTE

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Un profil NACA 6512 A 10° est placé dans le cône à potentiel d'un jet rectangulaire dont la vitesse moyenne est U = 20 m/s. Lorsque le profil est au centre du cône (γ = 0) le bruit émis est concentré autour de la fréquence de Strouhal soit f ≈ 1500 Hz dans l'essai rapporté. Mais lorsqu'on approche le profil de la frontière interne de la zone de mélange, sans toutefois la pénétrer (γ ≤ 60 mm), une autre fréquence pure apparaît, soit ici f ≈ 250 Hz.

On analyse ces phénomènes en mesurant les corrélations $R_{\tilde{p}\tilde{p}}$ entre signaux filtrés et en utilisant l'expression [1], [2]:

$$\frac{\tilde{p}^2}{\bar{p} a_o} = \frac{\nu}{2} \frac{\nu}{\rho a_o^2} \frac{x}{|x|} \int \int \left[ R_{\tilde{p}\tilde{p}} \right] \frac{dy}{4\pi}$$

La figure montre l'évolution de $\tilde{p}$, $\tilde{\alpha}$ et $R_{\tilde{p}\tilde{p}}$ avec la distance y à la frontière. En outre on observe une propagation vers le bord d'attaque, à la vitesse du son, de la composante $\tilde{p}$ à 1500 Hz, et une convection vers l'avant à une vitesse voisine de celle de l'écoulement de la composante $\tilde{p}$ à 250 Hz.

Ces phénomènes révèlent la présence d'un double mécanisme d'émission:
- l'un lié à la source de volume située au voisinage immédiat du bord de fuite et réagissant sur la surface solide.
- l'autre résultant de l'interaction avec le profil du champ aérodynamique (pression et vitesse) induit dans le cône à potentiel par la zone de mélange interne.

L'évaluation des intensités acoustiques [3] est en bon accord avec l'expérience, par ex., pour γ = 60:

$$\tilde{p}_{\text{exp}} (250 \text{ Hz}) = 43 \text{ dB}$$
$$\tilde{p}_{\text{cal}} (250 \text{ Hz}) = 43,5 \text{ dB}$$
$$\tilde{p}_{\text{exp}} (1500 \text{ Hz}) = 51 \text{ dB}$$
$$\tilde{p}_{\text{cal}} (1500 \text{ Hz}) = 53,5 \text{ dB}$$

REFERENCES

AXIAL FAN NOISE FROM INLET FLOW DISTORTIONS

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INTRODUCTION

Flow distortions ingested into a fan rotor disc can be scattered as both broadband and discrete frequency sound. The dipole noise centred at blade passing frequency and harmonics is related both theoretically and experimentally to the steady flow non-symmetry at inlet to a low tip speed ducted axial rotor.

EXPERIMENTAL

Inlet velocity perturbations were measured using hot wire anemometers rotating with the fan blade row. Cross and single wired probes established the magnitude and phase of the velocity distribution over the rotor disc. Naturally induced transient distortions generally produced weak and variable tone noise whilst impulsive distortions typical of strut wake flows generated strong signals at the rotor blade harmonics. Most naturally induced distortions radiate sound on the fan axis. Severe cross flows at the fan intake generated significant broadband noise. The influence of rotor blade steady loading on the noise generation was assessed by varying the fan speed and aerodynamic duty.

THEORETICAL

Rotor noise theory and an experimentally derived unsteady aerodynamic lift function (1) were used to calculate the noise radiation for a given flow distortion. The theory points out the importance of coupling between the rotor circumferential steady loading distribution and harmonics of the spatial inlet flow distortion. Such considerations indicate the high noise potential of impulsive type velocity profiles (having many harmonics) as seen in the experiments. Fans of the same duty but with widely differing rotor blade numbers, may respond identically at any one harmonic for impulsive flow distortions. Adverse noise levels could occur with low bladed fans that couple favourably with low order distortion harmonics typical of inlet cross flow induced velocity profiles.

CONCLUSIONS

The theoretical and experimental comparisons are generally favourable although noise directivity plots show discrepancies. The results are particularly interesting for aero propulsion and lift fan applications.

REFERENCES

FINITE STRIP ANALYSIS OF CURVED MULTI-SPAN STRUCTURES

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INTRODUCTION

A finite strip method is developed for analysing the free vibration characteristics of singly curved plates, stiffened by thin-walled open section beams in the longitudinal direction.

THEORETICAL ANALYSIS

The finite strip method (1) was applied by dividing the curved plate into a number of rectangular strips which extend the complete width of the plate in the straight direction. The motion in this direction was assumed to take the same form as the modes of vibration of a uniform straight beam. In the curved direction the motion was represented by a finite element shape function for a curved beam. The motion of the stiffeners was taken to be compatible with the plate.

RESULTS

The accuracy of the method is first demonstrated by considering a singly curved rectangular panel without stiffeners. The results for a five bay stiffened structure are presented next and are shown to agree closely with a transfer matrix solution (2). Finally, a single row of panels between two adjacent frames of a 5 x 15 orthogonally stiffened panel array is analysed. The results are compared with experimental measurements (3). Nine modes show good agreement.

REFERENCES

DETERMINATION OF DYNAMIC VISCOUS DAMPING COEFFICIENT
OF SOME ELASTIC REINFORCED CONCRETE STRUCTURES

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INTRODUCTION:
The prediction of reinforced concrete structures behaviour in vibrations entail a complete knowledge of the damping capabilities of this material. For the use of its computer programs ELECTRICITE DE FRANCE have different investigations in progress to obtain an evaluation of viscous damping coefficient of concrete beams in different structures.

THE STATIC RELEASED DEFLECTION METHOD:
This method was applied to concrete chimneys (150 meters high) in two power plants with different soil foundations. The static deflection was created by a cable stretched at the top of two chimneys. We used a mechanical of explosive cutter. The transient vibration collected by seismic accelerometers is processed on an H.P. Fourier Analyser with numerical filtering to obtain for each three first modes the logarithmic decrement and the damping coefficient.

THE HARMONIC FORCED VIBRATION METHOD:
This method is carried out in laboratory on several beams vibrating on the first free-free flexural mode. The research of the resonance frequency under harmonic force applied in one point give us different capabilities to determine the damping coefficient:
1. Measure of the dynamical amplification coefficient (Q factor) by the phase method using the admittance diagram (velocity/force) after mass cancellation of the exciter head.

2. Comparison between experimental and computed mode shapes for different values of the damping coefficient using the mathematical model of the straight beam in forced harmonic vibrations.

3. Identification of the equivalent one degree of freedom mechanical system using the experimental driving point or cross transfer function by least square reduction:

These methods are applied to different beams with the same steel/concrete ratio (2%) and we analyse the effect of geometry (3 different scales), of granulometry (2 concrete qualities), the effect of preliminary static deformation of the beams (concrete with and without fissures), the effect of the superposition of static and dynamic loading.
APPLICATIONS OF CURVED FINITE ELEMENTS TO SANDWICH BEAMS USING SHEAR STRAIN FORMULATION

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INTRODUCTION

The free vibration of curved sandwich beams was investigated previously by the author (1, 2). The object of this work is to assess the probable success of using a modified Wang theory. Therefore the main practical advantages of the present formulation may be summarised as:
(a) transverse, in-plane, and rotary inertia are accounted for in the kinetic energy expression
(b) higher order derivative of the transverse displacement w is included in the strain energy expression
(c) in addition to the above when \( q_e = 0 \) and only one face of the sandwich is considered, the analysis may be used for studying homogeneous beams.

THEORETICAL ANALYSIS

Based on the assumptions given in (3), the strain and kinetic energy expressions were derived as:

\[
U = \frac{1}{2} \int_0^L \left\{ A \left( \frac{\partial^2 w}{\partial y^2} + \frac{w}{R} \right)^2 \right\} \partial y + D \left( \frac{\partial^2 w}{\partial y^2} + \frac{1}{R} \frac{\partial w}{\partial y} - \frac{2\partial w}{\partial y} \right)^2 \right\} \partial y \tag{1}
\]

\[
T = \frac{1}{2} \omega^2 \int_0^L \left\{ M \left( \nu^2 + \omega^2 \right) + I \left( \frac{e}{c} + \frac{\nu}{K} - \frac{\partial \omega}{\partial y} \right)^2 \right\} \partial y \tag{2}
\]

where:

\[
A = 2E h_1; \quad D = 2E h_2 h_3^2; \quad B = 2h_1 e_c;
\]

\[
M = 2h_1 \rho_1 + 2h_2 \rho_2; \quad I = \frac{2}{3} h_1 \rho_1 + 2h_2 h_3 \rho_2
\]

using the potential energy functional the appropriate boundary conditions were derived. The free vibration of sandwich beams were investigated using the finite element displacement method incorporating elements having four, six, seven and eight degrees of freedom per node. A simple check on these models were made to determine the rigid body node representations. The convergence of the finite element solutions were obtained to determine the free vibration of curved sandwich beam clamped at both ends. It was found that the results of the present formulations converge very rapidly to those given in (1, 2) when using only few elements.

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EIGHTH INTERNATIONAL CONGRESS
ON ACOUSTICS. LONDON 1974

FREE VIBRATION OF NON-UNIFORM SPHERICAL SHELLS
SUBJECTED TO VARIOUS BOUNDARY CONDITIONS

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INTRODUCTION

A theory is presented for the determination of the free vibration characteristics of non-uniform, thin spherical shells. The formulation is based on a recently developed method for the analysis of cylindrical shells (1 & 2). It is a hybrid of the finite-element and classical shell theories. The "finite element" chosen is a spherical frustum and the displacement functions are derived from the exact solution of the shell equations for a thin sphere.

THEORETICAL ANALYSIS

The finite element is defined by two nodes, i and j, and nodal surface boundaries; the displacement functions are defined by

\[ [U(\varphi, \theta), W(\varphi, \theta), V(\varphi, \theta)]^T = [N][\delta_i, \delta_j]^T, \]

where \(U, V,\) and \(W\) are the axial, circumferential, and radial displacements, respectively, and \(\{\delta_i\}, \{\delta_j\}\) are nodal displacements at \(i\) and \(j\) associated with the \(n^{th}\) circumferential wave number.

The strain and stress vectors are written as

\[ \{\varepsilon\} = [B]\begin{pmatrix} \delta_i \\ \delta_j \end{pmatrix}, \quad \{\sigma\} = [P]\{\varepsilon\}, \]

where \([P]\), the elasticity matrix, characterizes the shell's anisotropy. To determine the elements of \([N]\) and \([B]\), Reissner's shell equations (3) are solved and the displacement functions of thin spherical shell are explicitly derived in terms of Legendre functions and constants \(C_i\) and \(D_i\) (\(i = 1, 2, 3, \) and \(4\)). Finally, the \(C's\) and \(D's\) are obtained in terms of the nodal displacements \(\{\delta_i\}\) and \(\{\delta_j\}\).

With \([N]\) and \([B]\) determined functions of \(n\theta, \varphi\) and the elements of \([P]\), the general terms of the stiffness and mass matrices for one finite element were analytically obtained after lengthy manipulations.

Given the dimensions and properties of the shell, a computer program assembles the global mass and stiffness matrices and calculates the eigenvalues and eigenvectors of axially non-uniform spherical shells for various boundary conditions. The frequencies of vibrations are compared with those obtained by other theories and with the experiments of others.

REFERENCES


* This work was carried out at Marshall Space Flight Center, NASA, Alabama, and sponsored by the National Research Council of America. The author wishes to express his thanks to Dr. John R. Admire for helpful discussions during the course of the investigation.
EIGHTH INTERNATIONAL CONGRESS ON ACOUSTICS, LONDON 1974

FREQUENCY COEFFICIENTS FOR RECTANGULAR PLATES WITH SYMMETRICAL SLOPE RESTRAINTS CARRYING CONCENTRATED MASSES

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INTRODUCTION

Small amplitude vibrations of thin, elastic plates subjected to in-plane normal forces $N_x$, $N_y$ and carrying a concentrated mass $M$ are described by the partial differential equation:

$$D M^2 \frac{\partial^4 w}{\partial x^2 \partial y^2} - N_x \frac{\partial^2 w}{\partial x^2} - N_y \frac{\partial^2 w}{\partial y^2} = -\rho h \frac{\partial^2 w}{\partial t^2} - M \frac{\partial^2 w}{\partial t^2} \delta(x - x_1) \delta(y - y_1)$$

(1)

A limited amount of information is available on the solution of special cases of this equation in the case of symmetrical slope restraints (1).

SOLUTION OF THE PROBLEM

Consider the following boundary conditions (Fig.1):

$$w = 0 \text{ on the boundary (2a)}$$

$$\frac{\partial w}{\partial x} \bigg|_{x = \pm \frac{a}{2}} = \rho \frac{\partial^2 \tilde{w}}{\partial x^2} \bigg|_{x = \pm \frac{a}{2}}$$

$$\frac{\partial w}{\partial y} \bigg|_{y = \pm \frac{b}{2}} = \rho \frac{\partial^2 \tilde{w}}{\partial y^2} \bigg|_{y = \pm \frac{b}{2}}$$

(2b)

For the first mode of vibration one can take the approximate expression:

$$w(x,y,t) \sim (\alpha x^4 + \beta x^2 + 1)(\alpha_1 y^4 + \beta_1 y^2 + 1)e^{i\omega t}$$

(3)

where the coefficients of the polynomials are evaluated using equations (2). Using Galerkin's method one obtains an expression for the fundamental frequency coefficient $\Omega_1$. A similar approach is followed for other modes of vibration.

REFERENCES

INTRODUCTION

A new approximate method is presented for the vibration analysis of thin, rectangular, edge-stiffened isotropic plates. The presentation of the edge-stiffened plate's frequency in the form analogous to that of a simply supported plate is postulated.

THEORETICAL ANALYSIS

The equation for the plate's free vibration reads

\[ D \left( \frac{\partial^4 w}{\partial x^4} + 2 \frac{\partial^4 w}{\partial x^2 \partial y^2} + \frac{\partial^4 w}{\partial y^4} \right) + \rho \frac{\partial^2 w}{\partial t^2} = 0 \] (1)

The boundary conditions for an edge-stiffened plate are

\[ l_{x,\alpha} \left( \frac{\partial}{\partial x}, \frac{\partial}{\partial y}, \frac{\partial}{\partial t} \right) w = 0 \quad \text{at} \quad x = 0, \ x = a, \]

\[ l_{y,\beta} \left( \frac{\partial}{\partial x}, \frac{\partial}{\partial y}, \frac{\partial}{\partial t} \right) w = 0 \quad \text{at} \quad y = 0, \ y = b, \] (2)

where the operators \( l_{x,\alpha} \) and \( l_{y,\beta} \) are given in (1). Following the author's earlier work (2), the relationship between the natural frequency and the unknown analogues of the wave numbers \( k_1 \) and \( k_2 \) is postulated below

\[ \omega = \left( \frac{D}{\rho} \right)^{1/2} (k_1^2 + k_2^2). \] (3)

The wave numbers are obtained from a pair of simultaneous transcendental equations, such, that products of the types

\[ w(x,y,t) = X(x) \sin k_2 y \exp (i\omega t) \]

\[ w(x,y,t) = Y(y) \sin k_1 y \exp (i\omega t) \] (4)

satisfy the initial equation (1) - subject to postulate (3) - and also, boundary conditions for the edge pairs

\[ l_{x,\alpha} \left( \frac{d}{dx}, k_2, i\omega \right) X = 0 \quad \text{at the edges} \quad x = 0, \ x = a \]

and

\[ l_{y,\beta} \left( k_1 \frac{d}{dy}, i\omega \right) Y = 0 \quad \text{at the edges} \quad y = 0, \ y = b \]

respectively.

REFERENCES

HIGH-FREQUENCY FLEXURAL VIBRATION OF THICK RECTANGULAR BARS AND PLATES

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INTRODUCTION

An extension of Bolotin's (1) solution fitting technique makes use of travelling wave superposition to represent behaviour in the middle region of the bar or plate and solution fitting to determine the wave-vectors which meet the, possibly mixed, boundary conditions. Middle region behaviour has been discussed in previous papers (2,3). Unlike most alternatives this method gives solutions for which the error decreases as the natural frequency increases.

COMPUTATIONAL PROCEDURE

Firstly the allowable wavevectors are computed by noting that, except for the very corners, normal displacement z given by

\[ z = B \exp(-k_2 x_2) \sin(k_1 x_1 + \theta_1) \sin(k_2 x_2 + \theta_2) \]  

is a solution of the wave equation, for which the second term is exact and the first term is an approximation which satisfies Lagrange's simplified equation for the plate, or with \( k_2 = 0 \), Euler's equation for the bar.

In equation 1 \( k \) is given by \( (k_1^2 + k_2^2) \).

Equation 1 together with its companion for conditions along the \( x_2 \) coordinate and the prescribed boundary conditions provide a pair of transcendental equations for the allowable values of \( k_1 \) and \( k_2 \) for the plate or of \( k \) for the bar.

The associated natural frequencies are then computed from

\[ \omega_1 = c(4k) \quad (\text{for the bar}) \quad \text{or} \quad c(4k_1^2 + 4k_2^2)^{1/2} \quad (\text{for the plate}) \]  

in which \( c \) is the flexure wave velocity obtained either by experiment, from Fig. 3 of (2) or by solution of Timoshenko's equation as described therein.

A short computer programme has been written which proceeds from the solution of the transcendental equations to the solution of the Timoshenko equation and provides a printout of the frequencies either as a mode array or in ascending order of natural frequency together with associated modes.

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(2) H.M. Nelson, J. Sound and Vib. (1971) 18 93;
ERRORS IN SHELL VIBRATION THEORIES

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THE THEORIES

The authors (G. M. L. Gladwell and D. Vijay, to be published) have developed a finite element method for the vibration of axisymmetric bodies. The element is a triangle rotated about the symmetry axis, is based on the element of R. W. Clough and C. A. Felippa (1) involves cubic variations within the element, and has high accuracy. This method is used to investigate the accuracy of three simpler finite element methods. The comparisons are made by applying each to the axisymmetric free vibrations of a uniform solid or annular cylinder with free ends.

Thin shell elements generally satisfy Kirchhoff's hypotheses and allow no normal or shear strain $\varepsilon_n = 0 = \varepsilon_n$. Typical thick shell elements assume $\varepsilon_n = 0$, but allow shear strain. The linear shell element assumes linear variation of displacements across the shell thickness.

RESULTS

Fig. 1 shows the 1% error curves for the natural frequency corresponding to the first antisymmetric mode. The symbols $L$, $a$, $H$ denote the length, mean radius and thickness of the cylinders. Arrows indicate directions of increasing error. Each region is labelled by the simplest theory that yields a frequency error of less than 1% when compared with the cubic theory. It is clear that, e.g. thin shell theory is applicable in a range that is quite inadequately described by the usual $H/a < 0.05$. It is not applicable to short cylinders, however thin, and it is appropriate for some long cylinders which are not thin.

Similar curves have been obtained for the higher frequencies.

REFERENCE


Fig. 1 1% error curves for the first mode.
EXPERIMENTAL AND THEORETICAL STUDIES OF THE VIBRATIONS OF DISKS, ANNULAE AND SIMILAR STRUCTURES

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INTRODUCTION

Until comparatively recently the contour extensional (in-plane) resonant modes of vibration in thin disks, have been of secondary interest compared with the flexural modes. Love 1944 produced the frequency equations, which were solved by Holland in 1966.

Using the Bell 1968 method to measure the resonant frequencies, gives excellent agreement between experimental and theoretical values.

EXPERIMENTAL

The contour-extensional modes subdivide into three groups. By observing two modes Poisson's Ratio can be determined. Tests were carried out on a variety of materials, against temperature. A typical curve for aluminium is shown in Fig. 1.

THEORETICAL ANALYSIS

The analysis is extended to cover the case of a disk with a central hole. Values of the normalised dilatational wavenumber have been tabulated for poisson's ratio and β from 0.00 to 1.00, both in steps of 0.02

\[ \beta = \frac{\text{Inside diameter}}{\text{Outside diameter}} \]

Existing theory caters for the case of thick rings, but errors soon become large as β reduces. Fig. 2 shows curves calculated from the theory and points obtained from experiment.

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PARTIAL DIAGONALIZATION OF SECULAR DETERMINANTS FOR LARGE PERIODIC SYSTEMS BY APPLICATION OF EXTRA-FORCE METHOD

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INTRODUCTION

Method of Lagrange's equations with extra forces has been shown to be useful in the normal-mode analysis of vibrating systems (1,2) and for lattice dynamical problems in solid state physics (3). Application of the method to more complicated periodic systems shows that a considerable amount of simplification in analysis can be achieved.

The method takes advantage of the symmetry properties of the vibrating systems and eliminates the need for artificial boundary conditions that do not yield satisfactory solutions.

METHOD OF EXTRA FORCES

For a frictionless vibrating system having N degrees of freedom, characterized by a set of generalized coordinates \( u_i \) (i=1,2,3,...,N), Lagrange's equations of motion are \( \Lambda L(u,\dot{u})=0 \), where \( \Lambda L=(d/dt)(\partial L/\partial \dot{u})-\partial L/\partial u \), and the Lagrangian for the system is \( L(u,\dot{u})=T(u,\dot{u})-V(u) \).

If the system under consideration consists of many elements in a periodic array, it is advantageous to regard such a system as a coupled or a modified set of basic systems and split the Lagrangian into two parts: \( L(u,\dot{u})=L_0(u,\dot{u})+L_1(u,\dot{u}) \), where \( L_0(u,\dot{u}) \) is the Lagrangian for the basic systems and \( L_1(u,\dot{u}) \) is that associated with the modification of the basic systems. The equations of motion may now be put in the form \( \Lambda L_0(u,\dot{u})=F_0=-\Lambda L_1(u,\dot{u}) \), where \( F_1 \) can be regarded as an extra force associated with the coordinate \( u_i \) that modifies the basic equations of motion whose solution is expressed in terms of a set of normal coordinates \( q_r \) as \( u_i=\sum_{r=0}^{N-1} a_{ir} q_r \). The essential step in the application of the extra force method is in transforming the Lagrangian \( L(u,\dot{u}) \) into \( L(q,\dot{q}) \) so that the equations of motion now become \( \Lambda L_0(q,\dot{q})=F'_0 \), where \( F'_0=\sum_{i=1}^{N} \left( \partial u_i / \partial q_r \right) F'_i \).

The solution of the transformed equations of motion yields the normal modes which are the results of coupling various normal modes of the basic systems according to the extra force \( F'_0 \). Simplification in analysis is achieved by the fact that the coupling of the basic normal modes permitted by \( F'_0 \) is restricted by the symmetry properties of the basic systems. Furthermore, the \( 1/\sqrt{N} \) dependence of the coefficients \( a_{ir} \) often leads to a reasonable approximation in systems containing many particles.

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STATISTICAL ENERGY ANALYSIS APPLIED TO A BEAM-PLATE SYSTEM

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In building acoustics the coupling between sound fields in plates and beams is of interest, where walls of framework are present. In order to study how SEA applies to beam-plate systems, a simplified system (fig.1) was considered. SEA (5,6) yields, when (1) or (2) is excited, energy-ratios:

\[
\frac{E_2}{E_1} = \frac{n_{12}}{n_{21} + n_2} \text{ or } \frac{E_1}{E_2} = \frac{n_{21}}{n_{12} n_2} \text{ resp.} \quad (3a,b)
\]

Among other difficulties in applying eqn. (3) is how to determine the coupling loss factors \(n_{12}\) and \(n_{21}\).

CALCULATION OF COUPLING LOSS FACTORS

a) \(n_{12}\): With the assumption that the beam (1) is not loaded by the plate (2) the displacement \(d_1(x,t)\) and the energy \(E_1\) of the beam, when exposed to an external point force, are first derived. Secondly the displacement \(d_2(x,y,t)\) and the energy \(E_2\) of the plate, when forced to follow the above displacement of the beam, are derived. \(n_{12}\) is defined by the power flow from (1) to (2): \(E_1 \omega n_{12}\), which on the other hand is equal to the power lost by dissipation in (2): \(\frac{E_2 - \omega n_2}{E_1 \omega n_{12}}\). So \(n_{12}\) is calculated from eqn. (4a), using statistical considerations (7) on the mode-distribution.

\[
\frac{n_{12}}{n_2} = \frac{E_2}{E_1} + 1, \quad \frac{n_{21}}{E_2} = \frac{E_1}{E_2} - 1
\]

b) \(n_{21}\): In analogy \(n_{21}\) is calculated from eqn. (4b), where \(E_2\) is derived with the assumption that the plate-beam support line is at rest.

EXPERIMENTAL RESULTS

Fig.1 Experimental set-up for determ. of acc. level diff.

- mean values from 10 plate lengths (0.5m < b < 2.6m)

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*) The basic work was performed at The Acoustics Laboratory Technical University of Denmark
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EFFECTS OF LATERAL INERTIA ON CRITICAL SPEEDS OF A MULTI-MASS ROTATING SHAFT

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INTRODUCTION

The variation of critical speeds of forward precession of a multi-mass rotating shaft depending on the value of lateral inertia of turbine and compressor is investigated.

In a previous paper (1) we have already discussed the effects of lateral inertia on the first critical speed.

THEORETICAL ANALYSIS

The problem formulation for critical velocities of a rotating shaft posed in discrete coordinates leads to the solution of the eigenvalue problem

\[(FM - \lambda I)v = 0\] (1)

where F is the flexibility matrix, M the mass matrix, I the identity matrix and \(\lambda = \frac{1}{\omega^2}\) (2).

In the present paper it is demonstrated that equation (1) gives n real and m imaginary values of \(\omega\) when the lateral inertias are taken into account.

The importance of taking into account the lateral inertia of turbine and compressor in computing the critical speeds is then shown with an example (fig.1). Because has already been shown that several classical methods of finding the first critical speed may fail when lateral inertias are taken in account (1, B. Atzori, to be published), the behaviour of several matrical methods for solving the eigenvalue problem is analyzed and the results compared.

REFERENCES

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Beim Rollen eines Rades auf einer Schiene treten fast immer Beschleunigungen auf, die wesentlich höher sind als die Erdbeschleunigung. Es ist also damit zu rechnen, daß das Rad für kurze Zeitintervalle "abhebt" und dann die Schiene impulsartig anregt. Betrachtet man als stark vereinfachtes Modell ein ideal glattes Rad auf einer Schiene der in der Skizze dargestellten Form, so läßt sich zeigen, daß schon bei sehr kleinen Geschwindigkeiten und extrem niedrigen Werten von h/l das Rad abhebt und etwa den durch die gestrichelte Linie angezeigten Verlauf nimmt.

Der Impuls, der bei jedem Stoß auf die Schiene auftritt, ist proportional der Radmasse und der Fortschrittsgeschwindigkeit $V_0$. Da außerdem die Anzahl der Stöße mit $V_0$ zunimmt, gilt

$$F^2 \sim v^2 \sim V_0^3,$$

wobei $F$ bzw. $v$ die pro Oktave bzw. Terz wirkende Kraft bzw. Körperschallschnelle sind. Bei Verdopplung der Geschwindigkeit $V_0$ erhöhen sich also die Körperschallpegel um 9 dB. Dieses Resultat steht im Einklang mit zahlreichen Messergebnissen.


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INTRODUCTION

This paper is concerned with a model which represents a severe requirement in most cable systems: consider a spring - rod - payload system moving axially with constant velocity \( V_0 \). The free end of the spring is suddenly brought to rest. Following a one dimensional wave approach one can show that the stresses are predicted by the expression:

\[
\sigma(x,t) = \frac{E}{c^2} \sum_{n=1}^{\infty} d_n \left( \cos \beta_n x - \frac{E A_s \beta_n}{k} \sin \beta_n x \right) \sin \beta_n t
\]

(1)

The expansion coefficients are given by the equation:

\[
d_n = \frac{V_0 \beta_n}{L/2} \left[ \frac{E A_s \beta_n}{k} + \frac{M c^2}{E A_s} \left( \frac{E A_s \beta_n}{k} \right)^2 + 1 \right]
\]

(2)

and the \( \beta_n \)'s are the roots of the transcendental equation:

\[
\beta_n \tan \beta_n L \left( \frac{M k c^2}{E A_s} + E A_s \right) + M c^2 \beta_n^2 = k
\]

(3)

where \( L \) : length of the rod, \( M \): payload mass, \( k \): spring constant, \( A_s \): cross sectional area of the rod, \( E \): Young's modulus and \( c \): speed of sound in the rod. The case where the spring is infinitely rigid has been previously discussed by the authors (Laura et al. to be published).

NUMERICAL RESULTS

Dynamic stresses can be calculated using the previous equations. An interesting feature of the problem is the fact that the displacement function is expanded in terms of non-orthogonal functions.

It is shown in this paper that an accurate, engineering solution can be obtained by means of a simulation approach. The continuous model is replaced by a discrete system of masses and springs and the temporal variable is varied in small time increments.
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MOVABLE EXPERIMENTAL FACILITIES FOR THE IDENTIFICATION
OF INDUSTRIAL STRUCTURES UNDER FORCED VIBRATIONS

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This laboratory vehicle was set out for the purpose of mechanical
identification of industrial structures under harmonic forced vibra-
tions. It was mainly designed for the types of structures encountered
in ELECTRICITE DE FRANCE power plants (turboalternators foundations;
turbines shafts; nuclear reactor pressure vessels; water, steam, gas
or liquid sodium piping systems; cooling towers; chimneys, ...).

EXPERIMENT'S PRINCIPLE
Vibratory forces can be imposed to the structure on four different
points with adaptation of the excitation to the investigated mode
shape. The deformation of the structure is picked up in many points
by fixed or movable transducers. The data are collected and analysed
in real time so as to modify the excitation according to different
adaptation criteria.

EXPERIMENT'S FACILITIES
1. Excitation: Several types of exciters can be selected according to
the dynamic mass of the structure. The dynamic forces come from
electrodynamic exciters (200 or 1500 Newtons peak amplitude) or hyd-
raulic ones (10000 Newtons peak amplitude). Four such exciters can
work together. Each one delivers its force between the structure and
a reaction inertial mass. Each force eventually imposed to the struc-
ture is measured by a force transducer inserted at the application
point and is amplitude and phase controlled monitored.

2. Data collection and analysis: 40 accelerometers are on the struc-
tures. Their signals, as those coming out of the force transducers,
are processed by analog multipliers changing the amplitude and phase
measurements into real and imaginary components towards the reference
signaig generator. The DC outputs of the multipliers are digitised and
processed by a digital computer working in real time. It displays the
vibratory results on listings expressed in physical units (accelera-
tions, velocities, displacements, mechanical impedance or admittance)
or in graphs (amplitude/frequency curve, Nyquist diagram, phase dis-
persion, mode shape). Besides, it ensures the storage of these results
on magnetic tape for further treatment.

LABORATORY VEHICLE
All these excitation, measurement and analysis facilities are gathered
in a steel container (10 metres long) which can be trailed on roads,
carried by train and even picked up by a crane. The container has
independent air and water-conditioning. It needs only but an electric-
ial mains supply. It can put up with measurement points located as
far as 100 metres and excitation emplacements up to 50 metres.

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The problem of the appropriation of the excitation forces in vibration
testing.
EIGHTH INTERNATIONAL CONGRESS ON ACOUSTICS, LONDON 1974

ANALYTICAL AND EXPERIMENTAL RESULTS ON A TOWER SHAPED STRUCTURE

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INTRODUCTION

Analytical and experimental results are reported on the dynamic behavior of a) a structural steel column-supported tank system and b) a reinforced concrete tower-tank structure (Fig. 1).

THEORETICAL ANALYSIS

It is assumed that both systems are described by the well known partial differential equation

\[ \frac{EI}{x^4} \frac{\partial^4 w}{\partial x^4} + \rho A \frac{\partial^2 w}{\partial t^2} = 0 \]  (1)

and appropriate boundary conditions. The following frequency equation is then obtained:

\[ \frac{1}{y} \frac{\cosh y}{\sinh y} = \frac{M}{M_b} \]  (2)

where \( M_b \) is the mass of the beam and

\[ f_i = \frac{1}{2\pi} \left( \frac{\sqrt{L}}{L} \right)^2 \sqrt{\frac{EI}{\rho A}} \]  (3)

EXPERIMENTAL RESULTS

The first two natural frequencies of both structures were measured and good agreement was obtained in general, between analytical and experimental results. The first value obtained (1):

\[ \delta = \ln \frac{x_n}{x_{n+2}} = \ln \frac{x_n}{x_{n+2}} = 0.05 \]

REFERENCES

EIGHTH INTERNATIONAL CONGRESS
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VIBRATION RESPONSE OF THE ROLLING TIRE

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INTRODUCTION
Tire dynamics are discussed in monograph by Pacejka (1).
In present paper, the response of the rolling tire is formulated in terms of the noncontact tire mode shapes and natural frequencies for contact loading due to the normal contact pressure, breaking, accelerating or cornering.

ANALYSIS
The modes that are required are the noncontact modes.
Natural frequencies and modes shapes are found experimentally or theoretically. By way of the three dimensional dynamic Green's function of the tire, the result in terms of equivalent viscous damping is

\[ \{u_i\} = \sum_{n=1}^{\infty} \frac{u_{in}}{\rho \gamma_n n^2} \int_0^t \int_0^1 \{u_{in}\}^T \{p_i\} e^{-\gamma_n \omega_n (t-\tau)} \sin \gamma_n (t-\tau) dAd\tau \]

The loading on the rolling tire may be expressed in terms of the three components of the contact force resultant

\[ \{p_i\} = \frac{1}{R} \{F_i\} \delta (\theta - \Omega t) \delta (s) \]

EXAMPLE
For instance, results indicate that during pure rolling, a standing wave is created whenever \( \Omega = \omega / n \) (n = 1, 2, ...).
Wavelength is given by \( \lambda = 2\pi R/n \). Extrapolating data from reference (2), using a theoretical frequency prediction, results indicate n = 9, \( \omega = 1260 \) rad/sec, \( \lambda = 17.5 \) cm and a critical speed of 136 km/hr.

CONCLUSION
Results show that it is sufficient to measure natural frequencies and tires for the noncontacting case in order to predict rolling contact behavior. It also underlines need for measuring higher modes than what seems to be present standard practice.

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RANDOM RESPONSE OF PERIODIC STRUCTURES BY A FINITE ELEMENT TECHNIQUE

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INTRODUCTION

A consistent finite element technique is developed for evaluating the response of structures to random excitations. Both finite and infinite periodic structures are considered.

THEORETICAL ANALYSIS

The technique employed is to Fourier Transform the equations of motion of the system. This enables the power spectrum of the response to be calculated. The cross spectral density of the excitation on each element is represented by a polynomial function. For the infinite system the concept of a complex propagation constant (1, 2) is utilised to reduce the size of the structural problem to that of idealising a single repetitive unit. The method is illustrated using random acoustic plane-wave excitation.

RESULTS

As an example of the technique applied to a finite, periodic system, a 5-bay, multi-supported beam is considered. The results are compared with a finite difference solution (3), and a finite element representation using the modal approach (4). These results agree well when the differences in the mathematical models are considered.

A plate model of an infinite skin-rib structure is then examined. The stress spectra obtained are compared with an analytical solution (2). The differences between these results lie within an error band defined by a static analysis of an individual skin plate.

REFERENCES

The equations of motion of a multiterminal linear mechanical system can be written in many forms. In the two more common (1), either the generalized motions (which include both lineal and rotational motions) are expressed in terms of the generalized forces (which include both lineal forces and moments) by means of admittance operators, or the generalized forces are expressed in terms of the generalized motions by means of impedance operators. When these sets of equations are transformed to algebraic form, the operator matrix which relates the motions and forces at the various mechanical terminals of the system is transformed to an admittance matrix, that relating the forces to the motions to an impedance matrix, these two matrices being square and reciprocal to each other. For many multi-terminal problems, the admittance and impedance forms of the equations provide straightforward solutions; however, when multiterminal mechanical systems are to be combined in tandem - as is the most common case - i.e. the output terminals of System I are joined to the input Terminals of System II, etc., it is more convenient to write them in a form in which the generalised forces and motions at the input of each mechanical system are expressed in terms of the generalised forces and motions at the output of the system (or vice versa) by transmission operators, since when these differential equations are transformed to algebraic equations, the rules for calculating the properties of combinations of systems are much more easily stated and applied. When working with the equations in this form, a problem appears to arise when the systems under consideration have different numbers of input and output terminals. For example, if a system has $m$ input terminals and $n$ output terminals, with six degrees of freedom at each terminal, the $12m$ input forces and motions will be expressed by $12n$ equations, each of which contain the $12n$ output forces and motions. Such systems of equations do not, in general, admit of solution, however, it is shown that for the multiterminal network problem, the elements in the transmission matrix can be expressed uniquely in terms of the elements of the admittance and impedance matrices, as can the elements of the left inverse of the transmission matrix.

Curiously the left inverse of the transmission matrix obtained by direct manipulation of the admittance and impedance equations is not equal to the left inverse of the transmission matrix obtained by use of the "usual" formula for a left inverse.

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EIGHTH INTERNATIONAL CONGRESS
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VORTEX EXCITED OSCILLATIONS OF A PAIR OF FLEXIBLE
CIRCULAR CYLINDERS IN FLOWING WATER

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INTRODUCTION
An examination is made of wake interaction effects between two
identical flexible circular cylinders in flowing water. The cylinders
are separated by a variable distance $A$ along a common centre line
in the direction of flow for the range $1.25d < A < 6d$ ($d$ is cylinder
diameter). Oscillations are induced by coherent vortex excitation
in the fundamental and second normal modes of the flow (in-line)
direction and in the fundamental mode of the cross-flow direction.
Results are presented from tests on the uncoupled cylinders and also
from tests with the cylinders coupled by elastic or rigid members.

EXPERIMENTAL
Previous research by the authors (1) has demonstrated that in-line
oscillations of a single cylinder occur in two distinct instability
regions separated by zero response. The first region is characterised
by symmetric vortex shedding and the second by the more familiar
alternate vortex shedding. In air flow, vortex excited oscillations
are almost exclusively confined to the cross-flow direction and the
importance of considering in-line oscillations of structures in
water is emphasised with the results (1) showing that instability
in-line may be excited in velocities of only one quarter those
required for the cross-flow direction. In tests with the two
cylinders, both types of vortex shedding are observed and each
induces substantially different levels of response in-line.
Stability of the coupled cylinders is explained by reference to
the Structural Stability Parameter $K_S$ (1) of the bent and
consideration of the separation and characteristic vortex shedding.
In the cross-flow direction, experimental results correlate well
with available wind tunnel data.

THEORETICAL ANALYSIS
The free and coupled cylinders are analysed by a transfer matrix
method within which the mode shapes are developed and integrated
numerically to formulate the equivalent mass per unit length $m_e$
in the definition of $K_S$ (1).

REFERENCE
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EIGHTH INTERNATIONAL CONGRESS ON ACOUSTICS, LONDON 1974

COMPUTATIONAL METHODS IN FLEXURAL WAVE MECHANICS

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In 1958 Powell(1) formulated the response of a shell type structure to Aero-Acoustic Loads. He expressed the structural response in terms of normal modes, but most flight vehicle structures are spatially periodic with a large number of fundamental periodic elements and the modal density tends to infinity with the number of elements. This fact makes Powell's theory very difficult to apply for two reasons:

- Eigenproblems for deriving normal modes become very large;
- Each point on the response power spectrum involves computation of $n^2$ double integrals where $n$ is the number of modes used to adequately represent the structure.

Periodic structures have been examined by Mead (2,3,4,5) employing a new concept— that of the structure as a medium supporting flexural wave motion identified by "propagation constants". Abrahamson(6,7) has formulated the response of a periodic shell type structure to aero-acoustic loads in terms of free flexural waves. This approach offers immense reductions in computational effort over Powell's analysis since:

- Only one structural periodic element need be considered;
- Eigenproblems are small;
- "Joint Propagation Selection Conditions", $(\mu_P = \mu_s \pm 2n\pi)$ $\mu_{ps}$—Propagation constants for pressure field define precisely the coupling between the aero-acoustic pressure field and the structural response.

Here it is shown how it is only necessary to evaluate the free waves on a grid within propagation constant band widths "2\pi" wide, with subsequent interpolation in the forced response evaluation. This procedure reduces computation substantially over a direct approach, even in problems with strong coupling.

REFERENCES

EINLEITUNG


MATHEMATISCHES MODELL

Von den verschiedenen Modellen, die z.B. Parkinson (1) zusammenstellte, wird hier als das erfolgversprechendste das von Hartlen, Baines & Currie (2) betrachtet. Es geht von zwei gekoppelten Differentialgleichungen aus, einer für den mechanischen Schwinger mit der Luftkraft senkrecht zur Strömungsgeschwindigkeit \( c_L \) als äußerer Erregung

\[
\ddot{x} + \dot{x} + x = A c_L
\]

(1)

und einer weiteren, die für die Luftkräfte \( c_L \) ein nichtlineares System annimmt, das die Schwinggeschwindigkeit des Körpers \( \dot{x} \) als äußere Kraft enthält:

\[
\ddot{c}_L + b \dot{c}_L + g \dot{c}_L^3 + \omega^2 c_L = a \dot{x}
\]

(2)

Mit diesem System lassen sich näherungsweise die Amplitude \( x \) in Abhängigkeit von der Strömungsgeschwindigkeit, die \( c_L \)-Verteilung, die zwischen beiden vorhandene Phasenlage sowie die Schwingfrequenz darstellen.

Weitere Phänomene, die durch dieses Gleichungssystem nicht erfaßt werden, sind z.B. die auftretende Hysteresis in der Amplitude. Das bedeutet, daß bei Vergrößerung und bei Verkleinerung der Strömungsgeschwindigkeit sich nicht dieselben Amplitudenwerte einstellen. Damit verbunden sind die aus nichtlinearen Systemen bekannten Erscheinungen des harten und weichen Schwingungseinsatzes. Um auch diese Phänomene berücksichtigen zu können, kann man in die Gleichung (2) ein nichtlineares Dämpfungsglied, d.h. ein Glied das \( \dot{c}_L \) enthält, von 5. Ordnung einführen. Damit ist es möglich, den Hystereseeffekt zu erfassen und gleichzeitig die Grenzamplituden im Falle des harten Schwingungseinsatzes zu berechnen.

LITERATUR

AN EXPERIMENTAL STUDY OF THE HYDRODYNAMIC DRAG ON COMPLIANT SURFACES: FLUID PROPERTY EFFECTS

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INTRODUCTION

A recent experimental study (R.J. Hansen and D.L. Hunston, to be published) has established that the hydrodynamic drag (and presumably therefore the flow-generated noise) associated with a water flow on an isotropic, compliant surface is significantly larger than on a rigid surface of the same geometry, above a critical fluid velocity. The present paper summarizes experimental studies of the effects on this critical fluid velocity of fluid viscosity and of the addition to the fluid of a drag-reducing polymer additive.

EXPERIMENTAL WORK

Experiments were conducted in the 10.16 cm diameter disk apparatus described elsewhere (R.J. Hansen and D.L. Hunston, to be published). The disk was coated with compliant materials (polyvinyl chloride plastisols) of elastic moduli 5250 ( ), 22,300 ( ), and 53,200 ( ) dynes/cm². Torque vs. rotational speed measurements were conducted in water, Fig. 1 The effect of viscosity on water-glycerine solutions, and aqueous solutions of a polyacrylamide drag-reducing agent (PAM).

The tests in water and water-glycerine solutions of various concentrations established the effect of fluid viscosity on the critical fluid velocity at which increased drag begins. The results are shown in Figure 1. The x-axis is fluid viscosity and the y-axis is the ratio of the critical velocity in the fluid of the indicated viscosity to that in water. At fluid velocities substantially in excess of the critical value in these tests, increases in the hydrodynamic drag by as much as a factor of four were observed due to surface compliance.

Addition to water of 100 weight parts per million PAM increased the critical fluid velocity for the 53,200 dynes/cm² coating by 10%. Above the critical velocity the polymer additive significantly reduced the magnitude of the increased drag associated with the surface compliance. At 1.50 times the critical velocity in water, for example, the drag on the coated disk was 50% less in the polymer solution than in water.
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RESPONSE OF THIN-WALLED CYLINDERS TO AERODYNAMIC
EXCITATION

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INTRODUCTION

Non-linear response of thin walled cylinders is investigated for measured aerodynamic forces. Pressure
and response measurements were carried out on 3-D clamped
free cylinders in the Reynolds number range of \( \approx 1.6 \times 10^5 - 2.9 \times 10^5 \).

THEORETICAL ANALYSIS

The rms value of the fluctuating pressure \( p \) is represented by the truncated Fourier series in the circum-
derferential direction \( \theta \) and by the power law in the axial
direction \( x \) in the form,
\[
p(x, \theta) = p_0 \kappa x^2 \sum_{n} a_n \cos n\theta
\]
(1)

\( \kappa \) and \( \alpha \) are determined by the wind data and \( a_n \)
by the measured pressure distribution. Using the theory
developed in the previous paper (1), the governing non-
linear equation for the loading in eqn. (1) is reduced to
a cubic in the amplitude \( C \) in the radial direction
with the frequency \( \Delta \) as parameter:
\[
\kappa_3 C^3 + \kappa_1 C + \kappa_0 = 0.
\]
(2)
The solution of (2) indicates hardening type of non-
linearity wherein an increase in frequency results in
an increase in amplitude.

EXPERIMENTAL

The fluctuating pressures were measured on 3", 6" and
8\( \frac{1}{2} \)"D rigid cylinders, and 4\( \frac{1}{2} \)" and 6\( \frac{1}{2} \)"D flexible cylinders,
using DISA Type 51F32 low pressure transducers. The
response of flexible shells was measured by strain gauges,
two for axial and two for circumferential modes. The
signals were recorded on Sangamo Type 3562 FM tape
recorder. Power spectra, correlations, coherence and
transfer function were evaluated on HP 5451A Fourier
Analyser. Results from flexible shells indicate "lock-
in"phenomenon, i.e., the influence of structural frequency
on vortex shedding frequency. (R. Nataraja, 1974,

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Linear Dynamics (1972), Paper B3, Loughborough.
The sound radiation from flow with obstacles is generally more efficient than the noise generated by the undisturbed flow. Many instances are reducible to the configuration of a jet impinging to a flat panel. In the subsonic velocity range it is possible to distinguish the following mechanisms of noise generation:

1. The undisturbed jet.
2. Modification of the radiation resistance caused by reflexion from the panel.
3. Deflexion of the flow at the panel.
4. Edge noise from the plate fringes.
5. Feed-back to the vortex generation between panel and nozzle.

The size and distance of the panel was chosen in such a way that edge and feed-back effects were not encountered. To separate the structure-borne sound from the aerodynamic sources experiments were made with panels of different elasticity.

Measurements with a rigid panel gave a radiated sound power level which was at most 6 dB higher than the noise of the undisturbed jet. This effect was independent of flow velocity. The spectrum shows an additional peak at a Strouhal-number based on the panel distance. With an elastic plate the sound power increased further depending on the stimulation and radiation characteristics of the panel. This effect is restricted to the lower frequency range. An attempt is made to explain and calculate this phenomenon.
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FUSELAGE VIBRATION AND INTERIOR NOISE DUE TO BOUNDARY LAYER EXCITATION

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INTRODUCTION

Measured vibration and acoustic radiation characteristics of a fuselage structure excited by a boundary layer, described (1,2), are related here in terms of radiation efficiency.

RADIATION EFFICIENCY

Radiation efficiency $\sigma$ is defined (3) by $\sigma = R_{\text{rad}} / \rho_0 c_0 A$ where $R_{\text{rad}}$ is real part of radiation impedance and $A$ is radiating area. Value of $\sigma$ depends on presence of acoustic fast and slow modes at frequency $\omega$, where $\omega$ is related to ring frequency $\omega_r$ and coincidence frequency $\omega_c$. For simply supported cylinder with preloads $\epsilon_x \epsilon_y$.

$$\left( \frac{\omega}{\omega_r} \right)^2 = \frac{\omega^2}{\omega_r^2} = \beta^2 a^4 k^4 + \sin^4 \theta + a^2 k^2 (\epsilon_x \sin^2 \theta + \epsilon_y \cos^2 \theta) \quad (1)$$

Coincidence frequency is obtained by equating right hand side of Eq.(1) to $c_0^2 k^2 / \omega_r^2$.

COMPARISON WITH EXPERIMENT

Neglecting preloads, radiation efficiency for a fuselage structure is estimated (Fig. 1), and compared with flight test data for fuselage under pressure differential of 8.5 psi. When $\omega < \omega_r$ there is good agreement but when $\omega_r < \omega < \omega_c$ the theory underestimates by up to 10 dB. Inclusion of preloads improves agreement but other factors such as non-ideal cylinder and non-reverberant vibration need to be considered.

REFERENCES

BUT

Diagnostiquer l'état mécanique de soupapes de réglage de turbines à vapeur de grande puissance par mesure et traitement du bruit transmis par voie solide relevé à l'aide d'accéléromètres piézoélectriques sur les parties externes de la soupape, en cours de fonctionnement.

PRINCipe du Diagnostic

Le diagnostic effectué est conforme au schéma de principe général indiqué Fig. 1

DONNEES PARTICULIERES DU PROBLEME

Les conditions soniques étant, en fonctionnement habituel, atteintes au niveau du clapet des soupapes considérées (soit globalement, soit localement dans l'écoulement), une excitation de type impulsionnel (due à l'existence d'ondes de choc aérodynamiques cédant une partie de leur énergie aux structures) se superpose à certains instants à l'excitation permanente de type aléatoire engendrée par l'écoulement de vapeur. Ces deux types d'excitation sont mis en évidence sur la Fig. 2 représentant un exemple caractéristique de réponse vibratoire.

Fig. 2 - Allure temporelle type du bruit transmis par voie solide
DYNAMICS OF A COUPLED SHELL/FLUID SYSTEM

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INTRODUCTION

Flow-induced vibrations are of great concern in the development and design of Liquid-Metal-Cooled Fast Breeder Reactors. One of the problems is the vibration of thermal shield, which is a cylindrical shell placed inside a cylindrical vessel and submerged in sodium. For the purpose of understanding the dynamics of thermal shield and controlling the vibration in reactor internals, this paper presents a study of two cylindrical shells arranged concentrically and containing separated by fluid.

ANALYSIS

The motions of the shells are described by the Flugge's shell equations and the fluid in the thermal shield and in the annular region is assumed to be an ideal compressible fluid. The solution is obtained using the traveling wave type analysis. An exact frequency equation is obtained for the general case and an approximate closed form solution is given for the shell system with an incompressible fluid. Frequency spectra are presented for steel shells containing liquid sodium.

RESULTS

A general method for calculating the frequencies of a coupled shell/fluid system is presented in the paper. Following are some important conclusions: 1. There exist structural modes and acoustic modes. If the structural motions are of primary interest, the fluid may be considered as incompressible. 2. There exist out-of-phase modes and in-phase modes. The lowest frequency of the system is always associated with one of the out-of-phase modes. 3. The lowest frequency of the shell system with fluid is significantly lower than those of the individual shells. 4. The manner of accounting for the effect of the fluid coupling via the added mass concept is described explicitly. 5. The distribution of stretching energy and bending energy of the shells within the coupled system is different from those of the corresponding empty shells.

With the suggested method, the frequency characteristics of thermal shield can be analyzed and design parameters can be explicitly related to frequency. An experiment to verify the theory is being planned.
THE RESPONSE OF LINEAR STRUCTURES TO WEAK EXPLOSIONS

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INTRODUCTION

The finite element displacement method is used to predict the vibrations of box-type structures induced by travelling blast waves in their region of Mach reflections. Modifications based upon experimental data are completely incorporated into the theory for the first time to account for the wave's convection. Although the method may be extended into the plastic region, only weak explosions with an arbitrary limit of 5 psi overpressure in the free field are considered. Thus the analysis is simplified by assuming materials remain linear because structures built to withstand much larger overpressures probably would be underground, invalidating the theory in any case. A series of numerical computations have been performed for three extreme but typically dimensioned buildings in order to assess the feasibility and limitations of simplifying the resulting theory.

THEORY

The structure is divided into a number of zones in the manner suggested by Norris et al (1). Then the theory developed earlier (2) is refined to evaluate average pressures in any one zone as the blast wave's shock front traverses the structure with constant speed. Previous theory for overpressures an order of magnitude less than those considered here, neglects vortices which not only have a variable speed but also change form as they move across the building. This paper presents a novel numerical integration technique to evaluate and incorporate these vortex pressures. Then a 4th order Runge-Kutta technique is used to evaluate the standard equations of undamped motion (2) obtained by applying compatibility and equilibrium conditions at nodal points of elements and assuming adjacent faces, at rest initially, remain perpendicular at their edges.

CONCLUSIONS

It is found that vortices become more important as the strength of the blast wave is increased. Also, in the limited range of overpressures considered, a designer reasonably may assume that doubling the free field overpressure essentially will double the structure's response. This helps to explain the reasonable correlation between previous sonic boom theory and experiment where vortices were neglected as low free field overpressures (~2psf) are usual there.

Except for the front face, cross-coupling between individual faces can be significant so that then it is inappropriate to assume that these faces behave as simple plates or oscillators. The relative importance under these circumstances of the wave's convection is discussed fully.

REFERENCES

INTRODUCTION. Noise in accommodation on board large ships is generally dominated by structureborne sound from main and auxiliary machinery and propeller. For the purpose of predicting noise in cabins, knowledge of decrease of vibration level along different hull transmission sound paths is essential. Some experience values are available, e.g. 5 dB reduction per deck (1).

EXPERIMENTS. To investigate the relative importance of different construction details for the total vertical transmission in the structure from machinery foundation to cabins, some experiments have been carried out on a model of a cross section through a machinery room, casing and deckhouse (scale 1:5, length 5 frame distances), see drawing. The investigations include the influence of longitudinal and transverse frames, webframes, and decks on the total transmission to cabins.

METHOD. The results consist of the transfer functions $F/p$. $F$ is the force ($F_x$, $F_y$, or $F_z$) at the foundation, and $p$ is sound pressure in the cabin. The measurements have been carried out by using the reciprocity theorem:

$F = F \frac{X_y}{p} = \frac{V}{x} \quad \text{Direct experiment.}$

$F = F \frac{X_y}{p} = \frac{V}{x} \quad \text{Reciprocal experiment.}$

$V$ is volume velocity in cabin, $v$ is velocity at foundation (2).

The fig. shows as example the transfer function $(F/p)$ with deck 0 and deck 4 only connected with a 3 mm unstiffened hullplate; decks 1, 2, and 3 were not mounted. $L_F$ is the force level re. $10^{-5}$ N, and $L_p$ is the sound pressure level re. 20 $\mu$Pa in cabin 1. The fully drawn curve shows results of direct experiment, the dotted shows results of the corresponding reciprocal experiment.

INTRODUCTION
In connection with the development of methods for predicting structure-borne sound transmission in ships, extensive measurements have been carried out on board three different types of cargo-ship during normal operating conditions. Among other things we have tried to estimate the importance of other waveforms than bending waves for the transmission processes.

EXPERIMENTAL PROBLEMS
Many measurements with triaxial accelerometers have been made. We were thus able to divide the vibrational motion into three perpendicular directions. The division of the motion into different possible waveforms will generally be incomplete, e.g. for plates we shall get a bending-wave component and an in-plane wave component, consisting of longitudinal and transverse waves. Another problem is that the different waveforms introduce motions of the accelerometer in all three directions, e.g. due to the torsional motion of the bending waves. In addition we have to regard the cross-axis sensitivity of the transducers.

EXPERIMENTAL RESULTS
We only report the general situation for plate elements such as the hull, bulkheads and decks. The measured velocities of the stiffening elements are more difficult to systemize and interpret.

Typical values of the velocity levels for a section of the hull are presented in Fig. 1. There is little difference between bending-wave and in-plane wave velocities. Since the modal density of bending waves are far higher than the modal densities of longitudinal and transverse waves, the modal energies of in-plane waves will be significantly higher than the modal energy of bending waves.

CONCLUSION
With dealing with structure-borne sound transmission in ships, the influence of other waveforms must be considered.

ACKNOWLEDGEMENT
The work has been supported by the Swedish Ship Research Foundation and by the Foundation of Axel och M. Ax:son Johnsons stiftelse.

Fig. 1. Measured velocity levels for a hull section, —— Bending waves, --- In-plane waves, ... Difference
THE EFFECT OF MATERIAL STRAIN-RATE SENSITIVITY ON THE DYNAMIC ELASTOPLASTIC RESPONSE OF THIN PLATES TO BLAST LOADING

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The effect of material strain-rate sensitivity on the elasto-plastic response of thin plates to laterally applied blast loading is investigated using an analysis based on the finite element method and employing hybrid (Pian-type) elements. Applications lie in the fields of military structures and explosive forming.

Elasto-plasticity is incorporated using the flow theory of plasticity. The Prandtl-Reuss relations and isotropic hardening are employed, together with an incremental residual force approach (1). Material strain-rate sensitivity is included by assuming that its effect is to translate, without distortion, the uni-axial elasto-plastic stress-strain curve parallel to its elastic modulus. This is achieved by defining the instantaneous uni-axial stress at first yield, \( \sigma_{u1} \), as a function of the instantaneous equivalent strain rate, \( \dot{\varepsilon} \), viz:

\[
\sigma_{u1} = \sigma_{u1}^S \left[ 1 + \frac{\dot{\varepsilon}}{G} \left( 1 + \frac{f}{G} \right) \right]
\]

where \( \sigma_{u1}^S \) is the quasi-static first-yield stress, and \( f \) and \( G \) are also material constants.

The equations of motion of the system are integrated using an incremental predictor-corrector procedure employing a Runge-Kutta fourth-order extrapolation technique. Geometric non-linearities due to large displacements are also optionally accounted for.

Computer programs to perform the above analysis have been written in FORTRAN IV and run on an ICL 1901A. Figs. 1 and 2 show typical results obtained for a square, simply-supported plate of L73 Aluminium Alloy without and with large-displacement effects, respectively. \( q_c \), \( t \) and \( 2a \) are the central deflection, thickness and side-length of the plate, respectively. It is demonstrated that when high-energy loading is involved, material strain-rate effects may be significant.

Fig. 1 Strain-Rate Effects on Responses (Small Displacements)  
Fig. 2 Strain-Rate Effects on Responses (Large Displacements)

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EIGHTH INTERNATIONAL CONGRESS ON ACOUSTICS, LONDON 1974

ANALYSIS OF RANDOM VIBRATION OF NON-LINEAR SYSTEMS BY A PERTURBATION METHOD

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THEORY

The one degree of freedom mechanical system

\[ \ddot{x} + f(x, \dot{x}) = n(t), \]

where \( n(t) \) is a random force with a flat power spectral density, can be analysed by means of the Fokker-Planck-Kolmogorov equation

\[ \frac{\partial p}{\partial t} = \frac{\partial^2 p}{\partial x^2} + \frac{\partial}{\partial x}(fp) - \dot{x} \frac{\partial p}{\partial x} = L_p, \]

where \( p \) is the transition probability density of \( x \) and \( \dot{x} \) (1). Statistical properties of the system can be expressed as expansions in the eigenfunctions of the differential operator \( L \). These are known analytically only for the linear system, where \( f(x, \dot{x}) = 2\xi \omega_n \dot{x} + \omega_n^2 x \), and where infinite excursions of \( x \) and \( \dot{x} \) are permitted. In this case, the eigenfunctions are two-dimensional Hermite polynomials (2). In other cases, write \( L = L_0 + \varepsilon M \) where \( L_0 \) is the value of \( L \) in the linear case, and find a solution in powers of \( \varepsilon \) (1,3).

EXAMPLE - DUFFING'S EQUATION

Here \( f(x, \dot{x}) = 2\xi \omega_n \dot{x} + \omega_n^2 (x + \varepsilon x^3) \). The coefficients in the perturbation series can be evaluated explicitly. Spectral densities found thus are compared with the results of numerical simulation.

THE THRESHOLD CROSSING PROBLEM

The failure of a vibrating structure is often associated with a variable (displacement or stress) first exceeding a specified value, \( |x| = a \) say. The corresponding Fokker-Planck-Kolmogorov equation has "absorbing" boundary conditions at \( |x| = a \) (4), and the statistics of the time to failure can be expressed in terms of its eigenfunctions. These can be found as a boundary perturbation (3) of the linear problem with infinite boundaries (\( L_0 \)). The first order result is the same as that obtained by the assumption of independent crossings (4).

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EIGHTH INTERNATIONAL CONGRESS ON ACOUSTICS, LONDON 1974

A TRANSIENT EXCITATION FUNCTION FOR USE IN THE MEASUREMENT OF DISPERSIVE WAVE PROPAGATION CHARACTERISTICS

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INTRODUCTION

The wave propagation approach is of increasing use in vibration problems and it is desirable that related experimental methods be made available. Many propagation mechanisms are dispersive and previous methods have had limited success. Conventionally, wideband excitation signals are necessary to achieve distinctive peaks in correlation functions; such signals disperse with distance and the correlation deteriorates (1). The requirement is evidently for an essentially single frequency signal which yields clear correlation functions.

THE TRANSIENT TONEBURST SEQUENCE

This requirement has been met by intermitting a pure tone according to a rapidly swept squarewave to produce a transient sequence of tonebursts of decreasing duration and interval. The squared response signal is then correlated with the intermitting sequence (2). This indicates how the envelope of the signal propagates and thus provides a measure of the group velocity and attenuation at the frequency of the tone. The deterministic signal permits the accurate prediction of spectra and correlation functions, examples of which are given for a practical case.

![Graph showing transient toneburst sequence](image)

Fig. 1. The transient toneburst sequence in flexural wave propagation

CONCLUSIONS

The usefulness of the technique has been demonstrated by a wide range of measurements; in particular, the transient signal considerably shortens analysis procedures (2). The ability to measure characteristics at a single frequency should be useful in tests on periodic systems and waveguides where velocity varies rapidly with frequency.

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A STETHOSCOPE FOR MONITORING SOUNDS FROM NUCLEAR FAST BREEDER REACTORS

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To prevent progressive damage in nuclear reactors, incipient malfunctions are acoustically diagnosed. A 10 m stethoscope conducting sound out of the reactor consists of a long metallic waveguide, a transducer, an absorber, and an intervening transition region, fig. 1.

The absorber minimizes resonances in the waveguide, thereby assuring flat transfer characteristics. High absorption per unit length is achieved with a low sonic velocity material comprised of heavy metal (Pb, W) particles (40-2000 μm) embedded in an elastomer matrix. In terms of sound velocities \( c_e, c_m \) and densities \( \rho_e, \rho_m \) in the elastomer and metal, respectively, the velocity in the mixture having metal volume fraction \( y \) is given by:

\[
c^{-2} = y(1-y)(c_e^{-2}\rho_m/\rho_e + c_m^{-2}\rho_e/\rho_m) + y^2c_m^{-2} + (1-y)^2c_e^{-2}.
\]

The lowest value of \( c \) occurs near \( y = 0.5 \), the velocity being about one-half the velocity in the solid elastomer. The stethoscope absorber had a density of 6 kg/lb and a sound velocity of 80 m/s.

The high specific acoustic impedance \( \rho c \) of the waveguide \( (40 \times 10^6 \text{ kg m}^{-2}\text{s}^{-1}) \) is matched to that of the absorber \( (0.5 \times 10^6 \text{ kg m}^{-2}\text{s}^{-1}) \) through the transition region. This region consists of a series of nesting re-entrant cones of materials having progressively lower specific impedance and greater sectional area so that for all planes normal to the axis the total impedance is constant. The condition for constant impedance between media with specific impedances \( Z_a, Z_b \) and diameters \( a, b \) and cone angles \( \alpha, \beta, \gamma \) is:

\[
\sqrt{Z_b/Z_a} = a/b = \tan \gamma/\tan \beta = \sqrt{1 - \tan \alpha/\tan \beta}.
\]

This matching of an absorber to a waveguide with 80 times higher specific impedance reduced reflected intensity to less than 10% in the frequency region 0.05-50 kHz.

*Work performed under the auspices of the U.S. Atomic Energy Agency.
METHODIC PRINCIPLES OF HYGIENIC ASSESSMENT OF MACHINES POSSESSING VIBRATION-ACOUSTICAL EFFECT ON MAN

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Objective information for the hygienic assessment of machines may be obtained only if one has a clear view of the relations in the system of "man-machine-work object-environment-persons involved into the system". With this end in view, a classification of intersystem relations is suggested making it possible to plan a method of investigation. The results of experiments with machines of various types made it possible to determine quantitative interaction of separate elements of the system and their function in assessing the machine as a whole.

The response of the operator to vibration effect depends upon his individual properties. For the hygienic assessment of hand machines a principle of the selection of operators is suggested taking into consideration the original threshold of response to vibration and the temperature of the skin of the hands. It is to be noted that the operator's individual properties may change the vibration characteristics of the machine in the range of 2 - 5 dB.

The assessment of the machine depends upon its design, technical state, service conditions etc. (1,2), the physico-strength properties of the work object may influence the vibration parameters of the machine (10 dB), the pressing force (3 kg) and physical strain.

Environment influences all the elements of the system. Its effect on man is the most pronounced.

Persons involved into the system are those who are in the zone of unfavourable influence of other parts of the system. When making a hygienic assessment of the machines, account must be taken of the ratio of the number of involved persons to the number of operators and involved persons.

As far as the method is concerned, a selection of the most typical cases of the interaction of all the parts of the system under consideration is essential for the assessment of the machines.

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A SCALE OF HUMAN REACTION TO WHOLE BODY, VERTICAL, SINUSOIDAL VIBRATION

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INTRODUCTION

Since previous studies which have attempted to assess the subjective impression of vibration intensity analogous to loudness have reached differing conclusions, further investigation was felt necessary.

EXPERIMENTAL

A relative intensity estimation procedure was used to obtain observers' estimates of changes in vibration intensity. In each test session, a pair of vibration stimuli of the same frequency were presented alternatively to the subject, the second stimulus being the more intense. When the subject judged how many times more intense the second stimulus was than the first, the experimenter withdrew the stimuli from the alternating sequence and introduced a new pair for judgement.

RESULTS

The experiment demonstrated that individual subjects were able to consistently estimate the relative intensities of two sinusoidal vibration stimuli over an objective acceleration ratio range of up to 6:1, using various reference stimuli in the frequency range 5 to 80 Hz. Logarithmic plots of subjective estimates against objective ratios produced good straight line growth functions, suggesting the validity of a power function. By pooling the data, it may be shown that a power law of the form \( Y = kX^{0.93} \) adequately describes human response to whole body, vertical, sinusoidal vibration, where \( Y \) is the subjective magnitude and \( X \) the objective magnitude.

By combining these results with those of an earlier study (1), a scale of human sensitivity to vibration is proposed. This scale comprises a set of equal comfort contours, named VICS (Vibration Contours), used in conjunction with the VIM scale (Vibration Magnitude), which corresponds to subjective estimates of relative vibration magnitudes. With one VIM equal to 40 VICS, the relationship between the units is

\[
\text{VIM} = \frac{\text{VIC} - 40}{10}
\]

These scales are, therefore, of the nature of the phon and sone scales in acoustics.

REFERENCE

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ANALYSIS TECHNIQUES FOR MACHINE HEALTH MONITORING

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INTRODUCTION

Analysis of the vibration signals is being used increasingly for monitoring the condition of operating machines, for both detection and diagnosis of faults. The present paper discusses the various techniques which can be used for this purpose.

DETECTION AND DIAGNOSIS

The most common analysis technique is spectrum analysis, for two main reasons:

1. Changes in minor spectral components will not always affect the overall vibration level, but can be detected by spectrum analysis.

2. The frequency at which a change occurs will often give an indication of the source.

The types of faults which can be detected and diagnosed using frequency analysis (1–8) include unbalance (at shaft speed, primarily radial), misalignment and bent shafts (shaft speed and low harmonics, radial and axial), oil whirl (slightly less than half shaft speed), hysteresis whirl (shaft critical frequency) and developing turbulence (blade and vane passing frequencies). Changes in natural frequencies can indicate crack growth or build-up of fouling. Local faults in rolling element bearings manifest themselves both at impact frequencies and component natural frequencies (9,5). Growth of sidebands indicates modulation by frequencies corresponding to the sideband spacings, eg, gearbox faults often cause modulation of the tooth meshing frequency and its harmonics (10,5).

Cepstrum Analysis is very useful in detecting and evaluating sideband growth, since it detects periodicity in the logarithmic power spectrum by carrying out a further spectrum analysis on it (11).

Spectrum analysis is not always the most powerful diagnostic tool. For example, with slow speed reciprocating machines the exciting forces tend to be impulsive and thus excite the natural frequencies of the structure (Prieide & Grover, 7). More information as to the source may then be given by the time of occurrence of the impulse, and a useful technique here is Time Domain Averaging (5,6) which enhances regularly repeating phenomena with respect to random fluctuations in the time signal.

REFERENCES

INVESTIGATION OF SOUND RADIATED DURING ELASTIC AND PLASTIC IMPACTS

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INTRODUCTION

The sound from the impact of two bodies is influenced in particular by the closing velocity and by the state of the material during impact. A different sound would be expected from an elastic impact than from an impact which caused a plastic deformation. By dropping a series of steel balls onto a plate of large mass the impact sound from both elastic and plastic impact was investigated.

RESULTS

Fig. 1(a) shows sound pressure level plotted against the height through which a 0.034 kg ball was dropped. A compression test for the same material as the ball and plate is shown in Fig. 1(b). The two curves are similar and in particular in Fig. 1(a) there is an area which corresponds to the yielding of the steel. This leads to the concept in Fig. 1(a) of a critical yield height (or critical yield velocity), above which plastic deformation will occur for a particular ball. A plot of sound pressure against height shows more clearly the distinction between the two regions.

From the theory of elastic impact (1)(2) the compression, during impact of a ball on a heavy plate can be obtained. This compression at the critical yield velocity is of the same order as that obtained at yield in the elastic compression test of Fig. 1(b), which represents statically a situation corresponding to the impact. These results show that there is a relationship between the sound and the impact stress and this relationship will be the subject of further investigation.

REFERENCES

MEANINGFUL CONFIRMATION OF WIDE-BAND RANDOM TESTS

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INTRODUCTION

Only the most modern equipment for controlling wide band random tests will allow performance of reproducible tests simply and quickly. Normally, the simple and quick test is not easily reproducible.

The reproducibility of a test depends on the relationship between the selectivity of the resonances in the test object and the narrowness of the equalizer/analyzer filters used.

For flexibility, test specifications have been set up defining tests with various reproducibility levels. Each test performed is required to conform to the reproducibility objectives prescribed and measurement must be made of system frequency response, acceleration spectral density during test, and overall RMS level applied to the test object.

METHODS

Test confirmation methods must themselves be simple, quick and contain minimum error.

Frequency response measurement is trivial with a constant level sine input to the shaker system.

Acceleration density can be confirmed using simple, sweep filter techniques. The long averaging times associated with low frequency measurements often curtails the usefulness of this method in conjunction with short duration tests, unless supplemented by a tape recorder. A useful technique using a solid-state digital recorder is described for increasing the flexibility of the sweeping filter method and significantly decreasing the total measurement time.

Alternatively, the fixed filters in the equalizer/analyzer can be used in the test confirmation. Considerable advantage can be gained by using a time-compression real-time analyzer. The bandwidths of such an analyzer is much narrower than the bandwidths of the equalizer/analyzer thus immediately indicating areas of poor equalization.

Another useful method utilizes a multi-channel tape recorder and a tracking filter. A sine steering signal is recorded on one channel of the recorder and the signals to be analyzed on the other channels. Subsequently the sine wave is used to control the filter centre frequency. This method retains full phase relationship between the signals measured.

The overall RMS measurement can usefully be supplemented by an out-of-test-band measurement to indicate distortion.

CONCLUSION

This paper shows various methods for obtaining confirmation of test. The method selected depends on the reproducibility level required by the specifying authority.
INTRODUCTION
Although it is recognised that the handling of vibrating tools can produce vibration induced white finger (VWF), there does not seem to be a clear indication of those aspects of the vibration characteristics which are important in the production of white finger. Ideally, a study of the vibration characteristics of a number of engineering processes should show a clear distinction between those processes which are known to cause white finger and those which do not.

PARAMETERS TO BE MEASURED
The vibration signal varies randomly with time and is therefore best described in terms of statistical parameters.
1) The mean square value $(\bar{X})^2$ and 
2) Power Spectral Density Function $G(f)$
The power spectral density function is defined as the mean square value of a filtered signal divided by the bandwidth of the filter i.e. 
\[ G(f) = \lim_{\Delta f \to 0} \int \frac{1}{\Delta f} \left( \int X^2(t,f,\Delta f) dt \right) \]
In practice it is usual to use octave $(\Delta f = 0.7f)$ or $\frac{1}{3}$ octave $(\Delta f = 0.2f)$ band filters. The RMS signal in each band is then obtained as a function of frequency.

EXPERIMENTAL PROCEDURE
Measurements were made using a piezo-electric accelerometer which was attached firmly to the vibrating object. The signal from the accelerometer was fed into a high input impedance preamplifier and subsequently recorded on an FM tape recorder. The system had a flat frequency response from 2 Hz - 3000 Hz. Above about 125 Hz, the recorded signal could be analysed directly by using a filter set and measuring the RMS value in each octave. Below 125 Hz the short lengths of recorded signal made it necessary to re-cycle the signal by using a digital event recorder. In this way the signal could be averaged for as long as is necessary to obtain a satisfactory spectrum.

RESULTS
Data will be presented which shows spectra which were recorded from Pedestal Grinders, Power Chain Saws and Pneumatic Hand Tools. Evidence will be given which indicates that the incidence of VWF is likely to be greater at low frequencies than at high frequencies. Comparison of data with existing Damage Risk Criteria for hand arm vibration will be made and comment will be made on the difficulties to be encountered in the evaluation of broken and interrupted exposure patterns.
INTRODUCTION

The influence of torsional vibrations on instrumentation tape recorders was investigated since rotating mechanisms are normally prone to such vibrations. In addition measurements were carried out to determine if significant torsional vibrations existed in environments where typical recordings of measurements need to be carried out.

EXPERIMENTAL ARRANGEMENT

The tape recorder to be tested was excited torsionally by two small vibration exciters in the frequency range 3 Hz to 1000 Hz, at constant vibration levels between 1.6 and 40 rad/s² while a pre-recorded tape was being played back. During the frequency sweep the flutter of the tape recorder was continuously recorded.

For measurement of torsional vibrations in the environment, a portable set of instruments was arranged for tape-recording in situ and later analysis in the laboratory.

MEASUREMENT RESULTS

The measurements revealed that the lower the tape speed the more sensitive the tape recorder was to torsional vibrations. However, it was found that the "differential capstan drive" principle in tape recorder construction was more immune to torsional vibrations than more conventional systems.
EINLEITUNG

Als Grundelement moderner Fernsprechwandler (z.B. Mikrofon) wurde eine piezoelektrische Biegeplatte (gleichzeitig schallumfängendes und -wandeldes Teil) gewählt. Sie besteht aus dem Träger (AlMgSi, ø = 43mm, h = 250µm) und der Piezokeramik (PZT, ø = 30mm, h = 150µm, k ≈ 50 %). Bei der speziell ausgesuchten Randlagerung in einem Silikonring liegt die Grundresonanz innerhalb des Fernsprechübertragungsbereichs.

EXPERIMENTE

Akustisch oder elektrisch angeregt schwingt die Piezoplatte gegen ein definiertes Luftvolumen V, in dem der erzeugte Schalldruck und die Auslenkung der Platte (holographische Interferometrie) gemessen wird. Die Messergebnisse bestätigen gut die erwähnten Abhängigkeiten für die Grundresonanzfrequenz \( f_0 = k_1 + k_2 / V \) und den in der Druckkammer erzeugten Schalldruck \( p = c_1 / (c_2 \cdot V + c_2/2) \). Aus den gemessenen Werten werden die äquivalenten Größen (Fläche, Steife, Masse) für die Kolbenmembran mit gleichem Verschiebungsvolumen und gleicher Auslenkung ermittelt (z.B. \( S_{\text{eff}} / S = 0,36 \)).

ERGEBNIS

Die Überhöhung von ca 25 dB bei der Grundresonanz der Biegeplatte (V = 1 cm³, \( f = 1,6 \) kHz) wird von akustischen Resonatoren absorbiert, so dass man einen ausgeglichenen Frequenzgang des kompletten Wandlers erzielt. Bei Beschallung mit 1 Pa ist die mittlere abgegebene Spannung des Mikrofons etwa 5 mV und die Mittelpunktlauslenkung der Platte nur etwa 0,05 µm.
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THE INFLUENCE OF ELECTRODES ON THE RESONANCE FREQUENCY TEMPERATURE DEPENDENCE OF COUPLED FLEXURAL AND THICKNESS SHEAR VIBRATIONS OF AT-CUT QUARTZ PLATES

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INTRODUCTION

The influence of electrodes on the resonance frequency temperature dependence of AT and BT quartz resonators was experimentally studied by Suk. In the report the theoretical considerations of the mentioned problem are given.

METHOD OF SOLUTION

Equally as in (1) we considered the rectangular quartz plates plated in the centre part by an electrode having the shape of a strip with the length smaller than the length of the plate. Unlike in (1) we expressed the elastic characteristics of the plate for the plated portions by the stiffnesses \( \gamma'_{12} \), \( \gamma'_{25} \) and \( \gamma'_{66} \) correspond to constant electric displacement and for the unplated portion by the stiffnesses \( \gamma^E_{12} \), \( \gamma^E_{25} \) and \( \gamma^E_{66} \) corresponding to a constant electric field. When calculating the resonance frequency temperature dependence of AT cut quartz plates we supposed that change of temperature involves changes in the values of the mentioned elastic stiffnesses on the one hand and in the density of the quartz as well in the thickness and length dimensions of the plate on the other hand. The influence of temperature was considered similarly as in (2) and the first order temperature coefficients of stiffnesses were taken from (3).

OBTAINED RESULTS

The resonance frequency temperature dependence of the square AT-cut quartz plates with length - to thickness ratio equal 20, 30 and 40 and with orientation angle 35°18', 35°21 and 35°24 was calculated. The length of electrodes - to length of the plate ratio was within the range from 0.25 to 0.75 and the thickness of electrode expressed by factor R between 0.0 to 0.04 were considered. In the report the comparison of measured and calculated temperature characteristics are given.

REFERENCES

INTRODUCTION

The Crucible is one of the most recent thrust stage theatres to be built. Little work has been done on the acoustical performance of this stage form in spite of obvious problems caused by the directional characteristics of human speech.

ACOUSTICAL PERFORMANCE

1) Noise Control
Whilst good design and construction prevent problems due to external noise sources measurements made inside the auditorium demonstrated that the desired NR20 level was being exceeded. Noise from the ventilation plant resulted in typical levels of NR30.

2) Reverberation Time
The measured reverberation times of Fig. 1 show that the reverberation characteristics are acceptable.

3) Sound Distribution
The measured sound distribution emanating from a point source on the stage demonstrated that the auditorium was free of dead spots. This technique, however, does not yield any information as to the efficiency of the reflectors intended to improve speech intelligibility. A statistical technique was employed to measure the attenuation of sound from a human speaker situated on the stage. Attenuation contours of speech sounds in the 1/3 octave band centred on 1000 Hz are shown in Fig. 2. The contours are of the level exceeded for 5% of the time to minimise the effects of background noise and reverberant sound. Whilst the directionality of speech is apparent the effect is not as marked as would be experienced in free field conditions.

CONCLUSION

The acoustical performance of the Crucible satisfies all design requirements apart from the background noise levels which are to receive attention in the near future.
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EARLY LATERAL REFLECTIONS AND CROSS-SECTION RATIO
IN CONCERT HALLS

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Many acousticians have commented on the desirable enveloping effects of
early lateral reflections in concert halls. Marshall (1) has suggested
that these effects, called here spatial impression (SI) have fre-
quently been ascribed to reverberation. Both Marshall (1) and West
(2) have proposed that the cross-section ratio in rectangular concert
halls (the ratio of hall width to height) determines the perceptibility
of early lateral reflections, i.e., the degree of spatial impres-
sion.

Subjective experiments (reported in part in (3)) have been conduc-
ted with simulated reflections in an anechoic environment to determine
the physical correlate of the degree of SI. It was found that the
degree of SI is substantially independent of reflection delay, but is
a function of reflection level, and direction relative to lateral. In
multiple reflection situations, the degree of SI is related to the
ratio of lateral to non-lateral early energy, S, assuming incoherent
energy addition, as in equation (1):

\[
S = \left[ \sum_{t=5\text{ms}}^{80\text{ms}} P \cos \phi \right] \div \left[ \sum_{t=0\text{ms}}^{80\text{ms}} P(1 - \cos \phi) \right], \tag{1}
\]

where P is the 'reflection' level ('reflection' including direct sound
here) and \( \phi \) is the angle of incidence of the reflection relative to the
axis through the listener's ears.

Impulse response measurements of the ratio, S, in two Australian
rectangular halls showed little variation of the quantity within halls.
Computer predictions (assuming specular reflection) substantiated this
result, particularly for smaller halls. Computer predictions suggest
that the degree of SI is a function of hall width but not hall height.
This can be explained by reference to the image array responsible for
early reflections. The small variation in hall height in contemporary
halls may explain the good correlation found by West (2) between cross-
section ratio and acoustical quality. Consideration of the effect of
audience attenuation at grazing incidence further complicates the
situation, but for halls of average height the significance of hall
width remains.

REFERENCES


* This work was conducted at ISVR, University of Southampton and
School of Architecture, University of Western Australia.

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CATHEDRAL ACOUSTIC REINFORCEMENT DESIGN

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The great Christian cathedrals were not designed for all the purposes they now have to serve, and acoustic requirements have become more exacting. Originally devotional activity was restricted to a small area, usually screened off, enclosing choir, chancel, High Altar, places for the clergy, and a few worshippers. The lay public only came for great events, and communication was less important than the sense of occasion. The screening often still exists, as at Westminster Abbey.

19th Century saw a growth of popular preaching to large congregations, and the splendour of long reverberation (6 secs for Westminster Abbey; 12 secs for St. Paul's) became a serious handicap. Today it is expected that not only will preaching be intelligible, but also speech and chanting from high altar, choir area and chapels, often widely separated. Delay systems are essential for clarity and realism from all positions, and become complex. Because the High Altar is at the east end, delays operate westward, but ceremonials often begin at the west portal and an eastward system is then required. Loudspeakers must generally be columns propagating sound direct to listeners to avoid arousing reverberation.

Some problems are subtle. Celebrants at the High Altar 100-150m distant from some listeners and out of sight should not be so loud and clear as the nearer and clearly visible preacher. There is a desire to correct for voice deficiencies of clergy. Speech, chanting, choir and organ must be in sensible balance, which generally implies that the choir needs reinforcement. This is particularly important and there are several reasons for this. Fundamentally a choir is disadvantageously placed, for it is at much the same level as the mass of people, and its direct sound is absorbed passing over them. Or it may be behind a screen. The choir sound is then weakened and dependent on reverberation. In choral responses between celebrant and choir, if the former has speech reinforcement and the latter has none, the imbalance is then unpleasant. A great imbalance between organ and choir also occurs because organs are not only advantageously located, but louder than formerly.

The column loudspeakers present an aesthetic problem. Existing types need tilting 7-10°, and are still somewhat bulky. Slimness is desirable, usable in a vertical position, i.e., with an acoustic tilt. An improved type has been designed by the Sound Advisory Panel of Westminster Abbey. This is slim and has an acoustic tilt of 3 or 4°, but a further tilt is desirable.

The whole purpose in cathedral system design should be, not to replace the natural acoustic, but to supplement it without distortion.
We describe the results of a correlation analysis between subjective preference comparisons and objective parameters of concert halls.

For the subjective evaluations, a piece of music was recorded in the halls with a specially designed artificial head(1). These recordings were played back over a compensation filter and two loudspeakers in an anechoic chamber(2). Thus, acoustic signals are created at the ears of a human listener which are nearly identical to those of a listener in the hall. This method allows instantaneous comparison of the acoustical qualities of different halls under realistic free-field listening conditions on the basis of identical musical source material. Listeners are asked to make preference judgments of musical quality in paired comparison tests. The raw preference data are processed by multidimensional factor-analysis programs yielding one "consensus preference factor" and several "individual difference factors" (3).

Concerning the objective data, the geometric parameters are obtained from the drawings of the halls, while the acoustic parameters are obtained from impulse responses measured at the ears of the artificial head with a powerfull spark source on the stage.

The results of the correlation analysis between the subjective and the objective data show that the variety of acoustic parameters introduced in the past can be reduced to a few significant different parameters. The importance of the reverberation time, definition, interaural coherence and initial time delay gap for the judgments of the listeners is discussed in detail. Furthermore, the relations between the acoustic parameters highly correlated with the "consensus preference factor" and the geometric parameters (volume, width, height) are considered.

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ACOUSTICAL DESIGN OF A MULTI-PURPOSE HALL IN "IBSEN HOUSE"

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PROJECTING HALLS BY COMPUTERIZED RAY TRACING.

The use of computerized 3-dimensional ray tracing in projecting halls opens new possibilities for the prediction of the acoustical behaviour of halls. This model technique enables us to study, simultaneously, the space, time and directional distribution of reflections on the audience area. We used this computer model in the projection of a multipurpose hall in "Ibsen House", investigating various ceiling constructions to get a uniform and high speech intelligibility. As an example are shown impacts of rays with time delay < 20 msec. as dots in a plan of the hall.

CONTROLL MEASUREMENTS OF PULSE RESPONSE.

The computed time distribution was confirmed by echograms from pulse measurements. The echograms computed by integrating energy over a small area within the audience area show similar shape.

CALCULATION AND CONTROLL MEASUREMENTS OF THE REVERBERATION TIME.

The projected reverberation time was 1.3 sec. over the middle frequency range. Measurements indicated 1.2 - 1.3 sec. for 250 - 3150 Hz, both with and without audience. From measurements before installation of the heavy upholstered seats, we gained information about the seat absorption. The scene house is heavily damped, and the reverberation is nearly as in the hall.

CONTROLL MEASUREMENTS OF THE SPEECH INTELLIGIBILITY.

Our method uses nonsens double test words, constructed of sequences of consonants (c) and vowels (v) in the order:

cvcvcvc   cv(cvc)vc

for instance: MEIVIKJASJ FU(byp)AUT

An untrained test audience of 600 college students answered our request for the 3 sounds in the square bracket. The mean value of 3 correct sounds for 4 x 4 seats are shown in a plan of the hall. The results indicate high speech intelligibility in the major part of the hall.

CONCLUSION.

Computations of sound distribution, using sound rays, are confirmed by controll measurements. Subjective impressions from opera and theater performances in the hall are favourable.
STUDIES OF IMPULSE RESPONSE IN MODEL AUDITORIA

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INTRODUCTION

This paper reports ongoing research into modelling of auditoria being carried out by the Department of Building, Heriot-Watt University. Steady-state and impulse response methods are being applied to model auditoria with the intention of gaining insights into the design of auditoria.

DISCUSSION OF AREA OF STUDY

Model study in acoustics using high-frequency sound, is divided broadly into two categories:

a. programmes of recorded anechoic music and speech passed through the model auditorium and after suitable processing, presented to a surrogate audience. Judgement is primarily subjective and may use descriptive terms unable to be supported by tenable causal hypotheses.

b. measurements which are the direct equivalent of measurements carried out in real auditoria, and are objective but indirect measures of acoustic performance. This would include the study of both steady state and transient conditions.

In our work, particular attention is being paid to transient phenomena. Much previous study has been concerned with CRO oscillograms where attention has been paid to potential echoes and relative energy distribution in the impulse response in real auditoria. With appropriate instrumentation it is intended to extract more information from the impulse response. Seven perceptual aspects have been identified for possible study:

1. Directional Impression (location of source)
2. Colouration
3. Echo
4. Reverberation (as distance impression)
5. Volume Impression (ratio of early to reverberant sound) (1)
6. Spatial Impression (breadth of source) (1)
7. Loudness Impression

Further information gained in these areas would considerably extend the usefulness of the model as an aid to the design of auditoria.

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The Royal Northern College of Music in Manchester was commissioned from the authors' firm in 1963. Its construction was delayed and it was not completed until 1972. Its principal accommodation is 90 practice and tutorial rooms, an opera/theatre, a concert hall, a recital room and several specialist and social rooms. Strict economy had to be observed in the design.

A major problem in music colleges is that the numbers of students to be exercised in public performance is large in relation to the audiences to be expected. Stages have to be full-size, but audience areas are disproportionately small in relation to them. The seating is for 600 to 750 in each auditorium.

The seating area of the opera is a single-slope rake. The walls, of exposed brick, are mainly parallel, but angle inward to the stage opening in a double angle to improve the distribution of first reflections. The ceiling is in timbers 5cm. thick and is shaped over the orchestra pit to disperse reflection of sound from it. Towards the rear it is shaped to diffuse reflections downward. The pit accommodates about 70 players. The hall proved to be very satisfactory.

In the concert hall the audience area is on a steep rake set around half of a boarded flat octagon, while a raked stage is set opposite, around the other half. The intention was that the octagon should provide distance between audience and major forces, while smaller groups could use the octagon itself. In the event the latter has been preferred even for larger forces and is visually compelling for use in this manner.

The walls are again of brickwork and the ceiling of the concrete planks. The audience and stage was intended to provide the absorption required for the intended reverberation. At the computed frequencies the predictions correspond accurately with measurements but immediately there are lengthenings which it is expected to bring back into line by adjustments. They appear to make the room over-sensitive at certain frequencies for full orchestral work, but conditions for smaller groups, soloists, organ and recordings appear to be good.
INTRODUCTION: The definition of the diffusivity coefficient /d/ according to Thiele is the following: at a given point of the sound field the input energy from the solid angle dΩ/rounddΩ/ is A(Ω)dΩ. In this case the average energy

\[ M = \frac{1}{\Theta} \int_{\Theta} A(\Omega) \, d\Omega \] (1), where \( \Theta = \int d\Omega \) the total solid angle or a certain part of it, in which the measurements are taken. The medium deviation from the average value is

\[ \Delta M = \frac{1}{\Theta} \int_{\Theta} [A(\Omega) - M] \, d\Omega \] (2). If in the examined room \( m = \frac{\Delta M}{M} \)

then in the free field /anechoic/ \( m_o = \frac{\Delta M_o}{M_o} \)

Consequently the diffusivity is \( d = 1 - \frac{m}{m_o} \) (3)

EXPERIMENTAL: A strongly directed receiver-equipment, a parabolic metal-mirror, with diameter 80 cm and with a focal point distance 13,1 cm, was used to measure value A(Ω). The condenser microphone was displayed in the focal point. The pulse-sound of pistol shot was used as a sound source. In each of the measuring points the linear-signals were recorded by a tape recorder with 31 different directions of parabolic mirror. The measuring data were prepared in the laboratory, using an octave filter.

Sound diffusivity measurements were realized in 3 rooms where some acoustic measurements had already been carried out /3/, and the more important room acoustic parameters were decided with new measurements. And besides some single criteria were calculated, available by the evaluation of the integrated values of short pulses.

At a few places of measurements the integrated signals of the directed microphone positions have been calculated in detail. These values were compared with the single criteria and with other room-acoustic parameters.

The results of diffusivity measurements /at 1000 Hz/:

- University Church  \( d = 0,75 \)
- Erkel Theatre  main floor  \( d = 0,28-0,43 \)
- balcony  \( d = 0,55-0,56 \)
- Large Hall of Music  main floor  \( d = 0,37-0,53 \)
- balcony  \( d = 0,69-0,73 \)

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THE EFFECT OF NON-UNIFORM DISTRIBUTION OF ABSORPTION ON REVERBERATION TIME

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INTRODUCTION

The statistical-geometrical theories of reverberation have received close experimental verification in situations where the absorbing material is distributed almost uniformly over the enclosure boundaries. However, when the above is not the case, the experimental observations have shown marked deviations from the theoretical predictions (1,2).

EXPERIMENTAL AND THEORETICAL OBSERVATIONS

The study of the effect of nonuniform distribution of absorption on reverberation time has been done experimentally in a small scale reverberation chamber measuring 15ft by 9ft by 8ft for frequencies between 1000 Hz to 4000 Hz at one-third octave bands. However, it was considered necessary to develop a theoretical technique which would render more general conclusions than are available from the limited experimental results. This has been done by simulating the reverberation process on a digital computer through the geometric principles without bringing in the simplifying assumptions for the statistical methods to apply. The simulation is based on the ray tracing method which assumes a point source of sound emitting a finite number of rays. The energy in each ray is assumed to travel along a straight line path until it strikes a boundary of the enclosure when a reflected ray is formed according to the Snell's law of reflection. The reflected ray is then re-reflected and this process continued until the energy content of all the rays fall below an insignificant level. The receiver is assumed to be spherical. No limitations have been placed on the shape and size of the enclosure so that the effect of shape could also be studied. The reverberation times (RT) obtained from the above digital model were far in excess of those obtained from the corresponding experimental situations. This was ascribed to the fact that the model, as such, does not take into account the diffusion of the sound field resulting from the diffraction effects at the edges and corners of the enclosure. The model was, therefore, modified to include the above effects by assuming that the rays, which strike an enclosure boundary at less than a certain distance D from a discontinuity, are completely diffused and hence decay in accordance with the Sabine-Eyring RT formula. It has been observed that when D is made equal to half the wavelength of the sound field the reverberation times obtained from the digital model are in good agreement with the experimental observations. The results obtained from the model are being analysed so that broad conclusions as to the effects of nonuniform distribution of absorption may be derived.

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ACOUSTIC SCALE MODEL EXPERIMENT USING MEDIUM OF
NITROGEN GAS

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INTRODUCTION

In scale model experiment, dry air is mostly used as medium in order
to simulate sound absorption by air. Dry air is not a mere
medium for scale model experiment, but nitrogen gas is also known as
a promising medium.

STUDY AND CONCLUSION

The frequency of sound applied to 1/n scale
model experiment should be n times as high as
in actual room. At such high frequency range,
oxygen gas in wet air absorbs much sound energy
(3), and it is necessary to reduce relative
humidity of air to simulate well the sound
absorption. If only nitrogen gas is used, it is
not necessary to control relative humidity and
temperature. Fig. 1 shows the relation among
the sound absorption coefficient of air m,
classical absorption $m_1$, and molecular absorp-
tion $m_2$ in full lines at 20°C and 60% R.H.,
where $m = m_1 + m_2$.

In 1/10 scale model experiment, sound absorp-
tion by air should be the dotted line, and this
curve is nearly equal to $m_1$. This means it is
necessary to make sound absorption comprise only
classical absorption $m_1$, namely to make $m_2$ value
zero. For this purpose well known method is to
use dry air, and another method is to use nitro-
gen gas, because it has classical absorption nearly equal to that of
air, molecular absorption nearly zero. To illustrate this fact,
reverberation time was measured in model echoic chamber filled with
either nitrogen gas or air at different temperatures (0.5°C~20°C) and
different humidities (12%~85%).
The following conclusions have been drawn.
1) In 1/10 scale model experiment, nitrogen gas does not cause
anomalous absorption like oxygen gas dispite under existance of water
vapour.
2) Physical constants such as sound velocity and characteristic
impedance of nitrogen gas are nearly equal to those of air.
3) Nitrogen gas is a suitable medium for scale model experiment in
acoustics, and tolerable limit of remaining oxygen gas is about 3%.

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SIMULATION OF SOUND PROPAGATION WITH BOUNDARY
AND SUBJECTIVE TEST

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INTRODUCTION

Based on the measured transfer function by pressure reflection \( W(\omega, r/r_o) \), \( r=(r, \theta, \phi) \) sound propagation over boundaries is simulated by the aid of computer and some subjective tests are performed.

TIMBRE OF REFLECTED SOUND

An example of the impulse response \( w(t, \theta) \) of an even boundary obtained from the measured \( W(\omega, \theta) \) is shown in Fig.1 as a parameter of the angle of incidence \( \theta \). Reverberation free speech signal \( s(t) \), \((0\leq t\leq 1.2\text{sec.})\), is convoluted with \( w(t, \theta) \) in time domain. The 11 subjects are requested to judge whether or not the same timbre between the signals \( s(t)*w(t, \theta=0) \) and \( s(t)*w(t, \theta) \) radiated from a loudspeaker in an anechoic chamber. When the boundary is a glass fiber or a perforated board, for example, more than 80% of them judged as different timbre for \( \theta \geq 60^\circ \) comparing with that of \( \theta=0^\circ \).

ECHO DISTURBANCE

Two loudspeakers are located at \( \pm 22.5^\circ, 2.41\text{m} \) in front of a subject. The subjects are informed which is the direct \( s(t) \) or reflected sound \( s(t)*w(t-t) \) in Stereo-System. In Mixed-System, the two signals are mixed before supplying the loudspeakers so as to feel frontal incidence. The 23 subjects trained judge whether or not the direct sound is disturbed by the echo.

When the boundaries is a rigid or a perforated board(with 200Hz resonance frequency \(|W|=0.4, \text{ and } |W|=0.5 \text{ at } 150 \text{ and } 250\text{Hz})\), for example, the test results are shown in Fig.2.

These results display that an important effect of the transfer function at the oblique incidence to boundaries on subjective evaluations particularly in room acoustics and noise propagation.

Fig.1 Impulse response of a glass fiber for various angle of incidence.

Fig.2 Echo disturbance as a function of delay time and a parameter of relative level of echo.

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Rigid.

Perforated board.
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THE ACOUSTICAL RECONSTRUCTION OF TEACHING STUDIOS AT THE HUNGARIAN ACADEMY OF MUSIC

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INTRODUCTION

The Hungarian Academy of Music was opened in 1907. In those days, students and professors were considerably fewer in number than they are today.

From the six reconstructed teaching studios, three and three are disposed to form two groups of adjoining rooms. One of these groups is on the second floor and the other on the third, above it.

Originally, in both groups, the intermediate studios belonged to the professor, and so acted as acoustical separation.

EXPERIMENTAL

Measurements indicated that the mean sound transmission loss (T.L.) of the partitions was only 35 dB instead of min. 50 dB. It was only in the third octave-bands with mid-frequencies of 400, 500 and 630 Hz that the value of 40 dB was reached. For increasing the T.L. between the teaching studios, there was only one possibility, namely, to add a resilient skin to the original heavy partition. Because of the need for increasing T.L. in a very wide spectrum, a double leaf resilient skin with only about 12 cm thickness was constructed. After effectuation, the noise coming through the partitions has practically disappeared. The mean T.L. now exceeds the value of 50 dB. With the inner leaf, the reverberation time was adjusted.

THEORETICAL ANALYSIS

The starting equation for calculating the improvement in T.L. was the one for point attachment of a simple resilient skin. Comparison with the measured values after effectuation gave interesting results.

REFERENCES

THE ACOUSTICS OF MUSEUMS

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INTRODUCTION

The acoustics of museums and similar buildings (e.g. art galleries) does not seem to have been given much attention until the recent past. Acoustical problems are present in existing buildings and there is little guidance available on the desirable acoustics for new or refurbished buildings.

DESIurable ACoustics

It is not the job of the acoustician to decide what the desirable acoustical environment should be in aesthetic terms. His role is to advise the director and architect of a museum how best they can achieve the acoustical environment they desire, if it is at all possible. Experience of existing museums is of limited value, as there appears to be a growing feeling (in the UK at least) that the traditional form of museum is unsuitable for present-day needs (1).

Nevertheless, there are three points which we think should be given special attention.

1. The avoidance of intrusive noise. This obvious requirement has been neglected frequently in the past. In old buildings this can be excused (e.g. very distracting traffic noise in the old Museum of Antiquities, Edinburgh) but it is very difficult to justify in more modern buildings. There is a well-known London art gallery, built in the 1950s, where the ventilation noise is extremely intrusive.

2. Good intelligibility. Increasingly, parties of school-children are given guided lectures in museum galleries. In some of these, the size of the group is limited by the ability of the guide to make himself intelligible to all in the group, in the highly reverberant environment found traditionally.

3. Problems with audio-visual aids. The increasing use of unattended audio-visual aids (such as tape/slide presentations, back-projection sound films) in an open-plan format in the galleries can lead to serious problems. Often, if the volume is set for comfortable listening by a few people at sparsely attended periods of time, it is inadequate against the background noise of a "full-house." Perhaps a form of automatic volume control governed by the prevailing background noise could assist here.

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EIGHTH INTERNATIONAL CONGRESS ON ACOUSTICS, LONDON 1974

ANALYSIS OF FACTORS RELATING TO ACOUSTIC QUALITY DERIVED FROM RECORDINGS MADE IN A MODEL MUSIC STUDIO

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INTRODUCTION

Acoustic modelling has been established as a technique where a model can be used to give sound quality very similar to that of a corresponding real studio (1). Variations in the acoustic treatment of the model studio have subsequently been tried out to see whether an improvement in sound quality was possible. The subjective appraisals consisted of an A-B-A-B comparison, A being the variant under test and B being the studio in a standard reference condition. The questionnaire had ten qualities listed as well as an overall assessment on a like/dislike axis. Scoring was done by marking a line whose extremities were defined by words appropriate to the quality. The centre position indicated "the same as" the reference.

FACTOR ANALYSIS

Preliminary tests gave disappointing results when analysed by factor analysis (principal components method). Improved presentation enabled eight observers to give highly consistent results. Factor analysis showed four orthogonal factors containing 82% of the total variance. Each factor includes all the original (ten) variables but the weightings may vary considerably for a given factor. Hopefully each factor will depend predominantly on only a few of the ten variables and the important variables will not be repeated in different factors. This desirable situation was found in the present studies: factor (F1) correlated highly with the overall assessment and consisted predominantly of the variables string tone, colouration, timbre, brilliance and definition. Factor F4 correlated significantly with overall assessment and had one predominant variable, fullness of tone. The other two factors F2 and F3 did not significantly correlate with overall assessment: the predominant variables were tonal warmth and hardness (F2), liveness and intimacy (F3).

CONCLUSION

Factor analysis has been used to determine two orthogonal factors which substantially account for the overall assessment of acoustic quality. It cannot be assumed, however, that the situation described is generally applicable to all acoustic environments: this investigation is limited by the reference condition (i.e. the particular studio used for this model), the music used in the tests and the qualities listed in the questionnaire.

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THE EFFECT ON ACOUSTIC QUALITY OF INCREASING THE HEIGHT OF A MODEL STUDIO

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INTRODUCTION

The effect on sound quality in a concert hall of early reflections has been commented on in the literature (1). A particular music studio used for broadcasting has a relatively low roof (average height to width ratio 1:2.5) and the first reflections came from the roof. Complaints have been made of the sound quality in this studio and the relatively low roof has been suggested to be the cause. Experiments with a one-eighth scale model of this studio have already been described (2-4) and advantage was taken of the existence of this model to investigate the effect of raising the roof in two stages, firstly the equivalent of eight feet, then by a total equivalent of sixteen feet. The average height of the real studio is only 26 ft, so the percentage increase in height is quite large. Two spaced omnidirectional microphones were used to pick up the sound and monitoring was carried out on two high quality loudspeakers.

EXPERIMENTAL

The first effect of raising the roof was an audible and acceptable increase in reverberation time above the designed value of 1.8 sec. The volume of the studio is 220K ft$^3$ so the conclusion is that a higher reverberation time than is normally accepted (5) is desirable. When, however, additional absorption was introduced to reduce the reverberation time to its previous figure the sound quality was identical with that of the original size and it was impossible to tell by ear which was the larger.

The model was then increased in height by the equivalent of another eight feet, making sixteen feet in all, and a similar result was obtained.

CONCLUSION

It is clear therefore that when listening to programme from a studio by means of a pair of spaced omnidirectional microphones that contrary to what might be expected from the literature the sound quality is not dependent on the height of the studio over quite a large range.

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APPLICATION OF COLOUR LIGHTS IN SOUND FIELD MAP
METHOD FOR BUILDING ACOUSTIC MODEL EXPERIMENTS

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INTRODUCTION

Model experiments are advantageous for the examination of
the sound field around buildings. The colour lights sound
field map method gives a continuous information on the
noise level.

EXPERIMENTAL

The experiments were performed on a 1:25 scale model in
the 5-20 kHz frequency range /8 and 16 kHz oktav bands/.
The field was free of reflections. A line radiation was
used for a sound source. For the mapping of the field a
1/4" microphone, a round of 5 coloured lamps with a 25 dB
potentiometer gave a resolution of 5 dB.
Measurements were done in 3 horizontal planes corres-
ponding to real heights of 1,75, 6,7 and 12 m. The isobars
were recorded photographically.

RESULTS

The map method gives a quick and
simple possibility for the evaluation of the sound field as com-
pared to the point by point measure-
ment. The results indicate that
- with a building parallel to the
road the noise level behind the
building is 10-20 dB less, than
that on the front side,
- the decrease of the noise level
is limited to the shadow area of
the building,
- between buildings perpendicular
to the road the noise level in-
creases as a consequence of the
reflections from the walls,
- passages perpendicular to the
road are disadvantageous.

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COMPUTATION OF SOUND RADIATION FROM LOUDSPEAKER
SYSTEMS WITH THE SYMMETRY OF THE PLATONIC SOLIDS

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INTRODUCTION

Sound sources built of small loudspeakers placed on the surface of regular solids are con-
sidered. These sources radiate isotropically at low frequencies, but at higher frequencies
the radiation depends on the direction. The sound intensity at these frequencies has been
calculated. These sources are used in reverberation rooms.

COMPUTATION METHODS

The solids are replaced by spheres, and the loudspeakers by point sources. With these
idealizations, the wave equation for the sound pressure is solved by the usual expansion
in spherical harmonics (1).

The theory of group representation is used to ease the calculation (2). Some invariant
spherical harmonics are given in (3), and one has been calculated for the present work.

In the lecture the calculation method will be described. The intensity in various directions
will be shown on drawings and discussed. The calculated intensity will be compared with
experimental results in the case of the dodecahedron.

INTENSITY VARIATIONS

The root mean square value of the variation of the intensity is given in the figure. The
mean is taken over all directions, and the intensity on the figure has been divided by the
mean intensity. The abscissa is proportional to the frequency.

![Diagram showing variation of intensity with wavenumber times radius of sphere]

REFERENCES

For the precise model experiments in room acoustics, the sound absorption in air and the absorption on the interior surfaces should both be simulated.

We have investigated nitrogen substitution method to satisfy the former condition, and for the latter, examined the simulation of the sound absorbing materials on 1:10 model.

The simulation was made such as a manner that every dimension of the tested materials was scaled down to 1/10 on the model, so that the reactance value in the normal impedance would remain unchanged, and the resonance frequency of the model material would be 10 times as high as that of the actual one.

We made 1/10 models of various kinds of absorbing materials such as freely suspended panel, resonator, perforated panel, rib wall and thin porous materials, and measured their absorption coefficient. A model reverberation chamber having 0.2 m$^3$ volume was used in measurements in which N$_2$ gas was filled in order to stabilize the sound absorption in the medium. As a result, it is made clear that almost all kinds of sound absorbing materials can be simulated as to their absorption characteristics on the model at 10 times higher frequencies.

Figures show two examples of the results.
Pour mettre en évidence de façon réaliste les différences entre réverbération naturelle et artificielle, on enregistre sur l'une des pistes d'un magnétophone stéréo, de très près, une séquence de bruit, de musique ou de parole (une phrase, ici). On tire le sonagramme, diagramme fréquence-temps hautement significatif (1).

Sur la deuxième piste on enregistre simultanément la même séquence, à une place normale d'audition dans une salle, et on tire de même le sonagramme(2).

On injecte alors l' enregistrement correspondant à (1) dans un dispositif de réverbération artificielle que l'on règle de façon à obtenir une durée de réverbération totale identique à celle de la salle normale (3).

On extrait ensuite, selon la méthode préconisée par Mr. E. JFPP (Revue d'Acoustique n°26, 1973) les points communs et les originalités des figures(2) et (3). On obtient ainsi l'information acoustique perdue par la réverbération artificielle(4) et l'information parasite qu'elle rajoute.

Le traitement peut être automatisé pour étudier, en général, l'acoustique d'une salle existante.
CAN STATIONARY SOUND FIELDS BE ASSUMED TO BE DIFFUSE FOR LABORATORY TRANSMISSION LOSS MEASUREMENTS?

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INTRODUCTION
In Round Robin tests of transmission loss (TL) measurements large deviations have been reported between different laboratories. This has made it motivated to develop an experimental procedure to investigate the importance of assuming diffuse sound fields.

As we do not know the TL-results at perfect diffusion such a procedure must be based on comparative measurements. First we will have to work with a quantity describing the degree of diffusion. Then the importance of observed differences in this quantity must be estimated by comparing the corresponding values of TL.

The procedure is restricted to looking upon the sound pressure level difference (D) as the investigation is concerning stationary processes only.

PROPER QUANTITY FOR MEASUREMENTAL GRADING OF DIFFUSION
In the literature a solution to this problem has been attempted by means of correlation analysis (e.g. 1). The normalized cross-correlation-function (\( \rho \)) of the sound pressure in two discrete spatial positions is studied. My own observations have shown that it does exist essential spatial unisotropy for \( \rho \) in normal reverberation chambers, especially at lower frequencies. Isotropy for \( \rho \) must be a necessary condition for a perfectly diffuse field. A new measure of the degree of diffusion (\( \varepsilon_d \)), which considers the spatial variance of \( \rho \) has been developed. The behaviour of \( \varepsilon_d \) in some well-known situations is controlled. E.g. experiments have been carried out in an anechoic chamber with different numbers of independent sound sources and sound wave directions. The results indicate that \( \varepsilon_d \) might be very useful in describing the degree of diffusion in different situations.

STATISTICAL SIGNIFICANCE OF D - ESTIMATE
Assuming that the mean square pressures are gamma distributed it has been possible to derive confidence statements for the estimates of D. The gamma assumption has been controlled by performing some goodness-of-fit tests.

FINAL COMMENT
The experimental procedure and the results of its usage will be fully documented in a thesis.

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EIGHTH INTERNATIONAL CONGRESS ON ACOUSTICS, LONDON 1974

SPECIFIC ACOUSTICAL FACTORS CONTROLLING SPATIAL CONCEPT OF ENCLOSURES

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INTRODUCTION

Spatial definition of an enclosure, meaning the awareness of the physical dimensions of that space, based on acoustical cues, in the absence of full visual perception of the boundaries, has been explored and users reactions noted. It is proposed that the "Acoustical Acceptable Usability" (AAU) of a sub-environment, created by partial visual barriers (screens) within a larger space, having a minimum area ratio of 1 to 15, used for speech communication, is dependent on Reverberation time and the &la of the boundary surfaces of that sub-environment.

EXPERIMENT

Field studies were carried out in a room used for open class tutorial learning situation with dimensions of 30m x 15m x 3m and having an &a of 0.13. At any given time, the room may be used by maximum of six groups with 6 to 15 students in each group. (slides 1-3). The investigation centered on the desirable visual and speech privacy and the degree of AAU needed for each group. Objective measurements performed in the room during the tutorials indicated an average SL of 43 to 46 dB(A). (slides 4-7). To provide speech privacy in open plan offices, several research workers (1,2) indicated the necessity of highly absorbent ceiling/floor or ceiling baffles or screens. AAU in the case study was to be provided by special absorbent mobile screens, forming sub-environments and/or electronic masking of 48 dB(A). Reverberation Chamber measurements of similar partial enclosure is shown. (slide 8) Subjective reactions of two groups of subjects (6-10) were recorded for the following conditions in sub-environment of 25m². (slides 9-11) Experiment 1: Screens with hard surfaces; 1a: As 1, plus masking. Experiment 2: Special absorbent screen; 2a: As 2, plus masking.

CONCLUSION

The summary of subjective reactions together with objective recordings confirms other workers findings(3): a shorter reverberation time in a room produces a less critical echo-delay difference, disturbance. It varies however, with findings described elsewhere, (4) that in the absence of ceiling reflections, screens will perform well, to provide acoustical privacy, but here, it is suggested that AAU be provided in certain conditions by special screens and local absorbers. (slides 12-15)

REFERENCES

For the identification of coincidence of reflections in the impulse response (IR) of a rectangular enclosure, the mathematical IR model calculated by a computer might be useful. The conception of the model is based on a system of image sources from which an infinitely short pulse is generated at the same instant. The arrival times of individual pulse waves are proportional to the distance of the image sources from the receiving point. Owing to the spherical divergence, the wave intensity is falling proportionally to the square of distance from the image source. Both the absorptive power of walls and the sound propagation attenuation might be included in the calculation. Besides the intensities and arrival times, the model also provides directions of sound rays arriving at the receiving point.

When judging the agreement of the measured and calculated IR, the most striking is the amplitude structure of reflections. The reason for the amplitude disproportion lies in the coincidence of reflections of the measured IR for which (in contrast to the calculated IR) the principle of purely energetical superposition is not valid.

The reproducibility of impulse measurements in enclosures is strongly influenced by the coincidence phenomenon. At repeated measurements, it is not difficult to replace both the source and receiver so that the time delay differences of corresponding reflections would not exceed 1 ms. This accuracy, from the point of view of the time structure of reflections, can be considered as sufficient. However, from the point of view of the coincidence reproducibility, at least two orders higher accuracy has to be required. The coincidence phenomenon can markedly effect the energy build up (integral of IR) in the initial time range wherein the rarely occurring high reflections exist. On the contrary, in the range of sufficiently great frequency of reflections, the coincidences can be mutually compensated.

Owing to the wave character of the acoustical field, it is principally impossible to suppress the coincidence phenomena occurring in IR. It could be possible to gain a higher degree of accuracy of the amplitude structure by averaging several IRs measured in the vicinity of the receiving point.
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SUBJECTIVE BEURTEILUNG DER RAUMAKUSTISCHEN EIGENSCHAFTEN VON KONZERTSÄLEN: ERFASSUNG DER URTEILE AND MEHRDIMENSIONALE AUSWERTUNG

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MULTIVARIATE ANALYSIS OF SUBJECTIVE MEASURES FOR SOUND IN ROOMS AND THE PHYSICAL VALUES OF ROOM ACOUSTICS

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INTRODUCTION
In order to obtain a relation between subjective measures for room sound effects and physical values, when time and space structures of early reflection sounds in an auditorium be varied, we conducted hearing tests for excellence of sound in the sound field simulated by a sound synthesizing system. In this paper, procedures and results of multi-variate analysis of the psychological scales obtained from the subjective assessments and of the physical values are presented.

HEARING TEST
For the original sound, string music played in an anechoic chamber was used. Duration was 10 seconds. Test sounds from the original were reproduced in pairs through a digital delay unit, attenuaters, and loudspeakers set in the anechoic chamber in accordance with test conditions. Fig.1 gives the arrangement of the loudspeakers. Regarding the test conditions, time-delay and the reproducing level of sounds from the loudspeakers were controlled, so that the number of reflection sounds within 85 ms of time delay was 2-8, and that of the conditions was 54. Other physical values at the listening point were as follows:
(a) Ratio of early to reverberant sound energy: 0.77-2.74,
(b) ratio of front to back sound energy: 3.03-5.04, and
(c) listening sound pressure level: 83.0-86.3 dB.
The hearing test was carried out on these 700 or so pairs of stimuli compiled at random, with two engineers selected as listeners.

ANALYSIS OF THE RESULTS
Data of the assessment were constructed in spaces from dimensions 1 to 5 by means of Kruskal's multidimensional scaling. However, the configuration had converged in 3 dimensions under the experimental conditions. Values corresponding to each configuration axis in 3 dimensional space were named as subjective measures I, II and III respectively, and the correlation between these and 10 physical values was studied.

In order to extract the physical values corresponding with 3 the subjective measures as independently as possible, the varimax method was used. That is, coordinates of the measures were rotated in a space of 3 degrees of freedom, and by multivariate analysis of the rotated measured and the physical values, structure vectors of each composite variable, which were orthogonal to each other, were calculated. As a result, high correlations were obtained for the following: (a) Subjective measure I; ratio of early to reverberant sound energy and level of reflection sounds from the front horizontal directions of the listener, (b) subjective measure II; listening sound pressure level and number of reflection sounds, and (c) subjective measure III; ratio of front to back sound energy.

Fig.1 Arrangement of sound field synthesizing loudspeakers
NOISE REDUCTION IN INDUSTRIAL HALLS
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INTRODUCTION

Sound-absorbing ceilings, and sometimes also walls, have more and more become a common measure of noise control in industry halls, especially in newbuildings.

It is very complicated to calculate the extent of this measure. Several authors have studied this problem in a theoretical way. (See ref. 1-10). In very few cases, however, also experimental measurements have been carried out. A serious problem by theoretical studies is to include the extent of the "furniture" (scattering and absorbing elements). And in practice it is indeed difficult to evaluate the height of the furniture and the average distance between the noise sources.

With respect to these factors we have used very simple physical-mathematical models as bases for our new formulas. The coefficients and factors in the formulas are however taken from a rather comprehensive series of experimental measurements in halls.

The aim was to get formulas useable for practical calculations.

NEW FORMULAS

\[ T, \text{ sek} = \text{Reverberation time at 500 - 2000 Hz} \]
\[ \Delta L, \text{ dBA} = \text{Sound level reduction/distance doubling. } \Delta L \text{ is assumed to be a straight line.} \]
\[ L_{\text{diff}}, \text{ dBA} = \text{General Sound level reduction due to the abs. treatment. A kind of "average" value in the hall in exception of spots close to the sound sources.} \]

\[ T = 1,2 + 0,28 \cdot h - 2a + K_T \]  \hspace{1cm} (1)
\[ \Delta L = 3 + 3a + K_\Delta \]  \hspace{1cm} (2)
\[ L_{\text{diff}} = 4,5\Delta - 4 + K_1 + K_2 \]  \hspace{1cm} (3)
\[ \Delta = \Delta_{\text{after}} - \Delta_{\text{before}} \]  \hspace{1cm} (4)

\[ h \text{ m} = \text{height of the room} \]
\[ a = \text{abs. fakor of the ceiling at 500 - 2000 Hz Osawver. measured by the reverberation-room method.} \]
\[ K_T, K_\Delta, K_1, K_2 \text{ = constants depending on furniture height, the distance between the noise sources and size of the room.} \]

REFERENCES

Eine Reihe raumakustischer Kriterien wird aus der Impulsantwort berechnet. Diese Impulsantwort ist von der Anordnung von Sender und Empfänger im Raum abhängig, so daß versucht werden kann, aus den unterschiedlichen Impulsantworten an verschiedenen Plätzen Einzahlkriterien so zu berechnen, daß dadurch subjektive Hörsamkeitsunterschiede dieser Plätze beschrieben werden.


Die Berechnung von Kriterien aus den Impulsantworten an 10 Plätzen der Berliner Philharmonie (Sendesignal: Gaußtöne 1000 Hz, 2 ms), wobei 7 verschiedene Sendepositionen auf dem Orchesterpodium gewählt wurden, ergab, daß Kriterienwerte an einem Platz bei verschiedenen Sendorten ebenso streuen wie bei festem Sendort an verschiedenen Plätzen. Weitere Messungen wurden bei sehr geringen Ortsveränderungen des Senders (5 - 40 cm), sowie mit mehreren gleichzeitig strahlenden Sendern durchgeführt. Um im Rahmen der Meßgenauigkeit reproduzierbare Kriterienwerte zu erhalten, muß der Sendort auf etwa 10 cm genau festgelegt werden.

Weiterhin wird über Untersuchungen berichtet, die zur Überprüfung der Sendortabhängigkeit bei breitbandiger Anregung durchgeführt wurden /2/.

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/2/ W. KRAAK, Hochfrequenz und Elektroakustik 65 (1957)
INTRODUCTION
Les locaux industriels et les bureaux pour le travail collectif sont normal-ment caractérisés par une grande extension, par une hauteur modeste du plafond et par un grand nombre de sources de bruit distribuées plus ou moins uniformément dans l'ambiance. Dans ces conditions, il n'est pas applicable, pour les calculs de projet d'insonorisation, l'hypothèse du champ sonore parfaitement diffus et, par conséquent, il n'est pas possible d'utiliser les équations de l'acoustique statistique.

CONSIDERATION THEORIQUES
Dans une salle ayant une extension assez grande et un plafond très absorbant placé à une faible hauteur par rapport aux autres dimensions, on peut schématiser le phénomène de la propagation de l'énergie rayonnée par une source sonore avec un modèle de propagation unidirectionnelle à divergence cylindrique dans un espace délimité par le plancher et par le plafond, c'est-à-dire borné par deux surfaces moyennement parallèles avec des coefficients de réflexion $r_1$ et $r_2$.

De ce modèle on a tiré une équation qui permet de calculer la réduction du bruit qu'on peut obtenir par le traitement d'insonorisation.
Cette équation est du type:

$$ L_x = L_{x_0} - [10 \log \frac{x}{x_0} + 20 \frac{\alpha m}{H} (x-x_0)]\ dB $$

où $H =$ hauteur du plafond
$$ \alpha m = (\gamma - \frac{r_1 + r_2}{2})$$

et donne le niveau sonore $L_x$ (dB) dans un point éloigné d'une distance $x (> x_0)$ de la source, $L_{x_0}$ étant le niveau sonore dans un point placé à une distance $x_0 = H$ de la même source. Ce dernier niveau peut être calculé en appliquant opportunément l'équation de Eyring.
Au cours de la conférence on illustrera quelques exemples d'applica- tion et les résultats obtenus avec des matériaux à haute absorption.
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COMPARISON OF CONTINUOUS AVERAGING AND DISCRETE AVERAGING OF SOUND PRESSURE LEVEL IN A REVERBERANT ROOM

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INTRODUCTION
In order to reduce the uncertainty in the measurement of sound pressure level it is necessary, among other precautions, to make measurements at several discrete positions in the room, or to average the spatial fluctuations by the use of a continuously moving microphone. Waterhouse and Lubman (1) demonstrated by considering the correlation coefficient for the mean squared pressure as a function of space, that the continuous average is less reliable than a discrete point average. Since the sound pressure in a reverberent field is a fluctuating function of time as well as space, and from other considerations, we dispute this conclusion.

COMPUTER CALCULATIONS
A computer simulation of the reverberent field was set up in terms of the natural modes of a room. The standard deviation of the sound pressure level was calculated for microphone traces in two and three dimensions, and compared with the standard deviation for well spaced discrete points. It was found that for a medium sized room, the continuous average using a three dimensional travelling microphone of radius 1 m is as precise as seven or eight discrete measurements at 100 Hz and this number will be greater at higher frequencies.

These results were confirmed by laboratory measurements.

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(2) E.A. Lindquist (Title: as above). Chalmers University of Technology, Dept. of Building Acoustics, report 73-15.
Pour évaluer l'intelligibilité de la parole dans une salle, les méthodes les plus élaborées se basent sur des échographies enregistrées. La détermination globale d'une énergie utile et d'une énergie nuisible permet de faire une estimation de l'intelligibilité (1). Nous avons calculé des échographies spatio-temporels (2), et cherché à obtenir, à l'aide d'un champ sonore synthétique une évaluation plus exacte de l'intelligibilité en effectuant des essais systématiques en chambre sourde sur les premiers échos qui suivent le son direct. Ces essais ont pour but l'étude de l'intégration auditive des échos. Ils permettent de définir un seuil délimitant l'énergie utile d'un écho, intégrée au son direct, et son énergie nuisible, responsable de l'effet psycho-acoustique d'écho, qui détériore l'intelligibilité.

1) On a effectué d'une part des mesures de seuil de perception pour le premier écho, et tracé des courbes de variation du seuil en fonction de son retard pour différentes situations de l'écho, repéré par ses coordonnées $\theta$ et $\varphi$, le son direct provenant de la direction $\theta = 0$, $\varphi = 0$, en face de l'auditeur. Des mesures analogues ont été faites lorsqu'on juxtapose au son direct un écho de niveau, de retard et de direction tels qu'il lui soit parfaitement intégré. On obtient ainsi des courbes de perception du deuxième écho. Suivant le même principe on a tracé des courbes de perception des échos jusqu'au septième.

2) D'autre part, on a effectué des mesures d'intelligibilité à l'aide de listes de logatomes enregistrées en chambre sourde et diffusées en faisant suivre le son direct d'échos variables en nombre, en direction, en intensité et en retard. A l'aide des courbes établies précédemment, on a pu déterminer l'énergie du son utile, celle du son nuisible, et en déduire une estimation de l'intelligibilité. Ces estimations ont été comparées aux mesures d'intelligibilité effectuées directement ; cette confrontation montre que l'on peut espérer une nette amélioration des prévisions d'intelligibilité, en utilisant cette technique.

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The rapid growth of the use of induction units with fixed terminal noise levels which cannot easily be attenuated has made the problem of predicting noise in open-plan offices more difficult. It is not possible to allow a safety margin of attenuation as is possible with ducted systems and consequently the accuracy of predictions must be better.

A comparison of various simplified methods of calculation has been made. Treatment of the office as a perfectly diffusing space, in which the conventional calculations are applicable, at one end of the scale, to treatment of the office as a duct in which the sound level from a single source falls off linearly with distance at the other end of the scale. Neither of the extreme methods fully explains the sound levels measured in practice but both can be applicable in certain situations.

An assessment of the various theoretical methods is made in the light of measured results, from units already installed in open-plan offices, and from synthesised noise played over a large number of loudspeakers to simulate air conditioning terminals.
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THE INFLUENCE OF FURNITURE AND BOUNDARY CONDITIONS
ON THE PROPAGATION OF SOUND IN A LANDSCAPED OFFICE

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INTRODUCTION

In a previous paper (1) the propagation behaviour of the Department of the Environment's office in Kew, was discussed. Some correlation between this behaviour and that predicted from a multi-image theoretical model was found. In this particular office, furniture, which did not interrupt the line of sight, appeared to have little effect on propagation behaviour.

THE HUCKNALL EXPERIMENT

A more detailed study has been completed in an empty office belonging to Rolls-Royce at Hucknall. This has floor and ceiling surfaces of relatively low absorption. Precision measurements of attenuation were carried out under the following conditions: (A) untreated office, (B) office furnished with 200 screens, (C) as B with 150 screens, (D) end walls treated, (E) all walls treated, (F) as E with part of the ceiling treated, (G) as E with the floor treated, (H) as E with part of the floor covered with carpet, (I) as F with the floor treated. In each case the treatment consisted of a 50mm layer of insulant.

CONCLUSIONS

The smoothed attenuation curves were resolved into deviations from free field behaviour. By comparing these deviations it is possible to assess the effect of a given treatment. Unlike Kew the screens in (B) had a surprisingly large effect. This effect was however confined to traverses which were not at right angles to either pair of walls. The reduction in screen density (C) was not detectable.

Wall treatment (E) tended to produce a reduction in deviations in the far field at and the higher frequencies but this was by no means as large as the effect produced by the screens in (B). Ceiling treatment (F) produced marginal improvements in near field performance but these were lost when the wall treatments were removed (I). The use of the carpet (H) was found to have a similar but smaller effect than wall treatment. When screens interrupted the line of sight a significant reduction in deviations occurred in all cases.

REFERENCES


The author wishes to thank the Building Research Establishment for their assistance in this project.
La réverbération dans les salles parallélépipédiques est étudiée en assimilant les 6 faces à 6 miroirs plans et en recherchant tous les rayons sonores qui vont d’un point source à un point réception par tous les chemins possibles. On détermine la position des sources images. Les sources d’ordre n sont en nombre égal à 4 n²+2 et se trouvent de part et d’autre de la surface d’un octaèdre (1, 3). Deux types de libre parcours moyen lm et l’m sont définis et 2 expressions approchées peuvent être calculées. La méthode des sources images permet de calculer l’intensité du son réverbéré et le temps de réverbération de la salle pour lequel deux formules sont proposées. (2). La validité de chacune d’elles est précisée en s’appuyant sur le tracé des rayons sonores dans un plan.

Plusieurs applications sont tirées de la nouvelle théorie :

1°) la détermination de la courbe de décroissance du son dans une salle et notamment dans une salle réverbérante dont une seule face est recouverte d’un matériau très absorbant

2°) une nouvelle manière de mesurer les coefficients d’absorption en salle réverbérante

3°) les corrections à apporter aux formules donnant l’isolement normalisé ou l’indice d’affaiblissement acoustique d’une paroi.

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Revue d’Acoustique 19, 107-113 (1972)
THE QUALITY OF THE ACOUSTICAL ENVIRONMENT IN OPEN
PLAN EDUCATIONAL FACILITIES

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Over the past decade the design of educational facilities in the
United States has departed significantly from the traditional approach
of placing equally sized, fully enclosed spaces along a common corri-
dor. During the past few years the majority of new schools across the
country have been of open design and the number of open plan facili-
ties being designed is still on the increase.

Various acoustical characteristics peculiar to open plan spaces
have been investigated for a number of years. These studies however
have dealt primarily with the aspect of communication, i.e., either
speech privacy or speech intelligibility. There has been little or no
study regarding the quality of the acoustical environment of these
spaces. Even less data are available on the different acoustical
characteristics of the various types of open plan spaces. For
example, the open plan design can have either fully open spaces, which
cannot be divided by a full height partition system, or modified open
spaces, which include the "Pod" or "Cluster" concept of locating
smaller semi-open spaces around a common open space.

To study this aspect acoustical data were collected in a number
of open plan schools throughout the San Francisco region and included
both elementary and high schools. One such study, completed for the
Building Systems Information Clearinghouse of the Educational Facili-
ties Laboratory (BSIC/EFL) of the Ford Foundation, involved obtaining
continuous, calibrated tape recordings of class noise during a typical
school day and at several locations within a high school open plan
space. Identical measurements were completed in the space before and
after the installation of a sound absorptive ceiling system. The
comparative data derived from this study provided significant informa-
tion, both objective and subjective, regarding the acoustical quality
of the space.

From the measured data obtained in a variety of school spaces
throughout the region a graph was derived which relates the number of
students per area (student density) in a class space to the average
noise level (in dBA) predicted for that space. The predicted noise
levels provided by the graph have been compared to measured levels in
actual open plan situations and are reasonably accurate for a large
number of space types and configurations.

Further analysis of the measured data was completed and the rela-
tionship of the statistical distribution of noise levels to the acous-
tical quality in open plan class spaces was investigated. The techni-
ques of this portion of the study are similar to those used in
exterior environmental surveys and the distribution curves indicate
another parameter in defining the acoustical quality of these spaces.
INTRODUCTION

There is still dissension, which reverberation time formula, the Sabine formula or the Eyring formula is to be preferred. Theoretical analysis and experimental verification indicates that it is improbable that the Eyring formula is the more accurate.

THEORETICAL ANALYSIS

Eyring follows a little "bit of sound energy" on its way through a room. By dividing the velocity of sound by the mean-free-path he finds the mean number of reflections per second and thus the mean energy loss per second. If he would have followed all "energy bits" in the room together, as he should have, he would have to consider that every bit undergoes its own number of reflections and, even more important, that those "bits" can't have an own identity. Whatsoever the distribution of the free paths around the mean free path and the relative part of the sound energy in every "path", the resulting reverberation formula would come nearer to the Sabine formula.

A general sound energy decay formula can be derived:

$$- \frac{d E_{\text{tot}}}{dt} = \iint_S p \bar{\nabla} \cdot \hat{n} \, dS$$

wherein $E_{\text{tot}}$, the total sound energy in the room, $p$ the sound pressure and $\bar{\nabla}$ the particle velocity at the boundary, $\hat{n}$ is the unit vector perpendicular on the part of the boundary $dS$. It is easy to see that in case the sound decay is exponential the sound energy density near and directed to the absorbing boundaries should, in order to get the Eyring reverberation time, be larger than to the non absorbing boundaries or in the middle of the room. The opposite assumption would however be more probable.

EXPERIMENTAL VERIFICATION

One may expect that the larger the room the shorter the reverberation times according to Eyring relative to those according to Sabine will be. In series of rooms of different dimensions but with boundaries out of the same materials this may be easily verified. This, however, has been found not to be the case.

It is possible to compare the sound levels in the decaying sound field at different places in a room. It has been found, to an amazing degree, that after a short time, roughly the time needed for the sound to traverse the room, the sound levels are and stay everywhere equal to each other in all kinds of simple rooms.
DIFFUSE SOUND FIELDS: MODAL AND FREE-WAVE MODELS

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INTRODUCTION

It is known that, when the usual procedures are followed in setting up a diffuse sound field in a rectangular reverberation chamber, interference patterns exist at the boundaries of the chamber. When a narrow frequency band is used to excite the field, the sound pressure level is significantly higher within the region extending to about λ/4 from the boundaries. Patterns of a similar nature hold for the kinetic energy density. Whereas these effects have been found experimentally, and have been accounted for theoretically on the basis of a free-wave model (1), they have not heretofore been shown to be consistent with a normal-mode analysis; it is the purpose of this paper to demonstrate the consistency of the two models.

ANALYSIS

We wish to show that when a normal mode, or a set of normal modes, are excited in a rectangular reverberation chamber, the potential energy density V(x,y,z) is not uniform throughout the volume of the chamber, but is greatest at the boundaries. We also wish to find the expressions for the space variation of the kinetic energy density T(x,y,z).

Starting with the case of an axial mode in which the field varies in one dimension only, the potential, kinetic, and total energy densities are calculated. V(x) has a maximum value on the boundary. Repeating the calculation for a tangential mode, in which the field varies in two dimensions, V(x,y,z) is found to have maximum values on the boundaries, and values varying between the maximum and zero in the interior region. For an oblique mode, in which the variation is three dimensional, the results are similar.

A table of the expressions for V(x,y,z) and T(x,y,z) for the three types of mode is given, together with some graphs.

DISCUSSION

Our model for the diffuse sound field in a rectangular reverberation chamber is that several overlapping oblique modes are excited together by a narrow band of noise. In such a case, taking the space-averaged value of V(x,y,z) as unity, the value in a corner is V(0,0,0) = 8, the value at an edge is V(0,0,z) = 4, and the value on a wall surface is V(0,y,z) = 2. The results agree exactly with those given by the free-wave analysis, as do those for the kinetic energy components.

We conclude that the free-wave and the normal-mode analyses are consistent.

REFERENCES

A new theoretical approach to the investigation of useful and disturbing sound reflections affecting speech intelligibility is described. The way of treating this problem can be explained by means of Fig. 1, where exponential laws for sound energy growth and decay were used. An assumption was made that the disturbing sound stems only from the preceding vowels, and that the consonants are the only phonemes contributing to the intelligibility. E, on Fig. 1 is the disturbing sound energy density, which depends on the vowel acoustic power, on the duration of the pronunciation of the vowel (\(t_v\)) and on the duration of the following consonant (\(t_c\)) including the preceding pause (if any). The useful energy density (\(E_u\)) depends on the consonant acoustic power and on the duration of the pronunciation of the consonant (\(t_c\)). Taking into account statistical distributions of the three important time intervals (\(t_v\), \(t_c\), and \(t_{cp}\)) and using average power relations in speech sounds, a weighting factor has been determined for all reflections, depending on their time delay (\(\Delta t\)) after the first direct sound. The sum of sound energies of all reflections, multiplied by this factor, yields the amounts of \(E_u\) and \(E_d\). The mentioned weighting factor, which is presented on Fig. 2, changes with the talking speed. Different measurements of intelligibility were carried out to confirm this theory: in normal rooms, with different tempos of speech; in electroacoustically coupled rooms; and in a case when the reflecting sound was provided by the reverberation plate, combined with single echoes and delayed reverberant sound. Experimental results are in good agreement with the theory.
THE INFLUENCE OF THE ACOUSTIC PROPERTIES OF ROOMS ON THE STRUCTURE OF SOUNDS

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INTRODUCTION

Acoustic signals propagating in enclosures are subject to many forms of deformation. These can be deformations of the frequency spectrum amplitudes and phase of the signal. The main aim of the experimental investigation was a quantitative analysis of these deformations and their relation to the acoustic properties of the enclosure.

EXPERIMENTAL

The quantitative analysis of the deformation of the acoustic signals in the enclosure was carried out using an FFT procedure to calculate the following quantities:-

- Power spectral density function,
- Cross spectral density function (consisting of coincidental spectral density function and quadrature spectral density function), and correlation function.

The experimental procedure included the following points:

(i) Estimation of the deformation of the amplitudes of the spectrum of the signal; also its deformation with the time and phase changes in relation to the nature of the acoustic field (free-field or reverberant).

(ii) Evaluation of these deformations with respect to changes in the acoustic parameters of the enclosure, particularly the reverberation time and frequency response.

(iii) Estimation of the above deformations, as affected by the spatial configurations of the interior when divided into acoustically coupled enclosures. The measuring apparatus, of which an essential part was a buffer memory, allowed the recording of a chosen section (in time) of the signal. Hence it was possible to carry out an analysis of the deformations described above for chosen stages of the growth, steady state, and decay of the signal under investigation.

CONCLUSIONS

Analysis of the experimental results indicated a relationship between the degree of deformation of the amplitudes of the frequency spectrum of the signal, the frequency response and the reverberation time of the enclosure. One conclusion is, for example, that with increasing time, the amplitudes of the low frequency components of the spectrum slightly increase whereas the amplitudes of the high frequency components decrease with respect to the spectrum of the original signal. The degree of these deformations has a different value for growth, steady state, or decay of the sound in the enclosure. There are indications, also, that a theoretical evaluation of the degree of deformation of the signal can be made on the basis of an analysis of the coherence function of partially correlated signals by treating the enclosure as a linear physical system.

*This work was undertaken at Chelsea College, University of London, U.K.
EIGHTH INTERNATIONAL CONGRESS ON ACOUSTICS, LONDON 1974

INFLUENCE OF SPHERICAL DIVERGENCE ON THE INITIAL PART OF SOUND DECAY IN ENCLOSURES

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Kyncl J

In order to describe the influence of spherical divergence upon the reverberation of sound energy in an enclosure, it is possible to introduce a time representing the limits of its predominance over the exponential decay:

\[ t_r = 0.145 T - t_0 \] (1)

where \( T \) is the reverberation time, \( t_0 = \frac{r_0}{c} \) where \( r_0 \) is the distance of the source and \( c \) is the velocity of sound propagation. The time \( t_r \) is the instant at which the slope of the divergence and of the exponential components equals. The sound decay (or build up) in the intervals \( t_0 \leq t \leq t_r \) is mastered by the divergence component, for \( t > t_r \) by the exponential one.

For practical cases having \( t_r \) ranging from 50 to 250 ms, the divergence of sound waves is of major importance for the initial part of the sound decay. Hence, to define the initial reverberation time in the time interval 0 - 160 ms is questionable. Even the level range from 0 to 10 dB proposed for the early decay time by JORDAN is in principle equivalent to the range of the predominant influence of spherical divergence. Taking into account the usually complicated structure of the initial part of decay, it is a question whether to express the character of this part by the reverberation time of the equivalent linearized decay.

Other criteria are influenced by the divergence as well. Mostly, the limits - up to which the "useful" energy component of the impulse response is integrated - are smaller than \( t_r \) and so it stands to reason that the intensity of the first reflections as well as the amount of direct sound in the initial energy are notably influenced by the divergence. In more complex room shapes, the diffraction of sound waves reflected from curved surfaces of the room might cause - even in later phases of the decay - parts influenced by the divergence.
NOISE ABATEMENT INSIDE AND OUTSIDE BIG INDUSTRIES

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SUMMARY

Noise abatement in big industries will be dealt with. A planned power station has been used as an example. Most of the work of the acoustical consultant can be done in the development stage of the project.

After fixing criteria indoors and outdoors and after stock-taking noise sources in order to obtain a prognosis of the sound pressure levels one has to prescribe dimensions of extra provisions, for instance of an architectural nature or those in connection with technical installations. In this stage a renewed prognosis of the sound pressure levels both inside and outside the power station must be made. If the criteria have been realized one can give all the detailed architectural and technical advices.

The assistance in the execution of the advices and the checking by means of acoustical measurements have to be done during and after the execution phase. The project will be concluded by a final judgement.
EIGHTH INTERNATIONAL CONGRESS ON ACOUSTICS, LONDON 1974

FIELD STUDIES OF VIBRATION AND AIRBORNE SOUND AT LOW FREQUENCIES

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INTRODUCTION

Special problems concerned with airborne noise in the two octaves below 63 Hz frequently arise in noise control practice.

Factors contributing to difficulty are: absence of laboratory and field data about sound levels, sound insulation and absorption at these frequencies, and the unreliability of theoretical extrapolation: presence of fundamental structural resonances and the change from mass to stiffness control: very long wavelengths in air and in building structures which lead to large scale interference effects and to the failure of the airborne sound field concept on which many measurements depend: low propagation losses: the existence of high-level sound sources within and external to buildings.

EXAMPLE

One of the cases studied by airborne and vibration measurements in considerable depth illustrates some of the problems.

Fig.1 shows airborne sound levels in a new air-conditioning plant room and on the floor below. Although sound levels in the higher octaves were consistent with available information about the fan sound power level, the peak in the 32 Hz octave was not predicted. It appeared only in certain positions in the plant room but was generally detectable as a high sound level on the floor below (where estimates based on sound insulation extrapolation were necessarily approximate) and in the wall surfaces on that floor, reappearing as a high airborne level several floors below.

The plant room airborne level did not respond to appropriate remedial modifications but the expected reduction did take place in the floors below, vibration measurements on the radiating surfaces of the machine and on the structure being consistent with the observed reduction.

OTHER TYPICAL EXAMPLES

Other problems studied in detail in this frequency region include traffic noise transmission through windows, and long distance vibration effects on houses due to airborne propagation from heavy compressors.
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ON ACOUSTICS, LONDON 1974

EARLY DECAY TIME AND PHASE IRREGULARITY

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The frequency domain description of the acoustic behaviour of a room was adopted in the classic theoretical work of Lord Rayleigh and by a number of theoreticians in more recent years, notably Morse and his co-workers. Experimental work in the frequency domain utilised "averaging" concepts such as "frequency irregularity" and "phase shift rate". Schroeder has shown that relationships between these quantities and reverberation time exist, but the analysis assumes complete diffusion in the decay and yields approximate relationships only.

The complex system function describing the signal at a receiver generated by a source of fixed amplitude and pulsatance \( \omega \) is given by:

\[
H(j\omega) = \int_{-\infty}^{\infty} h(t) \exp(-j\omega t) \, dt
\]

where \( h(t) = \sum a_n \delta(t-t_n) \) is the impulse response of the room and consists of a series of reflections with amplitude \( a_n \) and time delay \( t_n \).

The real part of the system response may be written:

\[
Re(H(j\omega)) = a_1 \cos \phi = a_1 \cos \omega + a_2 \cos 2\omega + \cdots + a_n \cos n\omega + \cdots
\]

where \( a_\tau \) = amplitude and \( \phi \) = phase between source and receiver.

We may thus regard the system response of the transmission path as a fluctuation with respect to frequency (rather than time) which has a "power spectrum" \( S(\omega) = \sum a_\tau^2 \delta(\omega - \omega_n) \) equivalent to the square of the impulse response of the room.

We may obtain a measure of the phase irregularity in terms of the zero-crossing density of the fluctuation, \( \lambda \). If the fluctuation is well-behaved, this is given by

\[
\lambda^2 = \frac{1}{\pi^2} \frac{\int_{-\infty}^{\infty} t^2 S(t) \, dt}{\int_{-\infty}^{\infty} S(t) \, dt}
\]

Since a zero crossing in the system response of the transmission path represents a phase coincidence between source and receiver, we see that there is a close relationship between the rate of occurrence of phase coincidences and the second moment of the impulse response of the room. If the impulse response is assumed to be a simple exponential decay curve with reverberation time \( T_{60} \), the Schroeder relationship for rate of phase-shift with frequency:

\[
\frac{\Phi_{60}}{1} = 0.64 \cdot T_{60} \text{ radians } \cdot \text{Hz}^{-1}
\]

is obtained as a special case.
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NOISE CLIMATE IN LANDSCAPED OFFICES

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INTRODUCTION

An investigation carried out in large, but not landscaped, offices (1) and (2) indicated that dissatisfaction with noise could be related to average level in dBA and the number of sounds rising substantially above background level per minute (peak-index). A similar investigation (3) carried out in landscaped offices indicated that it may be possible to relate dissatisfaction to a function of \((L_{10} - L_{90})\) dBA.

TEN OFFICE SURVEY

A survey of ten landscaped offices is being carried out to find a "unit" for office noise which can be accurately related to subjective dissatisfaction. 10% of the occupants of each office are interviewed in depth and all the occupants are asked to fill out a questionnaire. Following the questionnaire, tape recordings of noise are taken over a two or three day period at ten locations, a number of these corresponding to the in-depth interview positions. A large number of measurement positions is essential owing to the variation in noise characteristics from point to point.

SOURCE LOCATION BY OBSERVER

In order to identify the location, nature and duration of a given noise source a number of trained observers record commentaries of the detailed activities of the office while the above noise recordings are being made. This commentary is then turned into digital form and used together with a small computer to aid analysis of the noise recordings. By suitable programming it is possible to obtain dBA \(L_{10}/L_{90}\) values which can be weighted according to source location, category or duration or any combination of these so that some agreement with the subjective assessment can be achieved. The computer eliminates any noise made by the occupant of the station where a microphone is located, so that measurements do not have to be made at empty stations, where the noise field may differ considerably from that at interview positions.

REFERENCES

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THE IMPACT INSULATION OF COVERED CONCRETE FLOORS

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INTRODUCTION

The ISO Tapping Machine is criticised because the frequency content is wrong and the levels are too high when compared with real footsteps. This paper is concerned with measurements of the impulsive forces associated with blows from the tapping machine and from footsteps and the correlation of the force spectra with noise spectra created in a reverberant room beneath a concrete floor.

PRINCIPLE

Although the floor is subjected to a series of impacts it is possible to work in terms of equivalent levels by averaging over long time intervals. This is already done when impact noise levels are measured in a room, partly by the room acoustics and partly by the meter damping. The force spectrum of a hammer blow has been measured by placing an accelerometer inside the hammer and then Fourier transforming the force (mass x acceleration)-history. The cushioning effect of a floor covering is included in this spectrum because it directly influences the impact. The response and radiation of the concrete slab is lumped into a single term referred to as the floor transfer function, defined as the radiated noise spectrum divided by the force spectrum. It is hypothesised that the transfer function is linear and constant for any given concrete floor irrespective of the nature of the force inputs. The only difference with footsteps is that the force-history has to be measured directly by walking on calibrated piezo-electric discs let into the floor.

RESULTS

The transfer function of a concrete floor has been measured with the tapping machine operating on the bare surface. Comparisons have been made between measured noise spectra and spectra predicted from the appropriate force spectra for hammer blows on a range of floor coverings. The excellent agreement has already been reported (1). Results are now presented of measured and predicted spectra for footsteps. The floor transfer function is unaltered but the noise levels are lower and low frequencies are emphasised. Indeed, the noise from footsteps on good carpet is difficult to measure at all.

REFERENCES

INTRODUCTION

Influences of impact machines to evaluation of impact sound insulation of floor are discussed, by using ISC tapping machine and the tire.

THEORETICAL CONSIDERATIONS

Fig.1 gives a typical model of impact source and floating floor. Capital M means effective mass, K;stiffness, R;resistance. Suffix t means tapping machine, c;covering, f;floating construction, s;slab, and fs;floating construction directly to beam.

Its make another evaluation to the same floor, that in case of $M_t (M_f M_s$ and $K_t K_c$), as the tapping machine, impact sound level attenuation almost depends upon $K_t / K_c$ and $P/F_n$ (normal frequencies of floating construction), on the other hand, $M_f M_s$ and $K_c (K_f$, as the tire, it depends upon $M_s / M_t$ and $K_s / K_t$, and that generally, floor construction has a lot of nonlinearity.

Because of these facts, two types of impact source are required, such as, one corresponds to shoe and the other to hopping up and down without shoes.

EXPERIMENTAL

Fig.2 gives impact force characteristics of the tire(600:12, 4ply, about 10.4kg with wheel), about various kinds of pressure and falling height (1).

Fig.3, upper two lines are impact sound level difference between bare 120mm flat concrete slab and with 100mm ALC floated by 40mm:64kg/m³ glass wool, and lower two lines are that of same weight bare slab with rib (2).

It gives typical characteristics of the tapping machine and of the tire.

REFERENCES

VERGLEICHENDE MESSENGEN DES TRITTSCHALLSCHUTZES VON GEHBELÄGEN UND SCHWIMMENDEN ESTRICHEN MIT VERSCHIEDENEN METHODEN

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Hammerwerk mit jeweils 5 Hämern mit 2 cm Durchmesser mit 10 Hz Schlagfrequenz mit

<table>
<thead>
<tr>
<th>Hammermasse</th>
<th>Aufschlagfläche</th>
<th>Fallhöhe</th>
<th>Impuls</th>
</tr>
</thead>
<tbody>
<tr>
<td>0,5 kg</td>
<td>Stahl, Gummi</td>
<td>2,5 cm</td>
<td>35000 g cm/sec</td>
</tr>
<tr>
<td>4</td>
<td>44300</td>
<td></td>
<td></td>
</tr>
<tr>
<td>1,0</td>
<td>&quot;</td>
<td>2</td>
<td>62500</td>
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<tr>
<td>1</td>
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</tr>
<tr>
<td>0,2</td>
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<td>2,5</td>
<td>14000</td>
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</tbody>
</table>

Standardsessel gleichmäßig über den Fußboden geschoben 2 Versuchspersonen auf- und abgehend, die in Vorversuchen aus 11 Versuchspersonen ausgewählt wurden, da ihr Gehen auch unter hochwertigen Fußböden noch messbar war.

Ferner in Vorversuchen
Hammerwerk mit 5 Hämern mit je 0,5 kg mit Aufschlagfläche aus Leder und Holz mit 4 und 2,5 cm Fallhöhe
Einzelhammer mit 10 kg Masse mit 5 cm Fallhöhe und mit 5 kg Masse mit 1 cm Fallhöhe jeweils mit Frequenz 2 Hz und Aufschlagfläche aus Stahl mit 6 cm Durchmesser.


Die Versuche mit Gehlen zeigten, daß beim Gehlen eine Federung wirksam ist, die sich zur Federung des Belages addiert und deren Einfluß bei Fußböden verschiedener dynamischer Steifigkeit verschieden ist.

Die beste Korrelation zwischen Beanspruchung mit Gehlen und Sesselrücken einerseits und Hammerwerk andererseits ergab sich für das Normhammerwerk mit 5 Hämern mit 0,5 kg Masse mit 4 cm Fallhöhe und Gummiunterflächen. Der Übergang zur A-Bewertung erscheint zweckmäßig.
INTRODUCTION

The simplified theory of floating floors is very attractive for floating floor designers. Very few parameters are involved, and the dynamic modulus of elasticity is one of them. However, discrepancies sometimes occur between the theory and the measured impact-sound insulation of the floor. Keeping to the simplified theory as close as possible and studying the measured data, we propose a new presentation of dynamic modulus values which would lessen the discrepancies.

REASONS AND USE OF PRACTICAL VALUES

The impact-sound insulation of 25 floating floors have been measured. In most cases, the slope above resonance was 12 dB/oct which agreed with the theory, but the rest of cases shew 9 dB/oct, 6 dB/oct or even less. The slope was determined as the best statistical fit rounded to 12, 9, 6 or 4.5 dB/oct corresponding to the powers of \( \omega \) of 2, 3/2, 1 and 3/4 in the attenuation formula. The frequency dependence is not unique and should be counted for. We suggest therefore that two values represent the dynamic modulus in designing floating floors: one, \( E_d \), determining the resonance \( f_0 \) according to eqn. (1):

\[
f_0 = \left( \frac{E_d}{M.d} \right)^{0.5}/500
\]

\( (M,d \text{ layer surface mass and thickness}) \), and the second, \( E_s \), determining the slope above resonance. Thus, for rubber plate made of rubber slivers,

\( E_d = 4.7 \text{ kp/cm}^2 \) and \( E_s = 6 \text{ dB/oct} \);

for ground cork glued to waterproof paper,

\( E_d = 13.8 \text{ kp/cm}^2 \) and \( E_s = 12 \text{ dB/oct} \).

In designing a floating floor, to get the \( \Delta L \) of the floor, first the \( f_0 \) is determined and then the slope is drawn.

![Graphs showing impact-sound insulation](image-url)

Fig. 1 \( \Delta L \) for floating floors with rubber sliver plates (a), and glued cork granules (b).
IMPROVED TRANSMISSION LOSS FOR FLOORS

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Fry A T

The desirable need for flexible building design layouts has often led to the requirement for large transmission loss values from floors. Conventionally this is achieved by the use of thick concrete which is expensive in the total building costs more especially as a result of the more massive arrangements necessary to support its weight.

To help combat this weight increase, and the poor returns of "5dB per DOUBLING of MASS", double floors have been developed in which the upper skin is resiliently supported on the structural floor slab at a resonant frequency below 20Hz. This results in improvements to the transmission loss as illustrated below.

The paper presented will discuss the acoustic engineering of such a composite floor construction and the installation details required for a successfully functional noise barrier.

The problems of variable loads on such a resilient floor slab will be discussed and the pertinent properties of springs, rubber and high density glass fibre will be compared, together with other engineering details.
INTRODUCTION

When the leaves of double-leaf structure of partitions and floors are entirely separated, their mass may be lowered to 36 kg/m² in comparison with 375 kg/m² demanded for them if they form a simple element, with no inferior influence upon their field sound transmission loss (1).

Fig. 1 Vertical sections of dwelling units placed one above another

EXPERIMENTAL

A prototype house was prefabricated of dwelling units of shaped sheet-steel with anti-vibration & absorption layers. The units were connected only through isolation & distance sleepers, making of their walls and floors a genuine system of double elements the mass of which does not exceed 80 kg/m²; the STL-curve runs 5 dB above the ISO/R 717-1968 reference contour.

REFERENCES

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PREDICTION OF IMPACT NOISE INSULATION FOR ORTHOTROPIC JOISTS

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INTRODUCTION

The prediction of the impact noise insulation of a joist is in many cases a heavy task. For massive plates of concrete etc. we have a fairly well known relationship between thickness and impact noise level. For some commonly used orthotropic joists we also know, by experience their degree of impact noise insulation. For a new construction, however, we can only wait and see what the result will be in a completed building since methods of calculating the impact response for non-homogenous or complex structures are complicated as well as inexact.

The effect of a floor layer is often also a matter of trial and error. In (1) we have described a simple method, further developed from (2), for predetermination of the impact noise improvement achieved with soft carpets. Only a small sample of the carpet is required.

RECENT WORK

Granted by the Swedish Council for Building Research we have developed equipment for model measurements on orthotropic joists. The results from these measurements are compared to full scale data from impact noise measurements, giving a base for the prediction of impact noise insulation through scaled tests. To the result for a joist can be added the impact noise improvement curve achieved by the simplified method described in (1), giving the complete predicted impact noise level.

EXPERIMENTS

A model reverberation chamber, scaled 1:4, has been built of 10 cm lightweight concrete, with a hard painted surface on the inside. Flanking transmission has been examined for different mountings of the scaled joists, which are made of gypsum.

A model tapping machine has been constructed, also scaled 1:4, as close as possible to the Brüel & Kjaer Standard ISO Tapping Machine, to ensure a true relationship between model and full scale measurements.

The result from a series of measurements on models of orthotropic joists is forming the base for an "impact noise catalogue", which will be helpful in the acoustical design of new buildings.

REFERENCES


The number of people living in apartment houses constructed with concrete has recently been sharply increasing in this country. A survey of inhabitants in such houses revealed that the impact sound of the upper floor constitutes the principal source of internal noise. Research and development regarding the improvement of floor construction are underway at present in various fields. Along with this recent trend, and also for performance inspection and quality control of buildings at field, it has become necessary to establish the standard of the method for measurement of the impact sound insulation of the floor for obtaining reliable data over a wide range with a unified method of measurement. The JIS method has been established to meet this social requirement. As the impact sound source, a tapping machine under ISO standard is required to be used. However, with such a light impact as one obtained by using a tapping machine, the relative quality of the building structure that has an important relation with the impact sound insulation of the floor in the low frequency range will not become apparent. For this reason, along with the measurement by a tapping machine, the impact sound level has been measured by utilizing the impact of repeated droppings of an automobile tire weighing about 10 kg from a height of about 60 cm. The development of a heavy floating floor construction having a good insulating characteristic to a heavy impact such as caused by a jumping child that often causes the problem of noise in apartments is underway. Consequently, the establishment of JIS method of impact sound level measurement using the tapping machine alone at the moment will not present much difficulty in adding a method of measurement using a heavy impact source at the time of the next revision of the present JIS method.

The method of measurement concerned mainly covers apartment houses. For this reason, points of measurement for both sound source and sound receiving positions are to be selected so that they are equally distributed in the room in order to obtain measurements that reflect faithfully the real conditions that the inhabitants actually feel in their daily life. For causing the effects of selection of the points of measurement, comparison measurement was performed twice. From the results of comparison measurement, we concluded that 5 points were sufficient for the sound receiving points, but it was desirable to have 5 points for the impact sound source positions, and for performing measurements with 3 points that were the least number required by the JIS method, the recommended positions should be placed at 3 points evenly distributed along the diagonal line of the room. For obtaining the mean value of the various source positions, the arithmetic mean in dB shall be made even when the difference between the mean values of the various sound source positions exceeds 10 dB, because we considered that the inhabitants would feel the average value of the impact sound level in their life.
It is well-known that the results of the measurement of absorption coefficients in difference reverberation rooms are subject to a relatively wide variation. This has been found to be particularly true of wideband absorbers with significant thickness. However, the shape of the graph of absorption coefficient against frequency is shown to be quite consistent in the various rooms tested with essentially only a shift in position in the direction of the y-axis - except at low and high frequencies.

Reasons for these discrepancies are suggested and corrections made which reduce the variation or account for it. The corrections made include those for humidity, volume and surface area changes when sample is introduced into the room, and for use of the Eyring instead of the Sabine equation.

The reverberation measurements are also compared with those made in TV studios where in excess of 90% of the absorption is due to similar wideband absorbers at most frequencies.
INTRODUCTION AND ADVANTAGE OF PULSE METHOD

Single pulses have been successfully used by the author in carrying different acoustic measurements instead of the classical sinusoidal waveform (1-3).

Fourteen years ago when the method was first developed (1) I used a mechanical fourier analyzer and the time of analysis was relatively long. With modern electronic fourier analyzers the time of analysis is only few seconds. Measuring sound absorption coefficient in a tube using sinusoidal excitation takes about half an hour compared with few seconds when the pulse method is used.

![Graph: Sound absorption coefficient vs. frequency](image)

**Fig. 1** Sound absorption coefficient

EXPERIMENTAL

Using sinusoidal excitation the absorption coefficient in a tube is measured in the classical way and its values are given by the curve in Fig.1.

In the tube a single acoustic pulse is produced and travels towards the absorbing material where it is reflected. Both reflected and incident pulses are analysed. The quotient of their respective fourier-components amplitudes yields the reflection factor \( r \). The absorption coefficient \( a \) is: \( a = 1 - r^2 \). The values thus obtained are shown in Fig.1 as small circles.

The values obtained by the two methods are fairly close except at low frequencies. This has already noticed in previous applications of the two methods (2,3).

REFERENCES

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(2,3) M. Louden, Acustica (1971) 25 167; (1972) 29 60.
A previous paper (1) showed how much results of sound absorption coefficient measurements can deviate from one another when they are carried out at the same object in different reverberation rooms, all complying with ISO/1354. Now the fluctuations are investigated to which the results are subjected already within a single reverberation room.

The absorption coefficient depends not only on the location of the receiver with respect to which the results are usually averaged, but also on the location of the sound source. Very often this fact is not given the necessary importance in practical measurements. Fig. 1 compares the 95% -confidence intervals of measurements at two perforated panel absorbers with an air space of 40 cm behind them, corresponding e.g. to a suspended ceiling, showing high or low absorption resp. in the range of the lowest frequencies. The confidence intervals correspond to averages of 4 different locations of the source with 7 receiver locations coordinated with each of them. As long as high absorption is observed in the region below 515 Hz, the resulting uncertainty range is rather wide. The mentioned frequency corresponds exactly to the Schroeder/Reichov condition for a sufficiently high density of eigenfrequencies. For an absorber consisting of mineral wool of 24 cm thickness, very similar values are obtained as for the perforated panel absorber showing high absorption. Evidently in spite of the presence of (fixed) diffusing elements, highly absorbing constructions are of most unfavourable influence on the diffusion in the low frequency range which in this range is poor anyhow, so that the uncertainty in a result may easily reach 15 to 30 % of its value. The use of several sources sounding simultaneously will mask but not eliminate this effect. Moreover it seems to be unessential in such an array whether the sources are driven coherently or incoherently. Uncertainties due to the build-up of the absorber are found to be within the given confidence limits. Thus for a revision of ISO/R 354 the question gains special importance whether perhaps the uncertainty could be diminished significantly by using very large rotating sound diffusing systems (2) and whether such systems should be demanded mandatorily.

REFERENCES

PROGRESS IN THE SOUND POWER MEASUREMENT IN REVERBERATION ROOM AT DISCRETE FREQUENCIES

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The determination of sound power emitted by sound sources radiating broad-band spectra can be easily achieved in a reverberation chamber with sufficient accuracy. When the sound source radiates single frequency or contains strong discrete frequency components, the precision is limited by the standing wave pattern.

Paper summarizes the results of research on this subject and reports on conditions to be satisfied in order to achieve sufficient precision.

When determining the precision of the sound power measurement two major effects have to be considered: the variation of the sound pressure with location and the variation of sound power radiated by the source with frequency and source position.

In order to guarantee sufficient precision ISO standardized a qualification procedure for reverberation chambers. In order to satisfy the qualification criteria, the variance of both the spatial averaging of sound pressure squared and sound power radiation as a function of frequency and position have to be kept very low. As it had been found by measurements in existing reverberation chambers with no additional treatment the criteria for qualification can not be satisfied. In order to achieve the qualification the reverberation chamber has to be equipped with rotating diffuser and also sufficient modal damping has to be established.

Rotating diffusers modulate the amplitude of the sound field at each point and also due to the Doppler effect the sound field is also frequency modulated.

As the theory and results of measurement presented in the paper indicate, the modulation effect of the vane reduces both the variance of the spatial distribution of the sound pressure and the variance of the radiated sound power variations as a function of frequency. The application of the rotating diffusers is most efficient at higher frequencies, where the diffuser is large in terms of the wavelength.

The wall absorption of the chamber effects the bandwidth of modes and their overlap and has considerable effect of the sound field at low frequencies. Combined effect of the absorption and modulation is discussed in paper. Finally practical data for the chamber qualification are presented.
A NEW ANECOIC FACILITY FOR SUPERSONIC HOT JET NOISE RESEARCH AT LOCKHEED-GEORGIA

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Dean P D  Tanna H K

INTRODUCTION

Much confusion about jet noise has come about as a direct result of inadequate facilities and insufficient knowledge and control of test conditions, in many cases giving rise to completely erroneous conclusions. The facility described here has been carefully designed accounting for other facilities' shortcomings and being guided by the stringent demands of on-going jet noise research at Lockheed. The design goal was the capability of testing model jets up to 2000°F at pressure ratios as high as 8, in a free-field environment anechoic at all frequencies above 200 Hz.

MODEL TESTS

A comprehensive series of flow visualization and temperature mapping experiments in a one-sixth scale model anechoic room was conducted. The results dictated the design of the exhaust collector/muffler to provide entrainment and room cooling air in the quantities demanded by the jet operating conditions. In order to optimize the choice of material and anechoic wedge design to achieve the 200 Hz requirement, a special impedance tube was used extensively in performance evaluations.

FEATURES OF THE FACILITY

Some features of the facility are (i) an exhaust collector providing air in quantities dictated by the particular jet operating condition with no special forced-air injection or fan installation and (ii) a "cherry-picker" crane used to gain access to instrumentation, etc. for maintenance, calibration and set-up, thus eliminating the need for access platforms. The crane is stowed by remote control under an anechoic cover during all test operations.

PERFORMANCE EVALUATION

The facility was subjected to a rigorous series of evaluation tests to be certain that tests performed in the room would be jet noise dominated. These tests brought to light the importance of microphone mounting arrangements. The superior performance of the room is described by intensity/distance plots obtained from both a fixed loudspeaker source and operation of a cold subsonic jet utilizing a moving microphone installation. These plots were obtained along several different paths within the room. In addition, a series of tests were performed to establish that upstream noise sources, such as valve noise and combustion noise, would not affect jet noise measurements to be made even at jet velocities as low as 300-400 fps.
GENAUIGKEITSVERBESSERUNG BEI DER IMPEDANZMESSUNG

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Wir tasten das Schallfeld im Kund'schen Rohr an (ca. 20) äquidistanten Meßpunkten ab. Die Pegelablesung erfolgt digital. In einer Ausgleichsrechnung werden von dem rechnerischen Feldverlauf

\[ P(x) = A \left( \exp k_0 x + R \exp -k_0 x \right) \]

Betrag und Phase des Reflexionsfaktors \( R \) nach der Methode der kleinsten Fehlerquadrate bestimmt.

Mit folgenden experimentellen Voraussetzungen:

- Pegelablesung auf 0,03 dB genau; Frequenz digital gemessen; Temperatur im Meßrohr auf mindestens 0,5°C genau ermittelt und in die Schallgeschwindigkeit eingezeichnet; Dämpfung im Rohr nach Kirchhoff berechnet und für ein komplexes \( k_0 \) verwendet; Stetigkeit im schallhart abgeschlossenen Meßrohr größer als 45 dB; Vermeidung nichtlinearer Aussteuerung der Empfänger und mindestens 20 dB Rauschabstand; bleibt die Abweichung der Meßpunkte von dem gerechneten Feldverlauf 0,03 dB.

Bei der Messung der inneren Ausbreitungskonstante \( T_a \) eines Absorbermaterials wird das Schallfeld in einer Absorberschicht der Dicke \( d \) nach Betrag und Phase ebenfalls an äquidistanten Punkten (ca. 15) abgetastet und mit einer Ausgleichsrechnung im Feldverlauf

\[ P(x) = B \cosh T_a x \]

\( T_a \) bestimmt. Durch die äquidistante Abtastung läßt sich das Verfahren bei digitaler Ableitung im on-line Betrieb mit einem Tischrechner automatisieren. Es werden Meßbeispiele gezeigt.
THE MEASUREMENT OF ABSORPTION COEFFICIENT IN SITU BY A CORRELATION TECHNIQUE

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INTRODUCTION

It has in the past proved difficult to obtain meaningful values for absorption coefficient in situ. The methods available involve either dismounting the material and testing a sample in the laboratory or by substituting an 'identical' sample for the test. The method to be described, using a correlation process, shows that results can be obtained in situ. The original idea was proposed by Goff (1) and further work has been carried out using this process by Burd (2).

EXPERIMENTAL

Random noise is emitted by a loudspeaker and picked up by a microphone. After reflection from a sample surface the microphone signal is cross-correlated with the noise fed into the loudspeaker. A Saicor digital correlator is used with an additional magnetic tape delay unit to compensate for the long acoustical delays involved. Normal incidence absorption coefficients are derived from a comparison of the Fourier transforms of the correlation functions obtained by reflecting the noise from the sample and from a 'perfect' reflector at the same distance from the microphone.

Values of absorption coefficient have been obtained over a wide range of frequencies for several materials. These compare very favourably with those obtained by other normal incidence methods. The method can be used to measure the absorption behaviour of a material in situ which it would otherwise be difficult to estimate, especially in the presence of background noise.

REFERENCES

(2) A.N. Burd, Radio and Electron. Engr., (1964) 27 (5) 387
TRANSPORT RESPONSE OF SOUND ABSORBING MATERIALS

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INTRODUCTION

We treat the transient response of sound absorbers to an incident step pressure pulse and tone burst theoretically using Laplace Transform. They agree with experimental data.

EXPERIMENTAL

For the response of sound absorbers to incident step pressure, experimental data (J. Mason, to be published) show transient before steady state, with same shape as that of exponential charging of a condenser and Corkacoustic and J.M. Acoutest give relaxation times of 0.15 msec. and 0.50 msec. respectively. For incident tone burst, his experimental curves have same shape as amplitude-modulated wave with transient before steady state and transient periods for Acoustic Tile and one inch Boundacoust are both 5 msec.

THEORETICAL

Using Laplace Transform and transfer function from Momma (1), we obtain for incident step pressure,

$$P_{out}(t) = \frac{2\pi P + 4\pi \Phi t}{\cos i(1 - 2jm_o k_o)} + \frac{S_r}{2} e^{S_r t}$$ (1)

where $S_r = -3v_1 h + 3v_1 h - 3v_1^2 / \gamma_1^2$

$2(1 - 2jm_o k_o) \cos i(1 - 2jm_o k_o) 2(\cos i(\cos^2 i - 2jm_o k_o))$

For incident tone burst, $P_{out}(t) = 2\pi A (\gamma^2 - \cos^2 \omega t - e^{-S_r t} \cos gt)$

where $A = \text{constant}$ and $g = S_r \frac{\cos^2 i - 2jm_o k_o}{2v_1}$

$\omega = \frac{v_1}{v_o}$

REFERENCES

(1) A.F. Momma, Physica (1938) V 129
The acoustic testing and development of air handling equipment in a commercial environment requires rapid measurements of the sound power level radiated by various devices over their range of operating duties. Standard methods of spatial averaging of sound pressure within the reverberation chamber can be lengthy and an attempt has been made to devise a method of known accuracy which requires a single measurement of the sound pressure level in the corner of the chamber.

The paper describes the practical evaluation of the test method and its sensitivity to source conditions. The results obtained from over 200 calibrations of the method are examined and a simple statistical comparison made between this and other averaging techniques.

Initial analysis has shown that for a given source the single corner microphone reading may be used to give the spatial average within the 200m$^3$ chamber to within the following approximate accuracies for a 95% confidence:

<table>
<thead>
<tr>
<th>Octave band centre frequency (Hz)</th>
<th>63</th>
<th>125</th>
<th>250</th>
<th>500</th>
<th>1K</th>
<th>2K</th>
<th>4K</th>
<th>8K</th>
</tr>
</thead>
<tbody>
<tr>
<td>Accuracy (dB)</td>
<td>$\pm 3$</td>
<td>$\pm 2.5$</td>
<td>$\pm 1.5$</td>
<td>$\pm 1$</td>
<td>$\pm 1$</td>
<td>$\pm 2$</td>
<td>$\pm 3$</td>
<td></td>
</tr>
</tbody>
</table>
INTRODUCTION. Sound Transmission between two rectangular sectioned enclosures separated by a flexible panel is studied by means of acoustic and plate finite elements.

THEORY. The three dimensional problem is reduced to a two dimensional form by prescribing the mode form in one direction as in the Kantorovich method. The plate motion is expressed in terms of displacements $w = f_1(y) \sin k_m z / d \cdot e^{i\omega t}$, and the enclosure acoustics by the excess pressure $p = f_2(x,y) \cos \pi^2 d \cdot e^{i\omega t}$ - where $d$ is the depth of the enclosure. Finite elements are used for $f_1(y)$ and $f_2(x,y)$, thus the plate becomes a "beam" and the enclosure a "membrane". After substituting in the energy equations and applying Hamilton's principle, the resulting matrix is of an unsymmetric form. However, it can be made symmetric by a simple procedure due to B. Irons (1)(2)(3). The final equation has the form

$$\begin{bmatrix} KM^{-1} & KM^{-1} \theta T \end{bmatrix} \begin{bmatrix} w \end{bmatrix} - \omega^2 \begin{bmatrix} M & 0 \\ 0 & P \end{bmatrix} \begin{bmatrix} w \\ p \end{bmatrix} = 0 \quad \ldots(1)$$

Here $K$ and $M$ are the stiffness and mass matrices of the plate; $S$ and $P$ are symmetric matrices derived from the kinetic and potential acoustic energy; $\theta$ is a coupling matrix obtained from the power function which links the plate to the enclosures.

RESULTS AND CONCLUSIONS. The modes and frequencies obtained from the solution of (1) are useful for solving problems in low frequency sound transmissions. Figure 1 shows the effect of panel size on the 1st and 2nd transmission modes for symmetric rectangular enclosures.

Because the approach can deal with a structure which has a variable geometry in the $x,y$ plane, the method may be used for sound transmission studies in vehicles where the structures are lightly damped.

REFERENCES

EIGHTH INTERNATIONAL CONGRESS
ON ACOUSTICS, LONDON 1974

SOUND TRANSMISSION THROUGH SLITS AROUND DOORS

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INTRODUCTION

The problem of increasing the Transmission Loss of a door by sealing the slits around its edges has been investigated to check the previsions made in (1,2) with experimental data collected about some "normal" doors.

EXPERIMENTAL

The measurements were effected in two adjacent chambers (about 30 m$^3$ each): the receiving one was irregular in shape and highly sound absorbing, while the transmitting one was cube-shaped and semi-reverberant. The opening between the rooms had an area of 10 m$^2$ and was filled with a wall whose TL was about 10 dB more than that of the best door proofed in conditions of perfect sealing. A random noise generator was used as sound source and the signal received was analysed by third octave filters. The microphones were placed in front of the door, at a distance of about 1 meter.

The doors were perfectly sealed along their vertical and top edges, while it was possible to change the width between the threshold and the bottom-edge, inserting in it a ribbon of plastic, as shown in figure 1.

Figure 2 gives the mean values obtained with different width, with (—) and without (——) the ribbon in place.

CONCLUSION

The experimental curves show the different acoustical behaviour of the two slits, due to their different shape and size, in agree with the previsions suggested by (2): in reducing the width at the threshold, the transverse profile of the door has not to be modified.

REFERENCES

(1) M.C.Gomperts, Acustica (1964) 14 1; (2) M.C.Gomperts & T.Kihlman, Acustica (1967) 18 144
THE SOUND INSULATION OF PARTIALLY OPEN DUAL GLAZING

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INTRODUCTION

The possibility of obtaining a moderate degree of insulation from dual glazing with staggered openings has been investigated in a reverberation suite (1) and during an extensive programme of sound insulation tests on a house near Manchester Airport. An assessment was made of the ventilation rate; an important factor when considering the viability of open dual glazing as against sealed windows and mechanical ventilation.

TEST RESULTS

The sound reduction index was measured in the laboratory as a function of the spacing (or cavity) between 2 sets of glazing and the degree of opening. It became apparent that absorptive reveals give an improvement of 2 to 3 dB. Using typical aircraft and traffic noise the 'A' weighted SRI has been calculated. Similar windows (200mm cavity) were built into a house where both aircraft and traffic noise could be measured. The insulation in dBA is given in Table 1, as is the theoretical insulation given by:

\[(\text{Lo})_A - (\text{Lr})_A = (\text{SRI})_A - 11 + 10 \log \frac{V}{ST} \text{ dB}\]

where \((\text{Lo})_A = \text{outside level (dBA)}\)
\((\text{Lr})_A = \text{inside level (dBA)}\)
\(V = \text{room volume}\)
\(S = \text{window area}\)
\(T = \text{room reverberation time}\)

For the test house \(10 \log \frac{V}{ST} - 11 = 4 \text{ dB}\)

<table>
<thead>
<tr>
<th>Window foam reveals</th>
<th>Opening</th>
<th>Insulation dBA</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>Aircraft Movements</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Calculated</td>
</tr>
<tr>
<td>Single 200mm cavity</td>
<td>Closed</td>
<td>27</td>
</tr>
<tr>
<td>Single 200mm cavity</td>
<td>100</td>
<td>16</td>
</tr>
<tr>
<td>Dual 200mm cavity</td>
<td>Closed</td>
<td>42</td>
</tr>
<tr>
<td>Dual 200mm cavity</td>
<td>25</td>
<td>32</td>
</tr>
<tr>
<td>Dual 200mm cavity</td>
<td>50</td>
<td>29</td>
</tr>
<tr>
<td>Dual 200mm cavity</td>
<td>100</td>
<td>27</td>
</tr>
<tr>
<td>Dual 200mm cavity</td>
<td>200</td>
<td>24</td>
</tr>
</tbody>
</table>

Ventilation Rate Measurements: Ventilation rate was measured using a tracer gas (2) with various window openings. Natural ventilation using staggered open dual glazing was little different from single glazing open the same amount. With 5% open area and a typical weather condition a natural ventilation rate of 2 to 3 air changes/hour was measured. This is typical of a mechanical room ventilator.

REFERENCES
(1) R.D. Ford et al., Appl Acous (1973) 5 57; (2) IHVE Guide 1970

662
SOURCES OF VARIABILITY IN FIELD MEASUREMENTS OF SOUND INSULATION BETWEEN DWELLINGS

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INTRODUCTION

The inclusion of a performance standard as a deemed-to-satisfy requirement in the Building Regulations (1) has increased the need for a better understanding of how variability arises in measurements of sound insulation between dwellings.

An obvious source of variability is that associated with the finite precision of measurement; but it is not the only source nor necessarily the most important. Only by studying the individual sources in detail can one attempt to reduce the overall variability.

MEASUREMENT VARIABILITY

The measurement of sound insulation in dwellings made difficult by non-uniform sound fields and unless care is taken considerable variability can occur (1). It is clearly important to know what the typical variability of a sound field is in rooms in dwellings in order to estimate the precision of measurement and the number of fixed microphone positions which should be used. Other important considerations are (a) whether variability varies significantly with room size (b) whether sound fields are equally variable in source and receiving rooms.

NORMALISATION VARIABILITY

The use of the normalisation term $10 \log_{10}^m / 0.5$ does not take into account the volume of the receiving room or the common wall area although both of these can be important in determining the effective sound insulation. While there is no problem as far as tested houses are concerned difficulties can arise when the test results are applied to other dwellings with identical construction details but with different room arrangements.

CONSTRUCTION VARIABILITY

In many cases the major source of variability is associated with the inherent differences in performance of nominally identical constructions. It is likely that different construction types (cavity masonry, solid masonry, system-built etc) will have different variabilities. The value of construction variability must be known if the expected pass rate against a given performance standard is to be determined.

REFERENCES

This paper is concerned with the development of optimal design schemes for sound insulating partitions. The words optimal design are intended to be broad enough in scope to include structural variables, as well as the acoustic variables. Partition models will include conventional sandwich structures and lightweight composite materials.

The optimization schemes are based, as far as the acoustical variables are concerned, on recognizing the familiar dip in noise reduction at the critical frequency, and then trying to optimize the behavior either below or above that dip. Thus, one approach is to push the critical frequency as far as possible up the frequency scale from the lowest partition resonance curve, thus making the best possible use of mass law (M.A. Lang, Ph.D. thesis in preparation). The complementary approach is to reduce the critical frequency, so as to bring it below the audible range, which is possible for lightweight structures (R.H. Lyon, 1973, private communication).

It is important to note that the structural variables play an important role in practice, and they could significantly affect the acoustic behavior of a wall panel. For example, it can be shown that a compressive pre-stress can lower the panel resonance frequencies, increase the critical frequency, and increase the mass law transmission loss (C.L. Dym, to be published).
INSULATION OF REVERBERANT SOUND THROUGH DOUBLE GLASS' PANES

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INTRODUCTION

In previous papers (1,2) we have already discussed the sound insulation on scale models of single and laminated glass panes with different fixations. To complete this work a systematic research is done on the sound insulation of double glass with different edge constructions and fixations. Combinations with laminated glass panes are also measured.

MEASURING EQUIPMENT AND CONDITIONS

The sound insulation of the different glass panes is measured between two reverberant rooms by means of the well known formula given by the ISO-Recommandation R-140. The measurements are done completely automatic by means of a real time analyser connected to a computer, in order to achieve a high accuracy.

COMPARISON BETWEEN THEORY AND EXPERIMENT

A lot of measurements are done on double glass construction with all kinds of thickness and different sizes. The influence of size, fixation and edge construction is extensively studied.
The experiments are compared with London's theory (3) on the transmission of reverberant sound through double panels.

REFERENCES

(1) A. Cops et al., Proc. Internoise-73 (1973) p.276
(2) A. Cops et al., to be published in Acustica (1974)
EIGHTH INTERNATIONAL CONGRESS ON ACOUSTICS, LONDON 1974

FACTORS INFLUENCING SOUND INSULATION BETWEEN TWO ROOMS WHEN USING SUSPENDED CEILINGS

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BACKGROUND
In former days partitions were erected right up to the structural ceilings. Modern building methods mean that installations are located in the plenum and concealed by a suspended ceiling. Consequently the partitions are finished just above the suspended ceiling. By this procedure a new sound transmission path has arisen through suspended ceiling - plenum - suspended ceiling, see Fig. 1. In other words the sound insulating properties of the suspended ceiling must be taken into consideration.

THEORIES
Fundamental investigations were carried out in USA (1, 2) in the beginning of the sixties and the basic equation was formed. In the basic equation a wide range of dimensional and acoustical parameters are included. A standardized test method evolved from collected experience and is adopted principally in USA. This test method implies that all parameters, except the transmission through the suspended ceiling itself, are fixed.

Uncritical use of sound insulation values obtained from the test method can result in errors of several dB. You have to consider all possible transmission paths and check if there is any difference in parameters compared to the test chamber.

The basic equation has been developed further at The Lund Institute of Technology and a series of measurements carried out (3). The development work has resulted in general graphs making it possible to transfer values from the test chamber to real conditions.

To make listed sound insulation values easily accessible for consultants and architects we have made some simplifications. A reference room, which is acoustically similar to Swedish hospital rooms, has been specified. Each of our suspended ceiling products, being adapted for sound insulation, has an insulation value corresponding to the reference room. Correction for deviations in the parameters can easily be done by correction factors adjusted to each product and arranged in simple tables.

The above method of calculations has been applied in several cases with very good results.

REFERENCES
EIGHTH INTERNATIONAL CONGRESS ON ACOUSTICS, LONDON 1974

TENTATIVE MEASUREMENTS OF THE REVERBERATION TIMES AND INTERNAL DAMPING OF PARTITIONS

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INTRODUCTION

Measurements made in the last years seem to have practical value, e.g., in the design & control of partitions and their transmission losses to sound pulses.

MEASURING METHODS

The tests must be made in the lab and in situ. It is indispensable to submit the partition to oscillating pressures which are absolutely independant of the adjacent room's acoustics. To apply the most unfavourable conditions found in practice, the pressures are concentrated in the center of the partition. We use mechanical vibrations from an electrodynamic generator. As in standard reverberation tests we record decreasing oscillations.

We call "reverberation time" T the time corresponding to a 60dB decrease in acceleration amplitudes. To measure the internal damping, we also measure the average decrease in dB per cycle D.

TYPICAL RESULTS

Two tests made in the laboratory on 10m² samples.

Fig. 1 is from a light partition, dry construction.
Dynamic transmission loss: 30dB at 250Hz to 70dB at 2500Hz.

Fig. 2 is from a light concrete block wall, 9cm, 85kg/m².
DTI 25dB at 250Hz to 40dB at 1500Hz.

CONCLUSIONS

The longest T we found for normal constructions is 120mS.
T & D are not important in normal cases.

REFERENCE

REDUCTION INDEX FOR A PANEL IN A BAFFLE

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INTRODUCTION
To avoid small room volumes in transmission laboratories test specimens are often mounted in baffles. This results in that measured reduction indices vary with the baffle dimensions.

ANALYTICAL AND EXPERIMENTAL METHODS
The transmission problem for a rectangular and clamped panel can not be solved directly by using the eigenvector method. Instead a simplified modal approach has been used (1). The analysis is an extension of the results obtained for unbaflled, clamped and simply supported panels (1).
Experimentally the results are tested in two ways; first by keeping the panel area constant and changing the baffle area and thereafter by varying the panel area while the total area is kept constant. Experimental and theoretical results show good agreement.

CONCLUSIONS
The baffle does not influence the transmission for \( f > f_c \). For \( f < f_c \) the transmission is a function of panel and baffle dimensions as well as of the boundary conditions. If the total area of the baffle plus panel is kept constant the resulting effect of changing the panel area \( S \) is shown in Fig. 1. The panel material is gypsum board. The edges are clamped.

For low frequencies the variations of the transmission is mainly determined by the baffle dimensions, \( R \) is increased as \( S \) is decreased. For higher frequencies the baffle effect becomes less significant and is also counter-balanced by the radiation efficiency which tends to decrease \( R \) as \( S \) is decreased. Less constrained edge conditions and larger loss factors increase the baffle effect.

REFERENCE
(1) A. Nilsson. 1974  

Fig. 1. \( R \) as function of \( S \).
EIGHTH INTERNATIONAL CONGRESS ON ACOUSTICS, LONDON 1974

A DATABANK OF TRANSMISSION LOSS MEASUREMENTS

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INTRODUCTION

For the last four years data on the construction details and transmission loss of multi-layered panels has been collected from published and unpublished sources. Details are available on more than 1500 panels, although to date, only 1040 of these have stored on the computer system that is used to access the large amounts of data. The structure of the databank is described in a previous paper (1), and a general-purpose access system is being developed from a pilot model already in use (A.J.Gillam, to be published).

USE OF THE DATABANK

Several uses of the databank are envisaged, but two main areas have been explored so far. Firstly, there is a lack of generalised transmission loss theory to give predictions for the range of panel types considered for buildings. Such theories as do exist are heavily mathematical (requiring computer facilities) and are of limited scope. Thus those people involved with the design, construction and certification of buildings usually rely on measurements, but they do not find it possible to compile comprehensive collections due to the wide range of sources of relevant data.

During the last year, specific enquiries from such people have been processed using the present access system. The enquiries break down into two types: a) what is the performance of construction x?, and b) how do I get the performance y? Even with over 1000 panels to choose from, in some cases the construction x does not appear in our data. In these cases the parameters of search are broadened to find similar panels to x. Then the skill of the operator determines how much useful data is given to the enquirer, although a more sophisticated computer system is envisaged to do this job. It has been found that enquirers always take the attitude 'some information is much better than none at all' and if exact information is not available, will be glad to learn such things as the probable shape of the TL performance graph.

The second main area of use concerns the databank as a statistical store. By analysis on the data, generalised information (with confidence limits) is emerging on such things as use and spacing of studs, use of infill layers etc. It is hoped this work will lead to the formulation of a set of empirical rules for calculation of the transmission loss of any panel.

REFERENCES

EIGHTH INTERNATIONAL CONGRESS ON ACOUSTICS, LONDON 1974

L'OPTIMISATION DES STRUCTURES PHONOABSORBANTES UTILISÉES À L'INSONORISATION DES ESPACES INDUSTRIELS

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Vereș A Biborosch L D

On présente la conception et la réalisation des quelques traitements phono-absorbants pour des espaces industriels bruyants, constitués par la combinaison de structures absorbantes en champ continu, appliquées aux parois (1), avec des absorbants acoustiques individuels (réalisés de plaques planes) qui ont une efficacité absorbante maximum pour des longueurs d'onde supérieures ou égales aux dimensions de celles-ci.

On a étudié dans la chambre réverbérante l'influence de l'angle d'inclinaison des plaques de laine minérale, en rapport avec le support, pour des valeurs égales à 0°, 15°, 30° et 45° et pour un indice de revêtement, défini comme rapport entre l'aire de la plaque projetée sur le support et la surface afférente, ayant des valeurs entre 0,4 et 1,0, fig.1. L'absorption maximum correspond à l'angle d'inclinaison de 15° et à l'indice de revêtement de 0,8, fig.2.

L'efficacité technique-économique est exprimée par le rapport ΔL/C, en dB/lei, où C représente le coût spécifique considéré à la valeur de 100 lei, fig.3. La réduction ΔL, du niveau du bruit, avec la caractéristique de fréquence linéaire présente une atténuation typique, en fonction seulement des coefficients d'absorption du plafond, d'après l'expression (1):

\[ ΔL = 10 \log \left( 1 + \frac{1}{2} \frac{α - α_0}{α_0} \right) \]  

où \( α \) représente le coefficient d'absorption du plafond de la halle avant l'application du traitement phonoabsorbant.

Références
(1) V. Focșa et autres, la IVème Conférence nationale d'acoustique, Bucarest 1973, vol.1, 437; 453

Fig. 1 La position des plaques phono-absorbantes

ΔL en fonction de coût C

Fig. 2 L'atténuation

Fig. 3 Les valeurs α en fonction de l'inclina-
SOUND TRANSMISSION AND RADIATION OF PIPES WITH
INTERNAL ACOUSTIC EXCITATION

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INTRODUCTION

Measured transmission losses through cylindrical pipe walls have been found to be \( \approx 30 \text{ dB} \) less than predicted by breathing-mode theory \((1)\) for \( k_{\infty} r_i < 1 \) \((k = \text{wave number in fluid}, r_i = \text{inner pipe radius})\). It appears that pipe bending motion, excited by the \((1, 0)\) acoustic mode, can radiate more power than the plane wave excited breathing-mode.

EXPERIMENTAL

Figure 1 shows measured and predicted transmission losses. The low values of measured transmission loss result from bending mode radiation. The acoustic excitation of bending modes is essentially different below and above the \((1,0)\) mode cut-off frequency. Below cut-off the excitation is localised; away from any sources or discontinuities, the driving pressure decays rapidly. Above cut-off, the excitation is transmitted throughout the pipe. Thus in the transition region \(k_{\infty} r_i \approx 1.8\), these bending modes are driven by localised acoustic pressures (near any source or pipe discontinuity) below the \((1,0)\) mode cut-off frequency, and by the acoustic pressure along the entire pipe length above cut-off. Thus in the transition region \(k_{\infty} r_i \approx 1.8\), the "transmission loss" drops steeply.

THEORETICAL

For \(k_{\infty}/k_i \lesssim 0.6\) and \( k_{\infty} r_i < 1\), the radiation efficiency, \(R_{rad}/\rho c \pi a L\), for bending modes of a circular cylinder, outer radius \(r_o\), length \(L\), simply supported at its ends, is \((2)\):

\[
R_{rad} = \frac{4}{3} (\frac{\rho c}{\omega})^2 \left( \frac{k_i}{k_n} \right)^3 \frac{4}{3} \left( \frac{\kappa}{\kappa_n} \right)^4 \left[ 1 + \frac{3}{(k_{\infty} L)^2} \left( \frac{k_{\infty}}{k_i} \right)^2 \left( \frac{\sin k_{\infty} L}{k_{\infty} L} - \cos k_{\infty} L \right) \right]
\]

where \(\rho c = \text{characteristic impedance of fluid} \) and \(k_{\infty} = \frac{n \pi}{L} = \text{structural wave number}\). Even structural modes radiate less efficiently than odd modes when the beam is acoustically short, i.e., \(k_{\infty} L < \pi\).

REFERENCES

INTRODUCTION

Use of urea-formaldehyde foam and other thermally insulating fillings in cavity brick external walls is increasing; a case of poor sound insulation with a standard floor flanked by a foam-filled wall prompted a theoretical study and further experiment. Poor sound insulation between upper floor rooms has been found with a new type of ceiling.

FOAM FILLINGS

Search of records revealed an example in which the party as well as the external wall had been filled with foam; insulation averaged 8 dB below that expected for a solid wall of the same mass. A theoretical treatment for an infinite wall with similar leaves predicts such a reduction between the mass-spring-mass resonance frequency and an upper limiting frequency when the former exceeds half the critical frequency of each leaf. Symmetric and antisymmetric waves in the composite wall contribute equally to transmission. Analysis of transmission past a junction in a filled wall at which one leaf is clamped shows that when both waves are propagating half the incident power is transmitted. With such high junction transmission, a flanking wall transmits nearly as much power between two rooms as a similar party wall of half the area of the part of the flanking wall adjoining each room.

Vibration measurements in the building where the effect was first suspected confirmed that the filled walls radiated much more power into the receiving room than the ceiling. Party wall insulation in two pairs of houses was measured before and after external walls were filled. One, generally well above standard beforehand, afterwards had a total adverse deviation of 50 dB; reduction was very marked at low frequencies. With the other, insulation, initially poor, fell an average of 6 dB, with especially large high-frequency reductions.

The stiffness of a small sample of foam measured in the laboratory was less than that suggested by the field measurements, although high enough for serious effects to occur; the difference could be due to samples varying or to problems in testing a fragile material.

PLASTIC CEILINGS

In an extensive survey of sound insulation in new dwellings several examples were found in which insulation was much less between top floor rooms than between ground floor rooms. In all cases a lightweight plastic ceiling had been used. Sound level measurements in the roof space indicated that the insulation of the party wall there had been reduced by direct air paths and that the insulation provided by the ceilings was extremely low at the lowest frequencies. Attainment of adequate insulation with such ceilings would seem to require good insulation from the separating wall in the roof space.
In this paper I try to reinforce the simple chain filter methods. We know from window measurements that grazing sound gives a great amount of transmitted sound, but the usual integration to get the diffuse transmission factor $\tau_d$ is $\tau_d = 2 \int_0^{\pi} \tau(\phi) \sin \phi \cos \phi \, d\phi$. For grazing sound $\phi = \pi/2$, so $\cos \phi = 0$. In calculations it is thus better to start with a driving sound pressure instead of the projection of the incident intensity which is responsible for the $\cos \phi$-factor. The usual way of integration, not to $\pi/2$, but to almost grazing sound which has often been used for single leaf partitions is not suitable for small partitions and especially not for multiple leaf partitions.

If a correction is made in the radiation impedance instead, the discrepancy is eliminated. This shows that there is grazing sound and that the method of angle-limit is only seemingly correct for single leaf partitions.

If the incoming wave has the direction $\phi$ we use a tentative radiation impedance $pc/((\cos \phi + G/\pi))$ where $G$ is smaller for small dimensions of the partition. (For infinite area $pc/\cos \phi$). For a small, empty air space the adiabatic compression gives an admittance $Y_0 = j\omega d/(pc^2)$, but for oblique incidence and with flow resistance $\sigma$ and allowing for adiabatic ($\mu = 1.4$) to isothermal compression ($\mu = 1$) we get $Y = Y_0 \cdot (1.4/\mu - \sin^2 \phi / (1 + \sigma/j\omega))$, where the second term depends on the space wave. The leaf impedance has the usual form $Z = j\omega c (1 - (f/f_0)^2 (1 - j\eta) \sin^2 \phi)$, where $\eta$ is the resulting internal and boundary losses and $f_0$ the coincidence frequency.

For very small air spaces $\sigma = 12 \nu d^2$ ($\nu$ = viscosity of air, $d =$ space width), so there is flow resistance in the empty space which can be one explanation of the good performance of an extra leaf with only 0.5 mm distance from one leaf in a double glazing (20.5 mm). The curves are calculated for diffuse sound with $\eta = 0.01$, $G = 20$ and $\sigma/pc = R = 3$. 

\[
\begin{array}{c|c|c|c|c|c|c|c|c}
\text{TL dB} & \text{6 mm} & \text{3} & \text{20.5} & \text{3} & \text{4} & \text{20.5} & \text{0.5} \\
\hline
50 - & & & & & & & \\
40 - & & & & & & & \\
30 - & & & & & & & \\
20 - & & & & & & & \\
10 - & & & & & & & \\
\hline
100 & 200 & 500 & 1000 & 2000 & 3150 & \text{Hz} & \\
\end{array}
\]
ACOUSTIC REQUIREMENTS IN OFFICE BUILDINGS

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The National Swedish Board of Public Building has stipulated requirements including acoustics (traffic noise, installation noise, sound insulation and room acoustic conditions) for different types of public buildings starting with a code for office buildings in 1968.

As a basis for a revised code some public office buildings have been analysed in 1973 with interviews combined with measurements of the actual acoustic conditions.

15-18 individuals working in small room offices were statistically selected in each of 4 buildings and interviewed with up to 150 questions, 46 of which concerned different aspects of the experienced airborne sound insulation. Also a secrecy demand coding was made for each official both by the interviewer and the interviewed individual.

The actual acoustic data were measured for the rooms in question after the interviews. Thus 157 measurements of the airborne sound insulation index ($I_a$-value according to ISO R 717) were made covering nearly all of the possible partition and corridor walls adjacent to the interviewed officials' rooms.

Though the statistical basis is limited some clear tendencies can be read from the results. Examples regarding privacy/secrecy aspects are:

- $I_a$ 35 dB is sufficient for normal office privacy demands, also with a fairly low background noise level (below 35 dB(A)).

- $I_a$ 44 dB is sufficient to fulfill the secrecy demands for most cases claiming speech information secrecy.

For rooms with extreme secrecy demands $I_a$ 52 dB is chosen giving some safety margin for the more trivial means for eavesdropping.
When high airborne noise isolation is required for windows in the exterior walls of buildings the architects are recommended to use spaced glass assemblies, or laminated monolithic panes, or a combination of both.

Laminated glass employing two (or more) layers of glass laminated with a plastic layer, e.g. polyvinyl-butyrate, can improve the TL (transmission loss) near the coincidence region considerably. However, if improperly selected it may not change appreciably the STC rating or the sound insulation index $I_a$. The desired improvement of an effective noise reduction is achieved only by proper selection of the laminated glazing system, especially if spaced glass assemblies with combined monolithic solid panes and laminated panes are used.

In order to investigate the potentials and limits of increasing the sound insulation of windows by using laminated glass panes in monolithic and double glazed windows, a great number of experiments has been conducted in a standard two chamber sound transmission loss test laboratory. The results of these series of tests are presented in a manner to facilitate the selection of glass assemblies for windows in order to achieve prescribed sound insulation classes.

The experimental results are compared with values calculated for various panes with different mechanical loss factors. It is shown that the main improvements of the sound isolation are already gained with moderate mechanical losses of the laminated pane.
EIGHTH INTERNATIONAL CONGRESS ON ACOUSTICS, LONDON 1974

TRANSMISSION LOSS OF DOUBLE WALL WITH PARTITION IN AIRSPACE

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INTRODUCTION
On the transmission loss of a double wall, the influence of the finite sized airspace and the improvement by the partition in the airspace, are discussed.

THEORY
Schematic double wall with finite airspace.

In the finite sized airspace as shown in Fig.1, sound waves must satisfy the boundary condition at the edges or the partitions. Assuming that the radiation impedance is equal to that of the infinite leaf, and the leaf has a mass reactance only, the transmission coefficient is given by

\[ \tau_{i,m}(\omega) = \frac{\eta}{\eta + J\omega t_{i,m}} \]

\[ t_{i,m} = \frac{\alpha}{2\sqrt{\eta A + \beta}} \]

for the \((1,m)\)th mode with respect to the \(x,y\)-directions, where \(\tau_{i,m}\) is the normal incident transmission coefficient given by London(1). The random incident transmission coefficient become

\[ \tau = \sum_{i,m} \tau_{i,m} \cdot \tau_{i,m} \cdot \cos \theta_{i,m} \]

\[ \theta_{i,m} = \sin \theta_{i,m} \]

where \(\tau_{i,m}\) is the coupling factor between the incident energy flux density and the \((1,m)\)th mode.

On a double wall with the finite sized airspace, the mass-spring-mass resonances concentrate at natural frequencies of the wall,

\[ f_{i,m,n} = \frac{c^2 \omega^2}{\eta A + \beta} \]

\[ f_n = \frac{c}{2d} \left( \sqrt{\frac{2\rho d}{\pi^2 M + (2\pi)^2 n^2}} \right) \]

at any angle of incidence, though they exist at every frequency above \(f_n\) at some angle of incidence in case of the infinite. Then the T.L. of the finite is larger than that of the infinite, expecting \(f_{i,m,n}\).

EXPERIMENT

Fig.2 shows the measured T.L. of plywood double walls with and without the partitions in the airspace. The plywood partitions increased the T.L. 4-8dB.

REFERENCE


Fig.2 Transmission loss of plywood double wall with partition.
INTRODUCTION

The development of corrugated plates and the possibility of using them as load-carrying and stabilizing elements in buildings has created new possibilities for light structures. The sound insulation/radiation problem is however a preventing factor and very few investigations have so far been made.

EXPERIMENTAL

The transmission loss for corrugated steel plates with different through center (L) and constant depth (H) of corrugation has been studied. Fig. 1 shows a few plates and fig. 2 the corresponding transmission loss curves.

![Fig. 1](image)

![Fig. 2](image)

THEORETICAL ANALYSIS

The plates have approximately the same bending stiffness/weight relationship. The deep drop in the transmission loss curves occurs in different frequency ranges and is assumed to depend on resonance in the airspace formed by the corrugation.

A simplified theoretical model for sound radiation from a corrugated plate has been made. Calculations with the model show a resonance frequency which is dependent on the through center and show the same trend as the drop in transmission loss, so that the frequency decreases when the through center or depth increases.

Calculations of sound radiation from a corrugated plate using finite element method is going on and will be published in the summer of 1974.
A semiconductive element known as magnetodiode can be used in an electromechanical transducer. These elements have between the P and N zones an asymmetrically located recombination zone. Their conductivity is highly influenced by a magnetic field perpendicular to their surface; due to the recombination of deflected electrons. The paper describes a new type of electromechanical transducer with a moving-iron electromagnetic system in a push-pull arrangement. Instead of a coil around the moving-iron as in known systems, two magnetodiodes are used, each cemented to one of the pole-pieces in the air gap. Both diodes are fed through resistors connected in a bridge circuit. The polarity of the diodes is such that both contribute in the same sense to the output voltage on the diagonal of the bridge.

The advantage of this transducer is its high mechanical stability, due to its low d.c. magnetic induction, even when it is compliant. In contrast to other magnetic transducers this one is displacement-sensitive.

The author has produced several models using the Czechoslovak magnetodiodes made to order. (The inactive cover of magnetodiodes available on the market - Sony type MD-130C is too thick for this purpose).

The new transducer is suitable as a displacement-sensor in a frequency range 0 - 100 Hz with a threshold sensitivity (due to the noise level in diodes) of 63 nm. This transducer may also be used in a subsonic microphone.

Its only disadvantage is a high sensitivity drift due to temperature changes. This is however reduced by the use of a push-pull system with two diodes in the bridge circuit.
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ELECTROSTATIC TRANSDUCER IN SPIRAL ARRANGEMENT

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INTRODUCTION

The electrostatic transducer (Fig. 1) is provided with two fixed strip electrodes (5,7) and a vibratable thin strip diaphragm (6,8) all spirally wound on a cylindrical insulating core (9). The distance between the electrodes and diaphragm is determined by spacer strips (1,2,3,4) of insulating material interleaved between the electrodes and diaphragm.

DESCRIPTION

The transducer comprises a laminate structural assembly rolled into a spiral form of a cylindrical shape. The air gap is defined by cut-out portions in the spaced strips and by the thickness of the strips. The spacer strips are arranged so that the cut-out portions of the spacers in opposite sides of the diaphragm are contrary sense. In this way a separation of the oscillations from the two sides of the diaphragm is obtained. The acoustic outputs of this push-pull electrostatic transducer are located in both the bases of the cylindrical assembly.

APPLICATIONS

One of the applications of the transducer is the earphone. The frequency response of the transducer having the external diameter of 1" in connection with an artificial ear of the Brüel & Kjaer type 4153 is shown in Fig. 2. Polarizing voltage is 400 V, alternating voltage 2 x 50V, acoustic pressure is 108,5 dB SPL. The same type of transducer is also used as a tweeter.

Fig. 2 Frequency response of the electrostatic earphone

REFERENCE

Czechoslovak. pat. No. 144 974
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FINITE ELEMENT ANALYSIS OF TWO-DIMENSIONAL ELECTROMECHANICAL RESONATORS

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Procedures The finite element technique [1][2] is applied to two-dimensional electromechanical plate resonators that vibrate in plane (Fig. 1). The formulation is made for the electromechanical coupling included. The displacement functions for a triangular element into which the plate is divided are assumed to be linear. The computer program developed is applicable to calculating the natural frequencies, the modal shapes and the electrical motional admittances of an arbitrarily-shaped resonator and filter with the electrodes arranged arbitrarily.

Finite element procedure is such that the Lagrangian for the potential energy, the kinetic energy and the virtual energy externally supplied is to be minimized. The relation obtained for the nodal displacement vector $\mathbf{d}_e$, the nodal force vector $\mathbf{F}_e$ and the applied electric voltage $V_E$ is given for element $e$ as

$$\mathbf{F}_e + \epsilon^S V_E \mathbf{D}_e^* \mathbf{h}/u = (K_e - \omega^2 J) \mathbf{d}_e$$

where $\epsilon^S$ is electrical permittivity, $\mathbf{D}_e^*$ a coefficient matrix, $\mathbf{h}$ piezoelectric stress tensor and $K_e - \omega^2 J$ the system matrix of the element. The superscript $*$ indicates transpose. The applied voltage is expressed in terms of the nodal force. The equation for the whole system can simply be developed by assembling Eq. (1) that is for each element.

The electrical motional admittance is given for the element as

$$Y_e = j\omega \epsilon^S \mathbf{h}^* \mathbf{D}_e \mathbf{d}_e'$$

where $\mathbf{d}_e'$ is the nodal displacement vector for a unit voltage applied. The motional admittance for the whole plate is obtained by summing up for the elements with electrodes provided.

Calculated examples Modal shapes and electrical motional admittance of a square plate with a rounded corner are shown in Figs. 2 and 3, where the values are normalized. Separation of the degenerated modes, FL/L2 and F3/L2', is clearly seen and the electrical response is well demonstrated for partial electrode arrangement.

Fig. 2

Fig. 1

Fig. 3


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SONDE MICROPHONIQUE FINE AVEC COMPENSATION PAR RÉSONATEURS, POUR MESURE DE PRESSION DANS UN GAZ CHAUD OU EN MOUVEMENT

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PRINCIPE

Les sondes formées d'un tube fin et d'une cavité contenant le microphone permettent des mesures de pression dans de petits espaces, dans des gaz chauds ou en mouvement, sans perturbation du champ sonore ou du champ de vitesse. Mais leur réponse en fréquence présente une forte résonance si la sonde est peu amortie, et une perte de sensibilité très rapide aux fréquences croissantes si elle est amortie (fig. 1, courbes a et b).

Il est possible de compenser ces effets en utilisant plusieurs résonateurs en série (fig. 2) avec un emplacement judicieux du microphone. On obtient ainsi, par optimisation des résistances acoustiques, une courbe de réponse plus plate et plus étendue en fréquences (fig. 1, c).

ÉTUDE THÉORIQUE

Le calcul utilise la théorie linéarisée du résonateur d'Helmholtz, avec un terme résistif dans chaque col, et en écrivant, à chaque jonction, les conditions de continuité en débit et pression. On obtient ainsi une succession d'équations permettant d'introduire un nombre illimité de résonateurs en série. Avec deux seulement, le calcul numérique est aisé par ordinateur et permet de connaître l'influence des divers paramètres géométriques et de l'amortissement (fig. 1, c).

ÉTUDE EXPERIMENTALE

La valeur des termes d'amortissement ne peut être connue qu'expérimentalement. On a réalisé pour cela un ensemble de tubes et cavités interchangeables, avec un petit microphone à électret, permettant d'étudier des sondes à 1 ou 2 cavités. Les courbes de réponse obtenues (par comparaison avec un microphone étauon) confirment les tendances indiquées par l'étude théorique. Les mesures permettront de choisir les valeurs optimales des paramètres pour une bande de fréquence donnée.

L'utilisation de ce type de sonde dans un écoulement à faible vitesse sera évaluée dans une phase ultérieure de l'étude.
The subject of the investigation is the membranes of dynamic microphones produced by "Tonsil" MD9, which are made of polycarbonate foil Makropol N. In these membranes, shaped like a cupola, suspension is made as an indissoluble part of membranes in the form of concentric grooves, which - enlarging the membrane susceptibility - eliminate simultaneously possible circuit vibrations. The membranes are produced with the application of automatic method which in theory only guarantees the repetitions of successive copies. The aim of the investigation is - among others - the determination of repetition degree. It is necessary then to determine a certain number of the selected parameters of the membrane, which can be done - by way of experiment - for successive copies. The parameters allow also to achieve an absolute estimation of the membranes quality, as well as to introduce some modifications to the scheme of the substitutional microphone. The present substitutional scheme, joined to the membrane in the form of the elements - which are joined in a series - that correspond to the mass, susceptibility and loss coefficient of the membrane, is of a small adequacy to the real system. The selection of the parameters which characterize the acoustic properties of membrane depends on the qualification of a microphone membrane to a certain kind of a vibration system. Preliminary experiments allow to treat - with a certain approximation - the membrane motion in the range from 1st resonance as a motion of vibrating system of one degree of freedom. Thus, we assume - as it were - the concentration of the whole membrane mass in its centre, as well as non-existence of the marking circuit and radius vibrations on the membrane cupola. Such an approximation allows to make use of the theory and formulas referring to the vibration point of one degree of freedom, whose motion is described in the equation: $m\ddot{x} + r\dot{x} + kx = f(t)$. In such an approach it is possible to define the following parameters characterizing the membrane, basing on the analysis on the quantities $f_r$ and $\Delta f_r$, read from the resonance curves estimated by way of experimental rigidity $k = 2\pi f_r m$, resistance $r = 2\pi \Delta f_r$, quality factor $Q = \frac{f_r}{2\pi}$, suppression coefficient $\delta = \Delta f_r$, where $f_r$ - velocity resonance frequency, $m$ - membrane mass, $\Delta f_r$ - halves latitude. The estimation of the resonance curves requires a testing system which measures the value of deflection and membrane centre velocity for various frequencies in the resonance proximity, with excitation of the membrane by an acoustic signal. It is realized in an optic system with an amplitude modulation. The application of 2 beams - for the membrane centre and for another facultative point on the membrane cupola - with the simultaneous application of the phase displacement tester - allows to register a possible difference of the phases between the vibrations of the examined points. The measurements which have been carried out proved rightness of the qualification of the membranes to the systems of one freedom degree, and made it possible to determine the absolute values of the enumerated parameters as well as the effect of loading the membrane with a coil.

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THE ELECTRODYNAMIC TRANSDUCER AS AN IMPEDANCE TRANSDUCER FOR VIBRATIONS AND ACOUSTICS

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1 - Introduction - The transducer by itself

The electrical impedance $Z_{el}$ of the transducer is related to the mechanical one $Z_m$ attached to the moving coil by the well-known eqn:

$$Z_{el} = R + j\omega L + \frac{B^2 l^2}{Z_m}$$

It is then obvious that $Z_m$ can be deduced from $Z_{el}$ provided the intrinsic characteristics of the transducer be determined.

For such purpose, assuming this later to a single degree of freedom system, it can be drawn the Kennelly circle [1], [2].

Assuming besides the mechanical damping to be mostly of the viscous type, the Kennelly circle can be plotted according to the reduced frequency by the relatn:

$$\frac{\omega}{\omega_c} = \frac{Q(\alpha - \frac{1}{\alpha})}{\sqrt{1 + Q^2(\alpha - \frac{1}{\alpha})^2}}$$

The transducer being thus defined can be used for measurements:

2 - Vibrations - The electrodynamic exciter attached to a structure

The purpose is to compare the behaviour of a structure, near a resonance, to a single degree of freedom model. This can be reached by using either one of the following methods:

a) Electrical impedance plotting

Provided that the mechanical fixation of the moving coil to the structure be stiff enough, each significant resonance leads to an elliptical loop in the electrical impedance curve, from which can be deduced the parameters of the model equivalent to the structure.

b) Excitation interruption

By switching off, at a predetermined phase, [3], a steady sine excitation applied to a structure, the open-ended voltage given by the transducer can be expressed as

$$u = \frac{\lambda}{\delta} \frac{B^2 l^2 I}{\sqrt{1 + Q^2(\alpha - \frac{1}{\alpha})^2}} e^{-\frac{\omega_c}{\omega_c} \sin \omega t}$$

From its recording it can be deduced also the equivalent model.

3 - Acoustics - The electrodynamic loudspeaker radiating into a room

The purpose is also to compare the behaviour of a room, near a resonance, to a single degree of freedom model. The same above-mentioned methods can be used, account taken that all acoustical and most mechanical parameters of the loudspeaker vary according to the frequency.

References

FREE-FIELD CORRECTIONS FOR CONDENSER MICROPHONES

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INTRODUCTION

A comprehensive theoretical investigation of the free-field response of standard condenser microphones (B&K typ 4144 and 4145) together with a detailed analysis of factors influencing the free-field correction are given. The work continues and improves the knowledge of a topic treated previously by other authors (1),(2),(3&4),(5).

THEORETICAL MODEL AND THE SOLUTION PROCEDURE

The condenser microphone is mounted on a semi-infinite rod having the same diameter as that of the microphone. The direction of propagation of a plane, periodic wave system is perpendicular to the microphone diaphragm (0° incidence). The diaphragm vibrations are assumed to be rotationally symmetric. The protecting grid is removed from the microphone.

Sound field solutions in three regions: (a) the external free-field which is solved by Wiener-Hopf technique, (b) the internal field inside a recess and (c) the vibration of the diaphragm, are connected by appropriate boundary conditions. This leads to a system of simultaneous matrix equations in which sound pressure and particle velocity are represented by column infinite matrices whose elements consist of expansion coefficients arising from expansion by orthogonal functions. After insertion of expressions for diaphragm impedances, diaphragm tensions and volume compliances for the microphones the numerical calculations were carried out on a digital computer IBM 370/165.

NUMERICAL RESULTS

The numerical results are compared to experimental findings given in (3&4) and they comprise: (a) convergency criteria for the numerical solutions for the free-field corrections leading to an inaccuracy less than 0.03 dB, (b) excess pressure distribution across the diaphragm as a function of frequency, (c) diaphragm displacement distributions for a number of frequencies, (d) frequency response curves as a function of diaphragm parameters (i.e. impedances and tensions), (e) free-field corrections for the microphones investigated, (f) a detailed analysis of the free-field corrections dependence on (i) diaphragm impedance (ii) depth and radius of recess (iii) type of gas surrounding the microphones (H₂ and air investigated). A good agreement between calculated and measured results is found.

REFERENCES

EIGHTH INTERNATIONAL CONGRESS
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INSTALLATION FOR CAPTURE OF SOUNDS BETWEEN 20 Hz
AND 200 kHz USING TELETRANSMISSION

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INTRODUCTION
The installation permits the determination of the sound sources, their intensity and the frequencies emitted by sound sources. We used a capacitive transducer which modulates, in frequency, an oscillator with the frequency of 72 MHz.

DESCRIPTION
The installation consists of two distinct parts: fig. 1
1. The sound capturing system consisting of wide frequency band condenser-microphone, which contains into the same assembly the frequency modulated oscillator and the final amplifier. This system is transistorized and, consequently, small in size, it can be easily moved and used in difficult to access places for other types of equipment (which are usually bigger).
2. The receiver-recorder system is separated from the sound capturing system and can be moved at long distances.

This part consists of an input amplifier working on 72 MHz, a frequency mixer which gives an intermediary frequency of 56 MHz and a multichannel analyzer coupled to an oscilloscope and a recorder.

The determinations carried out, showed that the linearity in the frequencies band received is under 12 dB, permitting the sounds transmission to a distance of less than 500 m, from the receiver.

REFERENCES
Electrostatic influence is the basic principle of the new condenser-type gradient-microphone. At rest position the electret-membrane divides symmetrically the distance between two rigid condenser plates (Fig.1). The sound-pressure difference outside the perforated condenser-plates translates the metalless membrane, which electrostatically influences a charge on the condenser-plates. Through the symmetry of the construction and of the electric forces which act on the membrane, the microphone-response is correct both in amplitude and in phase (Fig.2) over a frequency range of more than four octaves. Thus the influence-microphone is the first gradient-microphone suitable for the construction of an acoustic wattmeter in the most interesting frequency range of about 50 Hz to 1000 Hz (1).

The presented phase characteristic is measured in reference to a 1/2" condenser-type pressure-microphone. The diameter of the influence-microphone was 20 mm. The relatively high sensitivity of 2 mV per N/m² of this microfone opens the way for the construction of microphones of half the diameter or less with still a sufficient sensitivity. Therefore it is possible to enlarge the frequency range of correct phase response.

The author owes a debt of thanks to the firm, Georg Neumann, Berlin for the construction of the first influence-microphone.

Reference: H. Ising, DAGA-Tagung Stuttgart (1972)

Vergleich einiger Realisierungsm. für ein ak. Leistungsmessgerät

Fig. 1 schematic presentation of the influence-microphone

Fig. 2 phase angle as function of frequency
EIGHTH INTERNATIONAL CONGRESS ON ACOUSTICS, LONDON 1974

A UNIFIED INSTRUMENTATION SYSTEM FOR ACOUSTIC DATA ACQUISITION

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INTRODUCTION

Acoustic measurements encompass such a broad range of requirements that a single measurement system, capable of simultaneously satisfying even most of these requirements, has not been commercially available. Extension of these measurements to large field programs, and perhaps including the generally omitted infrasonic range, imposes additional requirements on an already complex system. A system readily adaptable to a wide variety of configurations but maintaining a unified concept has been developed and is the subject of this paper.

DISCUSSION

A unified system should be readily adaptable to several common sized capacitor microphones (1/8", 1/4", 1/2" and 1"), capable of a minimum frequency response spanning 0.03 Hz to 100 KHz, useable with long cables of various types, remotely tuneable and relatively insensitive to environmental restrictions. The supporting electronics should possess a selectable bandpass, gain, calibration, tuning, metering and gain logging circuitry such that one basic system can be configured easily for a wide variety of tasks.

The system consists of two basic units, a converter where different microphone sizes are accommodated and one of two pressure ranges is selected and, at the far end of the connecting cables, signal conditioning electronics which contain the necessary power, filtering, tuning, gain ranging, gain logging and metering circuitry. Electrical tuning and a low frequency servo loop for automatic tracking of the tuning provide a convenient means for electrical calibration of the system.

STATUS

The unified system developed in our laboratory, although developed mainly with the 1/2" microphone configuration, does provide the features described. The system is electrically tuned and calibrated and has automatic frequency tracking; has successfully tracked through large temperature excursions input from a heat gun; has shown no detrimental effects caused by cables up to 3000 feet in length; has exhibited a dynamic range in excess of 70 db and can be ranged easily (50-150 db or 80-160 db); has a selectable signal bandpass and low-pass control filtering; and provides a voltage proportional to gain switch position for automatic gain logging.
VIBRATION AND SOUND RADIATION OF LOUDSPEAKER CONES

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INTRODUCTION

Below the frequency at which axisymmetric standing waves appear on the loudspeaker cone ("cone break-up"), the sound radiation can be calculated easily (1). Above that frequency, however, radiation calculations are very complex and none have been reported until now. In the past only a few experimental investigations have been carried out (2, 3).

THEORETICAL ANALYSIS

The forced axisymmetric mechanical cone behaviour was studied by numerically solving twelve simultaneous differential equations. This was done for a great number of frequencies and for various cones. The radiated sound pressure, directivity diagram and sound power were calculated by numerical integration of the transverse wave amplitude over the cone surface (Kirchhoff approximation).

EXPERIMENTS

To verify the above calculations mechanical and acoustical measurements were carried out on a few loudspeaker cones. The mechanical vibrations were visualized holographically and the voice-coil velocity was recorded as a function of frequency.

The acoustical measurements consisted in the standard recording of axial sound-pressure and sound-power response, in addition to directivity diagrams.

CONCLUSION

The experimental results are in good agreement with the calculations. It is shown that the upper limit of the loudspeaker sound-pressure response can be predicted relatively easily. Further calculations and experiments show that the rough shape of the entire sound-pressure and sound-power response can be predicted theoretically (F. J. M. Frankort, Philips Res. Repts. Suppl., to be published).

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A DIGITAL MEAN SQUARE MEASURING TECHNIQUE APPLICABLE
TO EQUIVALENT SOUND LEVEL, NOISE POLLUTION LEVEL AND
OTHER MEASUREMENTS

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INTRODUCTION
This contribution describes an algorithm for finding the mean square
value of a time function by a simple digital technique. It is useful
when a small computer is used in the measuring system. It also allows
fairly coarse quantisation of sample values including logarithmic
analogue-to-digital conversion. The algorithm is derived from a special
form of the statistical average form of the mean square (1).

THE ALGORITHM (2)

The measuring system contains a sampling element which takes a large
number of samples, M, from the time function, x(t) at instants t_m.
Each sample is quantised by an analogue-to-digital converter which
gives an integer, l_m, specifying the quantising interval that the
sample lies in. The quantiser thresholds are X_i where i=0,...,N.
The value of l_m is such that

\[ X_i \leq x(t_m) \leq X_{i+1} \]

It is necessary to have a table of constants stored in memory.
The table is constructed by finding an integration formula which has
abscissas equal to the amplitude thresholds of the quantiser. Let its
weights be w_i (i=0,...,N). The stored table has N+1 entries given by:

\[ S(r) = \sum_{i=0}^{r} w_i X_i \quad r=0,...,N \quad (1) \]

When a sample has been quantised the value of l_m is used to
specify a particular entry in the stored table which is to be added
to an accumulator according to:

\[ x^2 = \sum_{m=0}^{M} S(l_m) \quad (2) \]

This Technique is particularly useful when a relatively small
number of quantising intervals, say between 8 and 64 is to be used.
Logarithmic quantisation has been used to cover a wide dynamic range
without making N excessive while being accurate for noise measurement.

REFERENCES

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IEE Conf.Publication No 103 (1973) 120.
FAST STATISTICAL ANALYSIS OF SOUND

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INTRODUCTION

Statistical analysis of speech is an actual subject of research, especially in order to determine conversational speech loudness.

EXPERIMENTAL

We used an electronic statistical analyser with 16 levels spaced of 3 dB steps and sampling frequency range from 1 Hz to 10 kHz. The averaging time could be varied from 1 s to 10 µs (1).

In this study measurements were made by reading phrases, words and sets of syllables, at different levels and speed.

The same analysis were repeated with different sampling rate and averaging time.

RESULTS

According to the different values used for sampling frequency and averaging time, we obtained level cumulative distributions approximating logarithmic functions, with different slopes varying according to sample and speaking rate.

Presently we are still working out on some aspects of these experimental results.

REFERENCES

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(2) White S.D., Dunn H.K. - Statistical measurements on conversational speech - J.A.S.A. (1940), 11, 278.
The acoustic radar technique introduced by L.G. McAllister /1/ and C.G. Little /2/ for the remote probing of the thermal and wind structure of the lower atmosphere is limited in its application by environmental noise, by wind noise at the receiver antenna and by the thermal noise of the receiver circuit. Theoretical and experimental investigations have been conducted to minimize the noise by optimizing the electrical, mechanical and geometrical characteristics of a transceiver system operating either in a monostatic or a multistatic mode.

The system designed for a carrier frequency of about 2 kHz employs a horn-driver unit (rated at 100 W electrical power in the transmitting mode and operating with a maximum of 20 percent acoustic efficiency) which is connected to a cone-paraboloid antenna equipped with additional components for the suppression of side lobes and wind noise. Components investigated for the optimization of the antenna as the most sensitive element of the acoustic radar technique include delay lines in the horn exit, acoustical lenses providing surfaces for low-pressure fluctuations due to wind, and electromechanical transducers coupled to acoustic resonators at the throat of the horn. Except for a low-noise preamplifier (10nV/√Hz), standard electro-acoustical instrumentation is used both on the source and on the receiver side of the antenna system. Gating of pulses and data processing are effected according to the experience gained with acoustic radar systems at the Wave Propagation Laboratory in Boulder, Col. /3/. The operating characteristics of the system, its calibration and preliminary results of field experiments will be reported. (Work sponsored by the German Ministry of Defense)

References


CONTINUOUS AIR STREAM GENERATED BY LOUDSPEAKERS AT LOW FREQUENCY

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A plane vibrating surface generates into the air, under given conditions, a continuous air stream overlapping the acoustic wave. This phenomenon has been particularly investigated in case of a rigid piston and of a vibrating sphere: investigating the phenomenon of a vibrating disk, one can extend the problem to a loudspeaker in the range in which it operates like a piston.

Some quantitative elements have been measured in case of electromagnetic loudspeakers, in which the membrane is a non rigid cone, with particularly boundary conditions.

This preliminary study tends to relate the continuous air stream with other electroacoustic parameters.

Fig. 1 - Velocity of the continuous air stream produced by the continuous air flow a I S. versus velocity of the
has been detected with a high centre of its membrane (Rela-
sensitivity, hot wire anemometer values).

tre: for rather large loudspeakers, in proximity of the low-
est frequency limit and for power approximating the nomi-

Fig. 2 - Velocity of the continuous stream.

Number of measurements has been made in order to investi-

gate the velocity field of various loudspeakers, at diffe-

rent frequencies, and for given frequencies along specified di-

rections. The figures report.

A model will be nuous air stream versus the furtherly investigated, operating frequency: a) in the centre,

ing in a mean that allow to the module, b) by the frame, c) in the

ualize the continuous stream. centre of the plane of a I S.
Selenium and tellurium which show very high sensitivity for mechano-electrical transformation have not been practically applied, because of the difficulty of making large single crystals. We have found that a granule microphone using an alloy composed of Se, Te and other materials is superior to the conventional carbon microphone.

Fig. 1 shows the output voltage of various granules packed into a carbon chamber of transmitter of the type 600 telephone set. Se-Te alloys have higher voltages for low dc feed current than carbon. The output power, however, is somewhat less than that of carbon. Typical impedance of the alloy microphone is 200-1000 ohms, which is more convenient to couple with transistor circuit. To deliver the same output power, the overall dc feed power is less than that for the carbon transmitter.

It has been concluded that the alloy, composed of 0-5% Sb, 20-40% Se and Te as the remainder, exhibits so far the best performance.

Second harmonics in the output signals of the alloy and the carbon granules are shown in Fig. 2. The distortion of the alloy is considerably smaller than that of carbon. Output noise is less than the level of carbon.

Fig. 3 gives the real and imaginary stiffnesses of the various granules which are measured using a vibrometer having three transducers. The alloy granules have less nonlinear stiffness than carbon, which shows that the alloy granules hardly slip with each other at their intergranular contacts. It seems the reason for the small distortion of the alloy granule microphone.

The characteristics of the present microphone are unchanged in test operations of a million times, simulating normal usage of the telephone handset. Insensitivity to humidity of the alloy is also a merit.

It is concluded that this novel device has a good performance with high fidelity, high output voltage as well as a low cost.
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MEASUREMENT OF THE RESISTIVE COMPONENT OF THE MECHANICAL IMPEDANCE OF ELECTRET MICROPHONES

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INTRODUCTION

The resistive component of the mechanical impedance of a microphone can be determined from the noise voltage generated by such a transducer. Although the generation of electrical noise by mechanical resistances in transducers has been recognized early (1), noise measurements have so far not been used for systematic investigations of such resistances. In the present paper, an outline of the method and results for electret-condenser microphones are given.

ANALYSIS

The equivalent circuit of a condenser microphone is shown in Fig. 1. The noise voltage $V_n$ and the equivalent sound pressure $P_n$ of such a transducer, taken over the frequency band from $f_1$ to $f_2$, are given by (1,2)

$$V_n/M = P_n = \frac{4kTm(r_2-f_1)/A^2}{2} \quad (1)$$

where $M$ is the sensitivity, $A$ the membrane area, and $r_m = R_m(\alpha C)^2$ the mechanical resistance of the microphone with $\alpha$ being the transducer constant (2). In determining $V_n$, the preamplifier noise (due to voltage and current noise generators) has to be measured and analytically eliminated from the total noise output.

EXPERIMENTAL RESULTS

Measurements performed with a low-noise preamplifier (3) (short-circuit noise level 0.8 nV/Hz) on typical electret microphones (2.5 cm diameter, 25 µm Teflon electret, 25 µm air gap, perforated backplate, sensitivity 2 mV/µbar), yield an almost frequency-independent noise voltage of about 4 nV/Hz, corresponding to an equivalent sound pressure level of -40 dB re 0.0002 µbar at atmospheric pressure in the range 0.5 to 5 kHz [14 dB lower than conventional condenser microphones (4)]. The measured noise voltages correspond to a value of $r_m/A = 120 \text{ g/cm}^2 \text{ sec} = 3 \mu\text{c}$ (from eqn. 1). This value combined with results obtained under vacuum conditions shows the mechanical resistance to be largely due to losses in the air gap and backplate perforations.

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INVESTIGATION OF LOUDSPEAKER ARRAYS BY NUMERICAL
METHODS

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INTRODUCTION

By theoretical or empirical methods designing a loudspeaker array is a
time-consuming and costly procedure. Consequently, most arrays follow
set familiar patterns, innovation is suppressed and new methods of
solving propagation problems appear all too infrequently. To help
overcome these deficiencies a numerical simulation of loudspeaker
arrays has been developed, which provides fast accurate analysis and
simulated laboratory testing of large dimension, acoustically and
electrically complex arrays of loudspeakers.

PROCEDURE

The simulation computes the sound pressure observed at n points in an
acoustic field generated by m loudspeakers from the basic relationship,
\[ P_{ij}(\omega) = P_i(\omega) \cdot H_{ij}(\omega), \]
where \( P_{ij} \) is the pressure at the \( j \)th observation point caused by \( P_i(\omega) \)
the sound generated by the \( i \)th loud-
speaker modified by the transfer
function \( H_{ij}(\omega) \). \( P_i(\omega) \) is determined by
the driving signal and by the sensitiv-
ity, the frequency response and the
phase response of the loudspeaker.
\( H_{ij}(\omega) \) on the other hand takes account of:
the loudspeaker directivity, orienta-
tion and position; the position of
the observation point; and the acoustic
environment. The total pressure \( P(\omega) \)
is the vector sum of all \( P_i(\omega) \)and
the propagation pattern is a
graphical display of all \( P_j(\omega) \) at the
n points on an arbitrary observation
locus.

RESULTS

The propagation pattern for a sample
array is shown. This array consists of
a nine-speaker linear column which
includes two loudspeakers with shifted
phase and reduced amplitudes. The
results agree with measurements made on
an actual column.
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CANCELLATION OF BACK RADIATION IN ACOUSTICAL ARRAYS

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INTRODUCTION

The disposition and feeding of the loudspeakers in an array with the aim of obtaining a cardioid directivity pattern in a wide frequency range is described.

THEORETICAL ANALYSIS

Let three isotropic sources be disposed in a plane (see Fig. 1). If sources 1 and 2 radiate in phase, and the radiation of source 3 is dephased with respect to 1 and 2 for

\[ \psi = (2\mu + 1) - k_3 d/2, \quad (\mu = 0, 1, 2, \ldots), \quad (k = 2\pi / \lambda), \quad (1) \]

when the amplitude of radiation of source 3 is twice that of source 1 and 2, the following directivity function is obtained:

\[ P(\alpha) = \left[ \left( \cos^2 \left( k_3 d \sin \alpha \right) - 2 \cos(k_3 d \sin \alpha) \cos \left( k_3 d (1 + \cos \alpha) \right) \right) / (1 - 2 \cos k_3 d) \right]^{1/2} \]

This function is a cardioid, and the spatial radiation is a revolution cardioid.

EXPERIMENTAL

In Fig. 2 a result obtained is shown, using a phase shifter made according to eq. (1) (d/2=8 cm), for pure tone and pink noise filtered through the third octave band of f=315 Hz. The white dots correspond with those calculated according to eq. 2.

CONCLUSIONS

Using this system, and applying the directivity multiplication principle, a given directivity, without back radiation, can be obtained, even for very low frequencies, by controlling the number of cardioid sources, its separation and the type of amplitude distribution.
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NEW METHOD OF MULTI-CHANNEL STEREO USING COMB FILTERS

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INTRODUCTION

A new technique for the duplication of audio-signal channels using digital comb filters is presented, which can realize AM stereo broadcasting, FM 4-channel broadcasting, 4-channel disc record etc. The separation of more than 10 dB between two channels which being formed by this method has been attained not changing the sound heard by the listener.

PRINCIPLE AND EXPERIMENT

This method, Comb Frequency Division Duplex, is to be used for systems which deal with the transient signals such as speech or music. Fig. 1 illustrates the method, where two pairs of comb filters are used at both input and output sides. A pair of filters is characterized by the frequency characteristics such as their passbands and stopbands are mutually different and the frequency distance between neighboring passbands or stopbands are same, which can be accomplished by the simple digital filter (1).

![Comb Frequency Division Duplex system](image)

In operation two input signals which have past through the COS or SIN typed comb filter being rejected the frequency components at the stopbands are compounded and fed in one track, at the output the compounded signal is separated and reproduced by the pair of comb filters whose passbands coincide with the input one's respectively.

The cos² and sin² typed comb filters with the frequency distance of about 100 Hz are available for stereophonic sounds transmission, giving the separation of about 10 dB. For speech, the separation of about 30 dB has been attained with satisfactory using modified cos⁴ and sin⁴ comb filters.

REFERENCES

DOUBLING MID-FREQUENCY REVERBERATION TIME WITH A REVERBERATION REINFORCEMENT SYSTEM

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INTRODUCTION

Reverberation reinforcement increases the reverberation time of an auditorium by using the roof void as a coupled reverberation chamber (1). The sound in the roof void is relayed back to the auditorium via a 3-channel amplification system. The resulting r.t. of the Renold Theatre is curve b in fig. (1), curve a being the unaided r.t. Acoustic feedback plays a part by increasing the r.t. of the roof void by a factor 1/(1-G), G being the loop power gain (2). The r.t. of curve b has proved very satisfactory for music in a 3000m³ auditorium, but a further increase was needed for dramatic purposes.

FREQUENCY SHIFTING

Lengthening the natural r.t. of our roof void proved impracticable, so the only alternative was to increase the system gain and hence the acoustic feedback. This course can lead to coloration and howling, but Schroeder (3) has used a frequency shift to overcome these problems in P.A. systems. A recent shift circuit (4) has been designed to give a performance adequate for our application. An acceptable subjective effect is obtained with a 5Hz increase on one channel, a 5Hz decrease on another, and no change on the third channel.

As may be seen from the result (fig. (1) curve c), a long r.t. is more readily obtained at middle and lower frequencies than at high frequencies, though there is a measurable increase right up to 10kHz. The level and directional distribution of reverberant sound above 1kHz has been found to be of great subjective importance and may have a bearing on optimum concert hall design.

REFERENCES

AN AUTOMATIC MULTI-CHANNEL SPEECH REPRODUCTION SYSTEM WITH CHARACTER DISPLAYS

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INTRODUCTION

Some considerations on advanced automatic Public-Address Systems (PAS) have concluded to use character displays simultaneously. A new system has been proposed, where an Automatic Multi-channel Speech Synthesizer (1) and character displays are applied to put out speeches and characters synchronously. The automatic PAS, which has access to the sense of hearing and the eyesight, informs announcements completely to the public, even if there are noisy places or there are persons who are hard of hearing, because visual information adds redundancy.

SYSTEM REQUIREMENTS

(a) Various pieces of audio-visual information should be put out simultaneously and instantaneous-
ly to many channels on line real time. (b) Sound quality should be excellent. (c) Sentences should be displayed on a CRT in the mixed form of Chinese characters (KANJI) and Japanese characters (HIRAGANA and KATAKANA) for easy recognition. (d) The interface to peripherals and external systems should be flexible to meet the user’s demands. (e) The system should offer good price/performance and high reliability.

SOUND QUALITY AND CHARACTER STORAGE

The size of dot matrix for Chinese characters is rather large one, and to make the matter worse, we have 1850 Chinese characters for daily use. Then, an economical method has been contrived, where the magnetic drums, in which voice data are previously stored, are used as character storages. As the memory size for character data is about 1/80 of that for speech data or less, the LSB of sampled speech data (8 bits/sample) has been replaced by a character data at intervals of 8 or 10 samples. An auditory test showed that reduction in sound quality caused by data replacement was negligible.

SYSTEM SET-UP

The system which meets the requirements has been set up using the speech synthesizer (1) and character displays (Fig. 1). Frequency bandwidth and pause time between vocabularies are set in 5.1 KHz and within 250 ms respectively to yield natural speeches. Characters are displayed on standard TV sets. The system is controlled by a one-chip CPU and various pieces of information can be transmitted into up to 64 communications lines at the same time within 1 sec. after receiving request signals.

REFERENCE

RADIATING SYSTEMS. SPEECH EMISSION DIRECTIVITY

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INTRODUCTION

A new method for automatic evaluation of directivity patterns of speech electroacoustical arrays is described.

THEORETICAL AND EXPERIMENTAL ANALYSIS

Normally, the directivity of electroacoustical arrays is obtained for pure tones and steady state condition. The direct application of these results to impulsive signals whose energetical spectra is time-dependent, as for speech, seems doubtful.

In a previous paper, an anamorphical relation between the rms values of radiated impulsive signals, in different azimuthal angles has been found. Based upon this anamorphical relation a new method and a electronic device for automatic computing of directivity has been developed. The electronic device evaluates analogically the directivity according to the expression

$$D(\theta) = 10 \log \left\{ \int_{t_0}^{t_0+T} p^2(\theta, t) dt / \int_{t_0}^{t_0+T} p^2(0, t) dt \right\}$$

where the incremental rotated angle in the time interval T is assumed to be so small that can be considered as constant. The device performance includes the dynamic range of both the directivity pattern and the speech. This procedure has been applied to a great number of arrays and the results analyzed and compared with pure tone directivity. (Fig 1)

CONCLUSIONS

The possibility and suitableness of evaluation and automatic recording of the directivity patterns of sound systems for actual signals has been demonstrated. The feasibility of applying this procedure to related situations (speech privacy, acoustical propagation in dissipative media among others) is contemplated.
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CHARACTERISTICS OF A PARAMETRIC EARPHONE REPRODUCTION SYSTEM

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INTRODUCTION

A two-channel stereophonic sound reproduction system incorporating an artificial head and earphones exhibit high fidelity with respect to the position of the apparent sound source (PASS) (1), if the listener does not move his head. But in contrast to natural hearing, the PASS will follow the listener's head movements. To reestablish the PASS-constancy in case of headrotations around the vertical axis, it was proposed, to distort the electrical signals in both channels in accordance to the actual azimuth of the listener's head using a parametric system having the transferfunction (2)

\[ b(\omega, \beta) = (1 + im\beta) \exp j\omega(k+n\beta) \]

\[ \beta = \text{azimuth of listener's head} \]
\[ \omega = \text{angular signal frequency} \]
\[ m = \text{amplification coefficient} \]
\[ n = \text{delay coefficient} \]
\[ k = \text{bias delay} \]

In determining the operation of the system, observers had to judge the PASS under several listening conditions.

EXPERIMENTS

A) Speech signals from different front directions were recorded by using an artificial head. Observers had to point an arrow to the PASS of the reproduced signal. The introduction of the control system produced higher reliability of PASS judgements than in the case of using a constant parameter system (reduced stand. dev.).

B) Observers had to determine the PASS of real sound sources (speech, -30° ≤ β ≤ +30°) and of control system reproductions of the same signals. This was done by making the observer to turn his head towards the PASS. The head position was permanently recorded. The results indicate, that in both listening conditions the PASS were judged equally.

CONCLUSIONS

Incorporating an idealized control system providing a linear relationship between the observer's azimuth and the channel-delay and amplification results in an improvement in PASS reproductivity of headphone stereophonic systems. Furthermore the PASS is not affected by rotation of the listener's head around a vertical axis.

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(2) G. Boeger, U. Kaps, Fortschritte d. Akustik Daga '73 (1973) 398;
Ein elektroakustisches System ist bei den Frequenzen instabil, bei denen das rückgekoppelte Signal in Phase mit der Eingangsgröße und die Gesamtverstärkung größer als Eins ist. Eine Möglichkeit zur Erschwerung der elektroakustischen Rückkopplung besteht darin, das rückgekoppelte Signal in der Phase zu verändern, sei es periodisch /1/, stochastisch oder linear mit der Zeit /2/.

In diesem Bericht wird ein Gerät beschrieben, das es gestattet, eine Phasenmodulation der oben beschriebenen Art vorzunehmen. Das Übertragungsverhalten dieses linearen, zeitveränderlichen Systems wird angegeben.

Das Gerät wurde in einem Rückkopplungssystem eingesetzt, in dem vorerst der Raum durch ein Kundt'sches Rohr ersetzt ist, um die theoretisch ermittelte Stabilitätsgrenze an Hand dieses in allen Einzelheiten bekannten Schallfeldes zu überprüfen.

Literatur:
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ÉTUDES SUR LA PERCEPTION DES PRODUITS SUPPLÉMENTAIRES DE NONLINAIRITÉ INTRODUITS PAR UN APPAREIL RÉALISANT LA FONCTION $y = ax + bx^2$

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INTRODUCTION

La réponse d'un appareil servant à introduire sur un signal sonore des distorsions du 2ème ordre, s'écarte de l'idéal $y = ax + bx^2$ en raison de l'introduction simultanée de composantes non désirables. On pourrait donc considérer, la non-possibilité de perception de ces composantes.

EXPERIENCES

Nous travaillons en perception auditive sur deux essais constitués par un couple de deux signaux consécutifs — l'essai idéal: constituant du signal original et de sa distorsion du 2ème degré $s_1 + s_2$. L'essai déformé: comportant, outre le signal global précédent, des déformations supplémentaires: $s_1 + s_2 + s_3$. La détection des différences perceptibles sur deux signaux d'un même couple est effectuée par 70 sujets choisis. Les résultats, obtenus après élimination par conversion (1) du paramètre "erreur de décision" , sont donnés fig.1. Pour un niveau d'entrée de 9 dB et 2ème harm=25% les distorsions supplémentaires mesurables sont: 3ème harm=0,5%, 4ème harm=0,16% et 5ème=0,06%.

CONCLUSIONS

La méthode utilisée permet de déterminer les limites de perception des composantes additionnelles d'une part, ainsi que la dynamique à admettre en entrée d'un appareil pour n'obtenir en sortie que la distorsion désirée.

REFERENCES

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DIMENSION ANALYSIS AS A METHOD FOR SUBJECTIVE ASSESSMENT OF SOUND QUALITY OF SOUND REPRODUCING SYSTEMS

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In "listening tests" the persons acting as subjects are expected to make some kind of judgements about the sound reproduction of certain systems (loudspeakers, earphones etc.). In order to minimize or control the influence of other factors than the systems themselves a number of critical methodological questions has to be considered, such as how to choose a reference for the judgements, how to choose the stimulus material (music, speech etc.), how to choose subjects, how to design the specific judgement method, how to estimate the reliability of the data, how to treat the data in a statistically adequate manner, and so on. Questions like these are, however, seldom discussed in a more detailed manner.

A special question of considerable interest is the use of multivariate techniques developed within psychology like factor analysis (FA) and multidimensional scaling (MDS). FA may be applied to data from evaluative ratings or ratings on various adjective scales ("semantic differential") and MDS to ratings of perceived similarity between different systems. These analyses aim at revealing the perceptual dimensions underlying the ratings, that is, which dimensions may be involved in perceived sound quality of different systems. Experiments made by the authors (1,2) suggest dimensions as "clearness/distinctness", "brightness-darkness", "volume", "loudness", "hardness-softness", "room impression", and others. There are many problems connected with the use of these techniques, but applied with due caution and supplemented with other information (for instance, "free verbal descriptions") they may be convenient for establishing some probably essential dimensions of perceived sound quality. Such dimensions may be used for evaluative judgements of various systems and it might be possible to relate them to various physical characteristics of the soundreproducing systems.

REFERENCES
HERSTELLUNG BELIEBIGER WANDIMPEDANZEN MIT ELEKTROAKUSTISCHEN MITTELN

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EINFÜHRUNG

In eindimensionalen Schallfeldern in Kanälen bei tiefen Frequenzen ist die Schallausbreitung durch die örtliche Verteilung der Wandimpedanzen zu beeinflussen.

In einigen Fällen sehr übersichtlicher Schallquellen ist durch Veränderung der angekoppelten Impedanz die Schallabstrahlung zu verringern.

VERSUCHSDURCHFÜHRUNG

In einem Kanal wird vor einem reflexionsarmen Übergangsstück ein Mikrofon angeordnet, dessen Signal mit einstellbarer Verzögerung und Verstärkung mit entgegengesetzter Polarität dem ursprünglichen Signal überlagert wird. Dadurch ergibt sich resultierend eine Veränderung der Wandimpedanz, die durch Messungen der Eingangsplemadanz nachweisbar ist und sich in der verringerten Schallausbreitung über den Lautsprecher hinaus auswirkt.

In einer anderen Versuchsreihe wird bei einem Modell-Radialventilator durch einen elektroakustisch erzeugten Dipol die Abstrahlung des Schaufeltones unterdrückt.
ACOUSTIC DETECTION AND LOCATION OF OBSTRUCTIONS IN PIPES

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APPLICATION

Where large numbers of tubes of lengths up to several hundred feet are installed in industrial plant, it is essential that these tubes should be checked before commissioning for any foreign material that may have inadvertently entered during construction.

METHOD

A suitable method of detecting such obstructions is by means of a pulse echo technique using sound waves in the frequency range up to 10 kHz. Sound travels easily along uniform bore pipes and is affected little by bends. In order to obtain reflections from objects, the wavelength of the transmitted signal should be comparable to or smaller than the diameter of the obstruction, however the higher the frequency of the transmitted wave, the greater is the attenuation of the signal in the tube. In a 5 cm diameter tube, for example, the practical limit for the detection of an object of 1 cm diameter is about 120 metres for an initial sound pulse of 90 dB. This is probably sufficient for most industrial purposes.

In addition to the location of foreign material, considerable other information about the tube is also obtained. Since low frequencies are propagated with little attenuation, reflections from the end of the tube are easily seen. Furthermore, should a blockage occur at the end of the tube it may be detected from the phase difference that exists between the reflections from open and closed ends. The location of holes is also possible since they act as acoustic filters, reflecting the low frequency signals. For our prototype instrument, holes of about 1.5 mm are the minimum that can be detected.

A detailed quantitative examination of the echoes has not yet been made but the qualitative information given is ample for general inspection of tubes.

EQUIPMENT

The prototype instrument has been designed for the inspection of tubes in boiler plant. The transmitter consists of a small loudspeaker pulsed by a capacitor discharge, and the echoes are received by a small conventional microphone close to the loudspeaker. After passing through a swept gain amplifier (to compensate for the increase in attenuation with distance) the echoes are displayed on a storage cathode ray tube. Field trials have demonstrated the simplicity and usefulness of the system and the ability to operate successfully under conditions of industrial noise.
APPLICATION OF ACOUSTICS AS A REMOTE SENSING TOOL

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Experiments during the past five years have demonstrated that the acoustic wave is a powerful tool for remotely sensing the lower regions of the earth's atmosphere. The atmospheric boundary layer is rich with turbulent wind and temperature fluctuations which act to scatter acoustic energy. These omnipresent fluctuations, and the strong interaction of sound waves and the medium, insure that useable tracers of atmospheric motion will be present a large percentage of the time.

The acoustic echo sounder, a sonar like device, capitalizes on the turbulent scattering properties of the atmosphere by transmitting an acoustic pulse and then "listening" for the scattered returns. Early work with a simple monostatic version of the acoustic sounder has already led to a better understanding of boundary layer dynamics. The detailed time-height histories of temperature inversions and convective elements have been used to study problems associated with wave dynamics and boundary layer stability.

A major breakthrough, the development of frequency tracking capabilities for the acoustic sounder, has made it possible to employ the Doppler principle for deriving quantitative wind measurements along profiles up to heights of 1 km. This technique has greatly extended the usefulness of the acoustic sounder for making remote measurements of such boundary layer properties as wave motions, the velocity structure of thermal plumes and momentum flux.

Application of acoustic Doppler systems for measuring low level wind shear at airports is already being tested and could become a fully operational system within the next few years.

This paper describes the experiments mentioned above and summarizes the results of work to date. It further speculates on the potential future application of acoustic systems for land-sea interaction studies, urban meteorology, and the possibility of deriving additional quantitative data from the scattered acoustic signal.
EINLEITUNG


WANDLERPRINZIP (Fig.1)

Eine Aluminiumträgerplatte (1) ist mit einer beidseitig kontaktierten Piezokeramik (2) aus Bleizirkon-Titanat verbunden. Der Schalldruck p biegt die Trägerplatte und erzeugt verstärkte Zug- und Druckspannungen G in der Piezorschicht. Durch Deformationen im polarisierten Kristallgitter entstehen Verschiebungen der molekularen Ladungen, die an den Elektroden (3) als Spannungen abgegriffen werden.

ÜBERTRAGUNGSFAKTOR

\[ U_{\text{Leerlauf}} = \frac{d}{\varepsilon \cdot \varepsilon_0} \cdot \frac{\sigma}{p} \cdot y = 4 \cdot 10^{-3} \text{ V m}^2 \text{N} \]

\begin{align*}
  d &= 3 \cdot 10^{-10} \text{ m/V} \\
  \varepsilon &= 1500 \\
  \sigma / p &= 1300 \\
  y &= 150 \text{ µm}
\end{align*}

Die Grundresonanz der Biegeplatte (f₀=1,6kHz; Resonanzüberhöhung ca. 25 dB) wird durch einen Helmholtzresonator absorbiert (m₁ c₁ r₁).

Weitere Korrekturmaßnahmen (akustisches Netzwerk) ergeben einen ausgeglückten Frequenzgang des kompletten Wandlers.

SCHRIFTTUM

E. Martin und E. Müller, Siemens Z. 46 (1972) II. 4
Recently digital technique has been introduced to the field of electroacoustics. A digital tape recorder has been already in use as the master recorder of disc recording. We have made 12bit digital recording and control system for some experiments and analyses. This system has the function of programmable fade-in and fade-out, mixing and artificial reverberation in digital stage.

A block diagram of the system is shown in Fig.1. An analogue input signal is sampled at 48kHz and held, the samples are led to the 12bit analogue to digital converter.

For the tape recording, digital signals are led to three staggered 4-channel heads through the modulator, and recorded in parallel on the 14-track magnetic tape (12 for data, 1 for clock and 1 for direct recording). To eliminate jitter, demodulated signals are once stored in 49152 word shift register and read by jitterless clock pulse. When the error detector detects the error like drop out, the datum immediately before is held.

As for shorter signals, for example impulse responses of audio equipments, the 49152 word register can be used for recording and exact reproduction about one second is obtained with no jitter or error. Data in the register can be read by any clock frequency as we like, even word by word.

The function of fade-in and fade-out consists of the control signal generator and the digital adder or subtracter, and enables fade-in and out with any programmed slope at any time. The artificial reverberation is generated in digital stage by the five series Schroeder's 'colorless' circuit.

Fig.1 Block diagram of recording and control system

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THE DEVELOPMENT OF ULTRASONIC LENS TECHNOLOGY

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INTRODUCTION

Ultrasonic lenses fabricated from homogeneous solids, liquids, and gases were designed and tested for both underwater and in-air applications. Diffraction limited lenses corrected for temperature variations in the medium were made possible by the wide range of refractive indices available. Lenses were designed for $10 < ka > 1000$ and tested in the frequency range of 20 kHz to 3 MHz. This work shows that ultrasonic counterparts to multielement optical systems are both feasible and practical.

THEORY

For lens performance evaluation, a ray-phasor model of a generalized lens system was developed for the computer based on the Fresnel-Kirchhoff diffraction integral. Both axial and off-axis diffraction patterns are computed. In the model, individual rays are traced through the refracting elements with phase and amplitude updated at each surface. All rays passing through the system are summed at the field point in the image surface. Very few rays are required for convergence of the integral and a diffraction pattern containing several minor lobes can be computed in several seconds. With this technique of evaluation, conventional optical guides to good lens design such as blur circle diameter and aberration figures may be interpreted in terms of diffraction pattern characteristics. In addition, aperture apodization schemes may be easily incorporated into this model.

EXPERIMENTAL

A study of the ultrasonic parameters of hundreds of solids and liquids was undertaken to identify materials suitable for lens fabrication. These materials and their influence on lens design and performance will be discussed. Numerous lenses for various experimental applications were designed according to good optical practices and diffraction patterns were measured with excellent agreement with theoretical results. In this report, a solid thin lens doublet, a wide angle liquid/solid triplet, and a gas lens singlet are described with theoretical and experimental comparisons.
RESOLVING POWER OF ULTRASONIC RECEIVERS USED IN IMAGING AND HOLOGRAPHY

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INTRODUCTION

The ability of any ultrasonic imaging system to resolve detail has sometimes been discussed in terms of the aperture of the equivalent optical system. As a variety of methods have been used in imaging, the definition of the term effective aperture has to be related to the method being used (e.g. the use of a concave mirror with a Sokolov tube cannot be directly compared to a scanned receiver and transmitter holographic system). However in several presently used systems the maximum effective aperture is determined by the receiving system itself. Factors leading to this conclusion are discussed.

DISCUSSION

The aperture of a Sokolov tube using a tuned piezo-electric receiving element is determined by three factors: the critical angle of incidence of the incoming sound, the "Q factor" of the receiver, and the maximum acceptable size of the electron scanned area of the piezo-electric element. Consideration of these effects, particularly the latter two, leads to a criterion for the best choice of receiver material and consequent determination of the maximum useful aperture of any associated acousto-optical systems.

The Pohlman cell in its various forms appears to have a maximum useful aperture limitation determined by the thickness of the sensitive medium and the refractive index of the material admitting the sound (assuming small reflection from the rear enclosing surface). Liquid surface detectors have a limitation in their maximum aperture due to the decline in sensitivity of the receiver as attempts are made to increase the maximum spatial frequency of the hologram usually produced. Similar limitations apply to direct viewing.

The dynamic ripple detector limitation has been explained satisfactorily by Korpel et al. (1)

Scanned holographic systems appear to have a performance which is relatively independent of the materials in the receiver with limitations which arise solely from geometric effects associated with the arrangement of the apparatus.

CONCLUSION

A comparison of the resolution which might be expected from the different types of receivers is obtained from the detailed results for typical examples of each.

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THEORY OF SPACE-TIME PROCESSING AND ITS APPLICATION TO THE DESIGN OF LINE ARRAYS

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INTRODUCTION

Representation of line arrays in terms of their plane wave aperture transfer function provides useful insight into the selection and interpretation of array design criteria. This concept, introduced several years ago by Bracewell (1), coupled with two new algorithms (2, 3) permits development of a generalized approach to line array processing.

MATHEMATICAL BACKGROUND

A line array along the z-axis is modeled by its spatially dependent frequency response W(f, z). The array output Y(ν) is defined for all frequencies by

\[ Y(ν) = \int_{-∞}^{∞} \Omega_p(f, ν) P(f, ν) df \]  

(1)

where \( \Omega_p(f, ν) \), called the plane wave aperture transfer function, is the spatial Fourier transform of W(f, z) and P(f, ν) is a representation of the incident field in frequency (f) wave number (ν) space. For a plane wave propagating with velocity \( c (ν = \frac{ν_0}{f/c} \cos \theta_0) \), the output

\[ Y(f) = \Omega_p(f, ν_0) P_0(f) \]  

(2)

is a filtered version of the spectrum of the field at \( z = 0 \) according to values defined by \( \Omega(f, ν) \) along the radial line shown in Figure 1. For a spherical wave with radius of curvature \( r_0 \),

\[ Y(f) = \Omega_s(f, ν) P_0(f) \]  

(3)

where

\[ \Omega_s(f, ν) = \Omega_p(f, ν) \cdot \frac{\exp \left[ jπ/4 - (2πν c^2/2)/(2f r_0)^{1/2} \right]}{(2f r_0^{1/2})^{1/2}} \]  

(4)

the spherical wave transfer function is obtained from the plane wave aperture transfer function by convolution (in ν) with a linear chirp.

REALIZATIONS

A particularly important array geometry consists of N point sensors equally spaced at increments d along the z-axis such that

\[ W(f, z) = \sum_{n=0}^{N-1} W_n(f) \delta(tz-nd); \Omega_p(f, ν) = \sum_{n=0}^{N-1} W_n(f) e^{-2πjndν} \]  

(5)

In this paper, three design criteria (Delph-Chberyshoff, constant beamwidth, and maximum array gain) are analyzed and interpreted in terms of the plane wave aperture transfer function. It is shown that by proper choice of \( W_n(f) \), the array may be "focused" to sources in the vicinity of \( r_0 \) so that the overall response is identical to that for plane waves. A very general structure for realizing this entire set of processing functions combines the speed of the Fast Fourier Transform (FFT) algorithm (2) with the versatility of the chirp Z-Transform (CZT) algorithm (3). This structure capitalizes on the compact representation of propagating signals and sensor operators in f-ν space providing a new approach to the design of sonar array processors.

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WAVEGUIDES FOR ULTRASONIC VISUALISATION

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INTRODUCTION

Waveguides consisting of steel rods or tubes filled with a suitable liquid are normally used for getting acoustic signals in and out of hot sodium in fast reactors, for applications such as boiling noise detection and fairly simple object location systems. There is a need for a more general viewing system which can be used under sodium in a reactor and here the demands on the waveguide in terms of signal distortion and noise levels become much more severe.

SOLID WAVEGUIDES

Trailing pulses cause severe attenuation of the primary pulse and also complicate the signal from the object being examined. The attenuation coefficient of the primary pulse decreases exponentially with increasing waveguide radius, approaching the bulk attenuation coefficient at large radii. There are indications that the mechanism limiting the loss process is the rate of radial transfer of acoustic energy from the centre of the rod to the circumference. A screw thread on the outside of the rod cuts down the trailing pulses and reduces the attenuation coefficient to about one third of that of a solid rod.

LIQUID FILLED WAVEGUIDES

In a fast reactor, diaphragms must be fitted at the ends of the waveguide to keep the liquid in the guide separate from the sodium coolant. This contributes significantly to the attenuation and also lengthens the pulse by multiple reflections across the diaphragms. However, our results show that, provided any surface coupling problems are overcome, liquid filled waveguides offer better performance than solid ones for lengths (10 m) required in a fast reactor.

MULTI-WIRE WAVEGUIDES

A further approach is to use a bundle of fine wires as a waveguide(1). If the diameter of the wires is less than about λ/4, then there is only one mode of propagation at a given frequency, which travels at the bar velocity \( v_p = \left( \frac{V}{\rho} \right)^{\frac{1}{2}} \), with no trailing pulses. This type of waveguide does not distort the pulse which is of great advantage for a visualisation system. The attenuation is about the same as that of somewhat larger solid rods. It should be possible to improve this by further increasing the number of wires, by optimising the wire diameter and by improved design of the ends of the guide.

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MEDIUM-INDUCED SIGNAL DISTORTION: AN IMPACT ON SIGNAL PROCESSING

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INTRODUCTION

This paper develops the structure of an optimum receiver for which both the array configuration and the processor are "matched" to the medium distorted signal. For a "conventional" preformed beam system the apertures are described in terms of their beampatterns. This implies that the signal processing portion of the system is "separable" from the array design.

However, the separation of spatial sampling and processing is not fully justified. When the aperture is treated as an integral part of the receiver, the concept of beamforming must be viewed in a somewhat different manner than for the conventional preformed beam system.

APPROACH

The use of field theoretic transmission loss models permits an explicit representation of the amplitude and phase characteristics of the signal as seen by each element in the array. We examine the detailed effects of multipath induced signal fading in the horizontal and vertical dimensions for a narrowband signal. The time and frequency dispersion effects and the related temporal and spatial correlation of the signal are determined as a function of receiver depth for a selected environment and geometry of practical interest.

The problems associated with array gain degradation caused by spatial coherence losses in the vertical dimension are examined for conventional delay-sum beamforming and a beamforming technique based on normal mode matching.

715
EXTENSION OF THE WEAK-SIGNAL ENHANCEMENT TECHNIQUE TO SCANNED ACOUSTICAL HOLOGRAPHY

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INTRODUCTION

The holographic enhancement technique of R.K.Mueller et al. (1), is extended to scanned holography with the introduction of two extra steps. This process is theoretically investigated and is used to process in real-time acoustic holograms recorded with mechanical scanners.

THEORETICAL ANALYSIS

The original method of R.K.Mueller et al. involves four steps. In the last one, the original field is multiplied by its suitably filtered intensity, reconstructing an enhanced weak part of the field.

When the four steps are implemented with analog electronic processing, the result of the multiplication of step 4 leads to a time varying signal, eqn.(1), rather than to a time invariant complex amplitude (s is the object signal, $s_1$ and $s_2$ its strong and weak components).

$$\left[|s_1|^2 + |s_2|^2\right]s_2 + 2|s_2|^2 s_1 + s_1^*s_2^* + s_1^*s_2$$

(1)

In order to convert this R.F. signal into an optical field, it is recorded holographically by mixing with a reference signal $R$ (step 5). The hologram can then reconstruct in standard manner, the waves whose complex amplitudes are shown in eqn. (2) (step 6).

$$R^2 e^{-j k_0 x}\left[|s_1|^2 + |s_2|^2\right]S_2 + 2|S_2|^2 S_1 + S_1^*S_2^* + S_1^*S_2$$

(2)

($k_0 = 2\pi/\lambda\sin\theta$, $\theta$ is the effective offset of the reference, while $S$, $S_1$, $S_2$ are the complex amplitudes corresponding to $s$, $s_1$ and $s_2$).

EXPERIMENTAL RESULTS

A scanner described earlier(2) has been used to record at 2 MHz, holograms of a gun laid on a mirror (Fig.1). Holographic fringes on the gun, appears only when the described method is used. Optical reconstruction are shown in Fig.2; (a) is obtained from the gun laid on anechoic supports and (b) is obtained from the gun laid on the mirror and the processed hologram 1b.

REFERENCES

Surface acoustic wave (SAW) delay lines have been employed to raster scan and simultaneously focus acoustic arrays (1). Via multipliers, the outputs of the array elements set the tap weights of a linear transversal filter (Fig. 1). The tap weights, i.e. the element outputs, are convolurally scanned by injecting a function \( h(t) \) into one terminal of the transversal filter. The output at the other terminal of the transversal filter is the generalized time-variant spatial convolution of \( h(t) \) with the spatial input \( u(x,t) \). The exact form of the convolution depends on which terminal is used for injection of \( h(t) \). For temporal frequencies such that \( u(x,t) \) is time invariant during the delay line propagation time, both possible convolutions reduce to \( [u * h](t) \), the convolution for a time invariant system. This technique is powerful and versatile, since much of the existing technology for transversal filtering (2) can be extrapolated to convolurally scanned arrays.

An analogy existing between lens systems and multiplication and convolutional filtering by 'chirp' functions of the form \( C(t,x) = \exp(-i\pi f x^2 / \lambda) \) is shown in Fig. 2, where \( \alpha \) is the 'chirp' rate and \( \lambda \) is the wave length (3). By this analogy an image can be formed of an object at a distance \( d_1 \) in front of the array by setting \( h(t) = \delta(t) \), the delta function, multiplying the scanned array output by the 'chirp', \( C(t,-1/f) \), and then convolving the resulting output with the 'chirp', \( C(t,1/d_2) \) where \( 1/d_1 + 1/d_2 = 1/f \) and \( f \) is the focal length of the analogous lens. Beamforming is obtained when \( d_2 = f \). At a fixed frequency, imaging is more easily accomplished by setting \( h(t) = C(t,-1/d_2) \). The image then appears directly at the scanned array output.

Spatial Fourier transforms can also be implemented by convolutional scanning. Upon setting \( d_1 = d_2 = f \), \( \alpha(x) \) becomes a frequency scaled, phase shifted version of the Fourier transform of \( i(x) \). Thus, if in Fig. 1, the 'chirp', \( C(t,1/f) \), is applied as the scanning function, \( h(t) \), and if the scanned array output is multiplied by the complex conjugate of \( h(t) \) and then again convolved with \( h(t) \), the resulting output is the spatial Fourier transform of \( u(x,t) \). Although continuous representations have been used in the preceding discussion to facilitate the analogy with lens systems, the convolurally scanned array of Fig. 1 is a discrete system since the array elements and delay line taps are discrete. In this system there is no frequency dependent scaling of the output.

The preceding description has dealt with one-dimensional concepts. Multi-dimensional images, beams and Fourier transforms can be represented as the concatenation of one-dimensional components. Thus multi-dimensional arrays can be subdivided into one-dimensional line arrays, these line arrays convolurally scanned, their outputs then convolurally scanned in the next dimension and so on, to generate concatenated multi-dimensional images, beams and transforms.

Although applications of transversal filter convolutional scanning have been limited to date to acoustic arrays, the concept can be applied also to television, infrared and microwave electromagnetic arrays. In addition to SAW devices, semiconductor devices can be used for convolutional scanning as well as for post-scanning transversal filtering (4).

REFERENCES

Real-time multidimensional Fourier Analysis of acoustic fields can be achieved with convolution filters through the use of the chirp-Z transform (CZT). The CZT reduces the computation of the discrete Fourier transform (DFT) to a premultiplication by a discrete chirp, convolution with a discrete chirp, and postmultiplication by a discrete chirp. For the case of time signals, the multiplications and convolutions are performed sequentially in time on the data in real-time. For the case of spatial signals, the premultiplication is a spatial multiplication and the convolution can be performed by the same hardware that converts the spatial samples to a time sequence. Multidimensional transforms can be constructed by concatenation of one-dimensional transforms along each variable. When the convolution required for the CZT algorithm is implemented by transversal filters, hardware can be constructed which is of small size and low power consumption. Acoustic transversal filters can be employed as the convolvers, and a surface wave CZT of 32 points has been constructed which computes the DFT in about 6 μsec. When compared with a general purpose computer using an in-place computation of the DFT by means of the fast Fourier transform (FFT), and operating at a “Butterfly” rate equal to the CZT’s sample rate, the CZT implemented with a transversal filter convolver is \( \log_2 N \) times faster where \( N \) is the size of the DFT. Also, the CZT size may be arbitrary. Similar results hold in the multidimensional case. If temporal processing is included in addition to spatial processing, as is required for space-time Fourier transform processing, then there is a further increase in throughput for the CZT relative to the FFT. Thus the application of physical acoustics to the implementation of acoustic transversal filters makes possible the efficient processing of more general acoustic fields as well as a high throughput processing of temporal signals.

a. Figure 1. b. Figures 2 and 3.

REFERENCE
MINI-COMPUTERS ON-LINE IN THE "MIMI" ENVIRONMENTAL ACOUSTICS PROGRAM

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This paper describes the "Project MIMI" concept for the use of mini-computers as programmable, multi-purpose, real-time, field-based, digital data acquisition systems. The field systems are compatible with laboratory-based computers capable of batch processing and analysis of underwater acoustic signals and environmental sensor outputs. The "MIMI" study program generates long-period (one or more years), continuous time series from fixed sensor measurements of acoustic and environmental variables requiring reliable, unattended, field systems. The large quantities of data so acquired require efficient field-to-laboratory interfacing programs. These systems have been used on acoustic ranges of 1, 3, 5, 7, 42, 700, and 1200 nautical miles.

The original field system designed by Dr. T. G. Birdsall of the Cookey Electronics Laboratory, University of Michigan, has undergone several iterations of development. The newest system has a maximum throughput rate of 150,000, 10-bit samples per second. Computational demands reduce this total, and a typical system may have a real-time acquisition rate of 10,000, 10-bit samples per second. It is capable of simultaneous sampling of variables, entry of digitized values into memory, computation in background, and writing of summary outputs on magnetic tape in a format which is compatible with the laboratory processing system. This on-line field system is also capable of computing Fast Fourier Transforms (FFT) and displaying the results as a function of time on a high-speed storage oscilloscope. Under on-line CPU control, any convenient length time series of these transforms may be converted to a report-quality hard copy. Hard-copy devices, including teletype, multi-point recorders and oscilloscopes, are part of the system. A digital-analog section under CPU control drives analog "quick-look" instruments. Some of these receivers have been in the field for over two years and have produced continuous time series spanning more than one year.

The laboratory-based data processing and analysis system is of conventional configuration, including a CPU, magnetic disc, high-speed printer, digital plotter, and various magnetic tape storage devices. The utility of the system has been greatly enhanced by a superior file-management system developed in-house at the Institute for Acoustical Research and standardized data formatting which makes all of our field and laboratory systems tape-compatible and permits rapid analysis and data transfer in the laboratory. An extensive, well-known, statistical software package called "FESTSA" has been converted for use by our laboratory processing system.
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THE CONTINUOUS DISPLAY OF MOST RECENT VOICE PITCH AND AMPLITUDE PATTERNS IN REAL TIME

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INTRODUCTION

Visual display of pitch patterns provides feedback on certain acoustic features of the voice essential to the acquisition of good speech. It has been shown that a child with severe hearing impairment readily learns to use these patterns to control his voice pitch and rhythm, and suitable instrumentation for pitch display has been described (1,2).

Developing speech training programmes based on such a display and introducing these into a school are vast tasks which only the teachers can undertake effectively. Concurrently, close liaison should be maintained with the development engineer so that an instrument may be evolved from which maximum benefit will be derived. The device to be described is the result of several years of such co-operation.

METHOD USED

The display adopted is readily understood even by young deaf children. Unlike a storage oscilloscope which requires triggering and erasure - actions which present problems to the child - a continuously progressing time-base, equivalent to the rotating cathode ray tube type of display (3) is used. The long persistence screen of the latter is, however, replaced by a digital memory which stores samples of both pitch and amplitude signals. Its content is read out repetitively 80 times per second and displayed on a normal oscilloscope screen. For every new sample stored, the oldest sample in the memory, which was displayed at the extreme left hand side of the screen, is discarded. Thereafter all stored samples are advanced one position to the left thus creating a vacancy at the extreme right into which the latest sample is then placed. Since this updating action occurs at a high rate, the patterns progress across the screen smoothly.

By duplexing, two separate traces are formed on the screen and the following modes of operation become available: a) pitch and amplitude patterns are presented synchronously, b) a model utterance may be stored in stationary form on the upper trace while the child uses the lower to display his own pitch or amplitude patterns. Since present events as well as those of the preceding 3 seconds (the total memory capacity) are displayed at all times, instantaneous feedback is provided and, in addition, the relationship between the various parts of an utterance is revealed. Both traces may furthermore be stopped indefinitely for detailed study of the patterns. In addition simultaneous auditory or tactile feedback for multi-sensory stimulation is provided as well as a mirror for lipreading.

Several South African schools for the deaf find this instrument a valuable teaching aid and are introducing it into their curricula.

REFERENCES

A REAL-TIME DISPLAY OF NON-STATIONARY ACOUSTIC SPECTRA

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In the last decade, the "Fast Fourier Transformation" (FFT) has become a well known technique to calculate the spectrum from electrical signals in the acoustical domain (1). Using a fast general purpose computer and signal sampling rates in the order of 10 Kc, a real-time FFT is possible. The problem to be solved is to display instantaneously the results of the FFT in a permanent form.

It has been proposed (2) to use an electrostatic printing technique. A row of pins can charge selectively an electrostatic recording paper, which will then undergo a developing and fixing process. Since such an electrostatic technique normally does not allow to print different gray-tones one "point" of the spectrum will be represented by an assembly of e.g. 64 chargeable dots. A "point" means an area of $\Delta t \times \Delta f$, where $\Delta t$ and $\Delta f$ are defined by the duration of the time window, e.g. a Hanning window, from which one spectrum is calculated. Every $\Delta t$ a new spectrum will be recorded.

In the meantime a device has been realized using an improved digital technique but omitting any analog techniques. One possibility is to convert the digital values, which result from the FFT, directly into patterns of a 64-dot-matrix. 64 gray levels can be achieved by this linear conversion. A logarithmic conversion will be done by a simple algorithm, which has been implemented by pure digital circuits. Dynamic ranges of 12, 24 and 48 dB can be selected. Shifting of the ranges in 6 dB steps is also possible. A scrambling technique yields a random distribution of the charged dots of one "point".

The device can be used advantageously to record time varying spectra of phenomena in the frequency range 0-5 Kc. It has found a special application as "Visible Speech Recorder". The picture shows the spectrum of rectangular pulses with varying repetition rate.

REFERENCES
REAL-TIME STATISTICAL ANALYSIS OF SOUND PRESSURE LEVELS

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A method is presented to typify sound pressure levels (SPL) situations from a specific noise. The method gives out the particular set of samples out of a collective needed to obtain the probabilistic distribution of the SPL of the noise with a prefixed risk \( \alpha \) and precision \( q \).

The method is based on Simple Random Sampling (SRS) techniques with the following stages:

1) The original signal is sampled during 10 minutes interval at a rate of \( r \) sps given a collective of \( Q \) sets \( N_1, N_2, \ldots N_Q \), with \( n \) samples each such that : \( \sum_{i=1}^{Q} N_i \times n = 600 \times r \), \( Q \) and \( n \) fixed, being \( N_i \) the \( p_{rms} \) of the \( n \) samples belonging to the set \( i \).

2) First estimation of the standard deviation of \( p_{rms} \) using a pilot sample from the collective \( Q \) and then estimation of size sample, number of \( N_i \), for a given risk \( \alpha \) and precision \( q \).

3) Pseudorandom selection of those sets \( N_i \) out of the collective.

4) Actual precision computation.

5) Comparison between the actual and the desired precision, going to step 7) if the former is above the later.

6) If the actual precision \( q_\alpha \) is below the desired one \( q \), the incremental sample size is calculated followed by a pseudorandom selection of the new \( N_i \) within the collective \( N_1, \ldots N_Q \). Then this new sets \( N_i \) are added to the previous ones \( N_i \) and proceed with step 4).

7) Computation of the estimated probability distribution.

Thus for a particular noise \( R_i \), one obtains a set of parameters (standard deviation, mean, etc.) specifying the estimated probabilistic distribution of the SPL obtained with a risk \( \alpha \) and precision \( q \) from the sets \( N_1, N_2, \ldots, N_m \) out of the collective \( N_1, N_2, \ldots, N_Q \).

Traffic noise have been using to find out which are the appropriate \( N_i \) to obtain the estimated probability distribution with a prefixed risk \( \alpha \) and precision \( q \). Once the \( N_i \) are known one needs to read only these samples instead of the whole 10 min. allowing to compute the estimated probability distribution in real time.

The method is suitable to find out a typological model of different kind of noises \( R_1, \ldots, R_k \). The model would stratify noises in different types such that those belonging to the same type would be homogeneous and noises belonging to different types would be heterogeneous.
A series of experiments have been carried out to study the response to tone bursts of acoustic resonators placed at the end of a duct.

The apparatus used for this work comprises a tube 27m long and 126mm in diameter with a loudspeaker at one end, a condenser microphone let into the side of the tube at a point about half way along its length and facilities for mounting a resonator in the centre of a rigid plate at the other end. The loudspeaker was fed from an oscillator and a tone burst generator; the signal from the condenser microphone was digitised and recorded, via a buffer memory, on paper tape. Figure 1 shows a typical result.

TREATMENT OF RESULTS

The Fourier transforms of the incident tone burst and the reflected wave have been obtained numerically and compared; these frequency spectra are identical except near the resonant frequency of the resonator, where there is a sharp dip in the spectrum of the reflected wave, showing that there is very little absorption by the resonator except near this frequency.

By measuring the physical properties of the resonator it is possible to calculate component values for the electrical series resonant circuit which, terminating an electrical transmission line, would be expected to provide analogous behaviour. The result of applying the frequency response function of this electrical circuit to the frequency spectrum of the experimental tone burst, and then taking the inverse transform has been found to agree very well with the experimentally measured returning wave.

Fig. 1. Reflection by a resonator of an 8 cycle sine wave tone burst at the resonant frequency of the resonator.
LOSS OF COHERENCY

As a measure of loss of coherency due to (a) tape speed fluctuations and (b) inter-channel phase characteristics nonlinearity, the change $\Delta R(\tau)$ of the autocorrelation function $R(\tau)$ of a signal recorded in and reproduced from one (a) or two channels (b) of a tape recorder is estimated:

$$a: \frac{\Delta R(\tau)}{R(\tau)} = \exp(-\frac{4\pi^2 f_0^2 \delta^2}{\tau^2}) - 1,$$

$$b: \frac{\Delta R(\tau)}{R(\tau)} = \exp(-\frac{4\pi^2 \delta^2}{\tau^2}) - 1 \leq 0,$$

$$c: \frac{\Delta R(\tau)}{R(\tau)} = -4\pi^2 f_0^2 \delta^2 > 0,$$

$$d: \frac{\Delta R(\tau)}{R(\tau)} = \int g(f) \{ \cos [\phi_n(f) + 2\pi f \tau] - \cos 2\pi f \tau \} df, \Delta R(\tau) \leq 0,$$

$$e: \frac{\Delta R(\tau)}{R(\tau)} = -\phi_n(f) \leq 0.$$ 

Here $\delta v$ and $v_0$ are fluctuating term and mean value of a tape speed respectively, $\tau$ denotes averaging over speed in (1-3) and over frequency band occupied by signal in (5). In (6) broad-band signal with center frequency $f_0$ is assumed. Otherwise (1) holds generally, (2) for Gaussian fluctuations, (3) for small fluctuations with any distribution. $G(f)$ is signal power spectrum density (p.s.d.), $\phi_n(f)$ is nonlinear component of the phase characteristics. (5) holds for small $|\phi_n(f)|$ and essentially constant $G(f)$. The estimates are confirmed experimentally, whereas (4) and (5) as a measure of the order of magnitude only.

STATISTICAL ERROR ESTIMATES

If $\varepsilon_0$ is relative RMS statistical error of p.s.d. estimate of white noise (given pass band width $B$ and center frequency $f_0$, sample duration $T$, true integrator averaging time $T_0 = T$, RC integrator time constant $\tau_0$), the following bound is proved for the statistical error of the analysis of coloured Gaussian noise with p.s.d. $G(f)$:

$$\varepsilon \leq k\varepsilon_0, \quad k = \max\{G(f)/G(f_0)\} \quad (f \in \text{pass band}).$$

The dependence of $\varepsilon_0$ on $B, f_0, T$ and $\tau_0$ is analysed in detail. Simple bound is proved:

$$\varepsilon_0 \leq \left[ \min\{1, 1/(Bx)\} \times \min\{1, \min\{1, 1/[Bax(2f_0/B-1)]^2\}\} \right]^{1/2}.$$ 

Here $x = T$ or $2\tau_0$ for true or RC integrator respectively. ($\tau_0 < T$ is assumed).

In the case of mixed spectrum (Gaussian noise $G(f)$ + lines with amplitudes $a_1, a_2, \ldots, a_n$ in the pass band):

$$\varepsilon_0 \leq \left[ k/(1+r) \right] \left[ k^2/(2k^2) \right] \left[ 1 + (1-\delta_{n_1})/(1+\delta_{n_2}) \right] (4\pi\sqrt{2}/k + 1),$$

where $k$ is as in (6), $r = \max\{a_i/2\}/[G(f_0)B]$, $\delta_{n_2}$ is Kronecker’s delta. (7) holds for both true and RC integrator.
DIGITAL OR ANALOG SOUND LEVEL METER READ-OUT

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Using Sound Level Meters with analog meter read-out, as specified in IEC 123 and 179, to measure fluctuating signals, the user of the instrument is forced to make some evaluation of the reading to describe the measured level in a single figure.

Although the meter dynamics and detector time constants in Sound Level Meters, standardized by IEC, are not very well defined, the analog meter read-out may be used for a kind of signature analysis by the experienced acoustician using the dynamic movements of the meter needle to judge about the character of the noise.

Some have tried to set up rules for reading the moving needle in order to obtain just one figure characterizing the Sound Pressure Level to be measured. This may be the maximum deflection, the average deflection, the minimum deflection or an attempt to obtain the integrated RMS.

The use of digital read-out does not offer the same possibility of judging about the dynamic property of the Sound Pressure Level. The human eye cannot catch and evaluate a rapid varying digital read-out. At least one second intervals between new readings is necessary in practice. However, once having defined a read-out mode, one get a well defined single number easy to read, also for the less experienced technician.

As seen, it is necessary to introduce a new practice for manual digital read-out. The figures must be presented with certain fixed intervals. This Sampling Interval may be controled by an internal clock (1 sec.) or specified by the interval between two external trigger impulses as for instance touches on a push-button.

Different measuring modes may be:

- Max. RMS in sampling interval
- Mean RMS in sampling interval
- Max. Peak in sampling interval
- True RMS in sampling interval
- Instantaneous RMS at sampling time.

Both analog and digital read-out will have advantages and disadvantages. This paper intend to discuss these, having defined the actual measured quantities.