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PREFACE

The present book of proceedings includes 95 contributions to be presented during the 16th meeting of the Nordic Acoustical Society at Aalborg University on August 20-22, 1986.

The field of acoustics is characterized by interdisciplinarity. In spite of this, there is a general trend today that the research results are presented only at very specialized conferences.

Therefore, we are happy to see that the contributions to this meeting cover a wide range within acoustics. It is our impression that a widely composed forum for information exchange is important for new ideas in our field.

For example, research on sound reproduction through high fidelity headphones, audiometric headphones and telephone headsets is normally carried out by groups associated with different institutions. The acoustical problems are basically the same, and an increased cooperation would be profitable.

For the first time the Organizing Committee has suggested all contributors to give the written version of their paper in English. A large number of authors have followed this request. We believe the change is right, as most of the contributions are of interest also to people outside the Scandinavian countries.

The possibility of reaching an international forum will probably stimulate the interest in contributing to the Nordic Acoustical Meetings. The change will not affect the informal and cheerful atmosphere that exists during the meetings.

Ann Toft is acknowledged for her great work with the layout of the book.

Aalborg, June 1986.

Henrik Møller and Per Rubak

INVITED PAPERS

CONTRIBUTED PAPERS

SOUND PROPAGATION

NOISE, MISCELLANEOUS

OFFSHORE NOISE

TRANSPORTATION NOISE

INDUSTRIAL NOISE

PSYCHOACOUSTICS

ELECTROACOUSTICS

AUDIOLOGY

SPEECH

INSTRUMENTATION

ROOM ACOUSTICS

SOUND INSULATION

STRUCTURE BORNE SOUND

FLUID ACOUSTICS

PHYSICAL ACOUSTICS

MUSICAL ACOUSTICS

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Transmission of structure borne sound from vibrating structures into elastic media

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1. Introduction

Transmission of structure borne sound from a vibrating body into an elastic medium takes place when a vibrator is in direct contact with a solid material. One important practical example in this area is the generation of ground vibrations by trains or other vehicles below and above ground, or by forge hammers, large compressor units, printing machines etc. Other examples are the damping of plate vibrations by energy transfer into an adjacent thick layer of a lossy material (e.g. sand), or the generation of ultrasound for non-destructive testing applications. All these - and many other - cases have in common, that wave energy is radiated into a medium with certain elastic properties. Obviously this mechanism is very similar to the radiation of sound into air or water, because the energy is carried away by waves, which have propagation properties determined by a wave equation and amplitudes determined by the boundary condition at the interface between the vibrating structure and the surrounding medium.

In this paper this similarity will be used throughout; i.e. concepts such as radiation impedance, radiation efficiency, coincidence frequency, radiation loss factor etc, that have been developed [1-3] and used successfully in air and water radiation problems, are used here to describe the transmission of sound energy from a vibrating body into a solid material. Obviously there is also a difference between the two problems, because in a gas or liquid only compressional waves can transport sound energy over larger distances, whereas in a solid material compressional waves and shear waves and combinations of both are possible. As a consequence of that the radiation into a solid elastic medium is somewhat more complicated than the corresponding fluid dynamics problem and the equations that describe the situation are approximately twice as long. Another, but not major difference, between the two problems is due to the boundary conditions. When sound is radiated into a gas or a fluid the boundary condition is equality of the normal components (normal to the radiator surface) of the velocities in the medium and on the radiator; the tangential components are just not mentioned, because of the absence of shear forces in a non-viscous medium. In an elastic continuum the situation is different (see Fig. 1). Here in addition to the equality of the normal components another condition concerning the tangential components of velocity or the shear forces is neccessary. The two conditions used in this paper are shown in the lower part of Fig. 1. In the following the boundary condition in a gas or fluid will be labelled "F" the boundary condition with no slip motion will be called "NS" and the one with no shear force at the interface (e.g. because there is a thin layer of sand between a subway tunnel and the surrounding ground) will be called "S". Obviously many other boundary conditions are possible but the two used here are extreme cases, which probably are sufficient to understand the problem.

2. Basic equations

The basic equations describing the motion in a linear, elastic, homogeneous medium with shear modulus G and Poisson's ration ν are (see e.g. [3] chapt II.5)

$$G\left[\Delta \underset{\sim}{v} + \frac{1}{1-2\nu}\text{grad div } \underset{\sim}{v}\right] - \rho\frac{\partial \underset{\sim}{v}^2}{\partial +^2} = 0 \quad . \tag{1}$$

Here $\underset{\sim}{v}$ is the velocity vector and ρ the density of the material. If all motions are harmonic in time with an angular frequency $\omega=2\pi f$, eq. (1) becomes

$$\Delta v_i + \frac{1}{1-2\nu}\frac{\partial}{\partial x_i}\left(\frac{\partial v_K}{\partial x_K}\right) + k_T^2 v_i = 0, i = 1,2; K = 1,2^* \tag{2}$$

In addition the stress-strain relation is needed, which in the terminology used here can be written as:

$$\begin{aligned}\sigma_{ii} &= \frac{2G}{j\omega}\left[\frac{\partial v_i}{\partial x_i} + \frac{\nu}{1-2\nu}\left(\frac{\partial v_K}{\partial x_K}\right)\right]\\ \sigma_{ij} &= \frac{G}{j\omega}\left[\frac{\partial v_i}{\partial x_j} + \frac{\partial v_j}{\partial x_i}\right] \quad \text{for } i \neq j.\end{aligned} \tag{3}$$

*) For the K index the summation convention is used

In these equations v_i are the velocity components, σ_{ii} the normal components of the stresses and σ_{ij} the shear stresses; $j = \sqrt{-1}$. The time factor exp $(j\hat{\omega}t)$ is, as usual, omitted. k_T is the shear wave number given by

$$k_T^2 = \omega^2 \frac{\rho}{G} = \frac{\omega^2}{c_T{}^2} = \left(\frac{2\pi}{\lambda_T}\right)^2 . \tag{4}$$

(c_T = shear wave speed, λ_T = shear wave velocity)

If the velocities and stresses are expressed in terms of spatial Fourier transforms; i.e. if all field variables are considered to consist of a combination of many plane waves of the form

$$v_i = \frac{1}{(2\pi)^n} \int_{-\infty}^{+\infty} \check{v}_i(k_K)\, e^{jk_K x_K}\, dk_1 \ldots dk_n , \tag{5}$$

(n = number of dimensions) eq. (2) and (3) become two sets of linear equations. It turns out that eq. (2) can be solved only if

$$k_3^2 = q_T^2 = k_T^2 - (k_1^2+k_2^2) \text{ or } k_3^2 = q_c^2 = k_c^2 - (k_1^2+k_2^2) . \tag{6}$$

Thus there are two types of waves and therefore the most general solution of eq. (2) in cartesion coordinates is

$$v_i = \frac{1}{(2\pi)^{n-1}} \int_{-\infty}^{+\infty} \left(\check{v}_{ic+} e^{-jq_c x_3} + \check{v}_{ic-} e^{jq_c x_3} + \check{v}_{iT+} e^{-jq_T x_3} + v_{iT-} e^{jq_T x_3}\right) . e^{jk_1 x_1} e^{jk_2 x_2} dk_1 dk_2 . \tag{7}$$

The quantities $\check{v}_{ic}$ and $\check{v}_{iT}$ are not independent; between them the relations

$$\check{v}_{ic} k_j = \check{v}_{jc} k_i \quad \text{and} \quad \check{v}_{KT} k_K = 0 \tag{8}$$

hold. Here the wave number for compressional waves is used in addition. It is given by

$$k_c^2 = k_T^2 \frac{1-2\nu}{2-2\nu} = \frac{\omega^2}{c_c^2} = \left(\frac{2\pi}{\lambda_c}\right)^2 \tag{9}$$

3. Radiation impedance of an infinite plane radiator

In the following the calculations are restricted to two dimensions, i.e. k_1 set equal to zero and $n = 2$; furthermore the elastic continuum is assumed to be unbounded so that waves are only travelling away from the radiator. Under these circumstances eq. (7) becomes (see e.g. [4])

$$v_i(x_2,x_3) = \frac{1}{2\pi}\int\left[\check{v}_{ic+}e^{-jq_c x_3}+\check{v}_{iT+}e^{-jq_T x_3}\right]e^{jk_2x_2}dk_2 \,. \tag{10}$$

Dropping the index + and taking only the transformed quantities at $x_3 = 0$ one finds

$$\check{v}_3 = \check{v}_{3c} + \check{v}_{3T} \;; \quad \check{v}_2 = \check{v}_{2c} + \check{v}_{2T}. \tag{11}$$

Because of eq. (8) there are the additional conditions

$$-\check{v}_{2c}\, q_c = \check{v}_{3c}\, k_2 \;; \quad \check{v}_{2T}\, k_2 = \check{v}_{3T} q_T. \tag{12}$$

If as a next step eq. (3) is Fourier transformed and if the transforms are taken only for $x_3 = 0$, those quantities that are needed in the following are obtained as:

$$\begin{aligned}\check{\sigma}_{33} &= \frac{2G}{\omega}\left\{-q_c\check{v}_{3c}-q_T\check{v}_{3T}+\frac{\nu}{1-2\nu}\left[-q_c\check{v}_{3c}-q_T\check{v}_{3T}+k_2(\check{v}_{2c}+\check{v}_{2T})\right]\right\}\\ \check{\sigma}_{23} &= \frac{G}{\omega}\left[-q_c\check{v}_{2c}-q_T\check{v}_{2T}+k_2\,(\check{v}_{3c}+\check{v}_{3T})\right].\end{aligned} \tag{13}$$

Eq. (11), (12), (13) can be reduced to

$$\begin{aligned}-\check{\sigma}_{33} &= \frac{G}{\omega}\left[\alpha^2(q_c-q_T)\,\check{v}_{3c}+\alpha^2\,q_T\check{v}_3 - \frac{2\nu}{1-2\nu}\,k_2\check{v}_2\right]\\ \check{\sigma}_{23} &= \frac{G}{\omega}\left[(k_2^2+q_T^2)\check{v}_{3c} + (k_2^2-q_T^2)\check{v}_3\right]\frac{1}{k_2}\\ \check{v}_2 &= -\left(\frac{k_2}{q_c}+\frac{q_T}{k_2}\right)\check{v}_{3c} + \frac{q_T}{k_2}\check{v}_3\end{aligned} \tag{14}$$

Here $\alpha^2 = 2\,\frac{1-\nu}{1-2\nu} = \frac{k_T^2}{k_c^2} = \frac{c_c^2}{c_T^2}$. (15)

For the ideal no-slip condition; i.e. for $\check{v}_2=0$, eq. (14) yields for the radiation impedance Z_{RNS}

$$Z_{RNS} = \frac{-\check{\sigma}_{33}}{\check{v}_3} = \rho c_c\,\frac{\alpha^2\sqrt{\alpha^2-x^2}}{x^2+\sqrt{1-x^2}\sqrt{\alpha^2-x^2}} \tag{16}$$

with $x = \frac{k}{k_c}$.

Similarly one finds for the "slip condition"; i.e. for $\check{\sigma}_{22} = 0$:

$$Z_{RS} = \rho c_C \frac{1}{\sqrt{1-x^2}} \left[\left(2\frac{x^2}{\alpha^2} - 1\right)^2 + \frac{4x^2}{\alpha^2} \sqrt{1-x^2} \sqrt{\alpha^2 x^2} \right]. \tag{17}$$

It is also possible to relate the shear force $\check{\sigma}_{23}$ to the tangential velocity $\check{v}_2$ at the interface. Using as a second boundary condition $\check{v}_3 = 0$, the following expression is obtained (see Fig. 2)

$$Z_{RLNS} = \frac{-\check{\sigma}_{23}}{\check{v}_2} = \rho c_C \frac{\sqrt{1-x^2}}{x^2 + \sqrt{1-x^2}\sqrt{\alpha^2 - x^2}} \quad . \tag{18}$$

In eq. (16) - (18) the compressional wave impedance ρc_C is used as a leading factor; this way the similarity to the radiation impedance into a fluid which is given by

$$Z_{RF} = \rho c_C \frac{1}{\sqrt{1-x^2}} , \tag{19}$$

is made more evident. Furthermore ρc_C is the limiting value for Z_R when $x \to 0$ in eq. (16) and (17). In eq. (18) the limit is $\rho c_C / \alpha = \rho c_T$.

In practical problems the radiation quite often is due to the bending or longitudinal motion of plate - like structures.

In these cases the normal and the tangential component of the velocity are connected by

$$\check{v}_{2B} = jk\frac{h_P}{2}\check{v}_{3B} \text{ (bending) } \check{v}_{3L} = jkh_P \frac{\nu_P}{2-2\nu_P} \check{v}_{2L} \text{(longitudinal)}. \tag{20}$$

(h_P = plate thickness, ν_P = Poisson's ratio of the plate material).

These relations can also be included in the derivation of the radiation impedance. If this is done it turns out that eq.(16) and (17) safely can be used for bending waves, because kh_P always is very small. Something similar is true for longitudinal waves where for almost all materials eq. (18) is sufficiently accurate. An exception occurs, when $\nu \approx 0{,}5$; i.e. when the material is almost liquid (e.g. rubber), in this case the radiation impedance Z_{RS} or Z_{RNS} should be used in addition. (See also Fig. 2, where the formula for the radiation of sound power P from a very large radiator of area S is also given). For the Poisson's ratio $\nu = 0{,}3$ (normal metal or building material) and $\nu = 0{,}48$ (rubber-like material) the real and imaginary parts of the radiation impedance are shown on Figs. 3 and 4. As abszissa the wave length ratio $\lambda/\lambda_c = k_c/k$ is used. Thus for $\lambda=\lambda_c$ the wave speed on the radiator and the compressional wave speed in the surrounding medium are equal (coincidence). There is, however, another important value, namely $\lambda=\lambda_c/\alpha$; in this case $\lambda=\lambda_T$; thus there is coincidence of radiator wave speed with the shear wave speed.

Some conclusions that can be drawn from the formulas for the radiation impedance are:

- for $\lambda<\lambda_T = \lambda_c/\alpha$ the real part of the radiation impedance vanishes;
- for $\lambda>\lambda_c$ the imaginary part of the impedance vanishes and the real part becomes ρc_c for the bending vibration and ρc_T for the longitudinal vibration;
- for $\lambda_T<\lambda<\lambda_c$, which for typical materials covers appr. one octave (for rubber-like materials much more) the real part is in the average roughly $\pi/4\alpha$ for bending motion and somewhat lower for longitudinal motion. The imaginary part is positive (i.e. has the character of a mass load) for a fluid material (F) in the whole range, for the no-slip (BNS) condition in the range $\lambda_T<\lambda<\lambda_c$, for the slip condition (BS) in the range $\lambda_R<\lambda<\lambda_c$. Here λ_R is the wavelength of the Rayleigh wave, which is always slightly lower than the shear wave. For longitudinal motion and for bending motion in the short wave length regime the radiation loading acts like an added stiffness.

4. Radiation efficiency of a finite, plane velocity source

As soon as the radiation impedance for plane waves is known, the radiation from a finite radiator of any velocity distribution can be found fairly easily. In this paper the radiation efficiency, defined by

$$\sigma_R = \frac{P}{\rho c_c S\overline{v^2}} , \tag{21}$$

is considered to be of major interest. Here P is the radiated power, S is the area of the radiator and $\overline{v^2}$ is its mean square velocity. Unfortunately definition (21) obscures the fact that the value of σ_R depends on the space dependence of

the velocity and on the boundary conditions at the edges of the radiator.
Analog to the radiation problem in fluids [5] the radiated power for a one dimensional source is related to the Fourier transforms of the stresses and velocities by (see also Fig.2)

$$P = \frac{1}{4\pi}\int\left[\mathrm{Re}\left\{\check{\sigma}_{33}\check{v}_3^{\,*}\right\} + \mathrm{Re}\left\{\check{\sigma}_{23}\check{v}_2^{\,*}\right\}\right]dk =$$
$$= \frac{1}{4\pi}\int\left[\mathrm{Re}\left\{Z_{R3}\right\}|\check{v}_3|^2 + \mathrm{Re}\left\{Z_{R2}\right\}|\check{v}_2|^2\right] dk. \qquad (22)$$

(the star denotes complex conjugate, thus $\check{v}_3 \cdot \check{v}_3^{\,*} = |\check{v}_3|^2$.)

Since the radiation impedances are known as function of k or $x = k/k_c$ eq. (22) can be evaluated when the Fourier transforms of the driving velocities are known. Thus when $v_3(x_2)$ is given as a function of the coordinate x_2 one has to find

$$\check{v}_3 = \int_{-\infty}^{+\infty} v_3(x_2)e^{-jkx_2}dx_2 \qquad (23)$$

(similarly for v_2) and insert the results into (22) and (21). Fig. 5 shows some results obtained this way when the driving velocity is given by

$$v_3(x_2) = \begin{cases} v_o\ e^{-jk_Bx} & \text{for } 0<x_2<l \\ 0 & \text{otherwise.} \end{cases} \qquad (27)$$

Thus it is assumed that a bending wave of wave length $\lambda_B = 2\pi k_B$ travels from left to right but radiates only over a "window" of size l.

For discussing Fig. 5 it is useful to introduce the coincidence frequencies f_{gc} and f_{gT}. When $f = f_{gc}$, the bending wave length is equal to the compressional wave length of the material and when $f = f_{gT}$ it is equal to the shear wave length. Thus for $f = f_{gc}$ the relations $k_B = k_c$ or $c_B = c_c$ hold, and for $f = f_{gT}$ the relation $k_B = k_T$ or $c_B = c_T$. The frequency parameter used in Fig. 5 is $f/f_{gc} = k_c^2/k_B^2$; these are the same quantities that are used for describing bending wave radiation into air. The other parameter used in Fig. 5 is the number of coincidence wave lengths λ_{gc} that fit into the radiator length l.

As preliminary approximations for the radiation efficiencies for bending motion one might use

$$\sigma_R \approx \begin{cases} 1 & \text{for } f > f_{gc} \\ 1/\alpha^2 & \text{for } f_{gT} < f < f_{gc} \\ \lambda_g/2\pi l & \text{for } f < f_{gT}. \end{cases} \quad (25)$$

For longitudinal motion σ_R is a few dB lower.

Fig. 6 shows some results when the driving velocity is given by a standing wave of the type

$$v_3(x_2) = \begin{cases} v_o \sin n\pi x_2/l & \text{for } 0<x_2<l \\ 0 & \text{otherwise} \end{cases} \quad (26)$$

Some conclusions that can be drawn from these curves are:

- for $2l/n>\lambda_c$; i.e. for high frequencies $\sigma_R \approx 1$;
- for n = 1 the system behaves like a line monopole, giving $\sigma \sim \omega$ for $\omega \to 0$;
- for n = 2 the system behaves like a line dipole, giving $\sigma \sim \omega^2$ for $\omega \to 0$;
- for n = 3,5,7,... and $2l/n<\lambda_c$ there is a canellation effect ("hydrodynamic short circuit"); the sound radiated in this case comes from line monopoles at x_2 = 0 and x_2 = l;
- for n = 4,6,.... and $2l/n<\lambda_c$ there is even stronger cancellation; in this case the remaining sources at the edges are dipoles.

The most important conclusion that can be drawn from the calculations and from the figures, lies in the fact that the radiation efficiency for sound transmission from bending motion into a surrounding elastic medium is very similar (only slightly higher) than the radiation efficiency for the corresponding problem in a fluid with the same value of ρc_c.

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[3] Cremer, L.; Heckl, M.; Ungar, E.: Structure Borne Sound. Springer 1975

[4] Heckl, M.: Vibration Transmission and Sound Radiation. AGARD Report No. 700 (Modern Data Analysis Techniques in Noise and Vibration Problems).Neuilly sur Seine 1981

[5] Heckl, M.: Abstrahlung von ebenen Schallquellen. Acustica 37 (1977) p 155 - 166

Boundary conditions for radiation problems

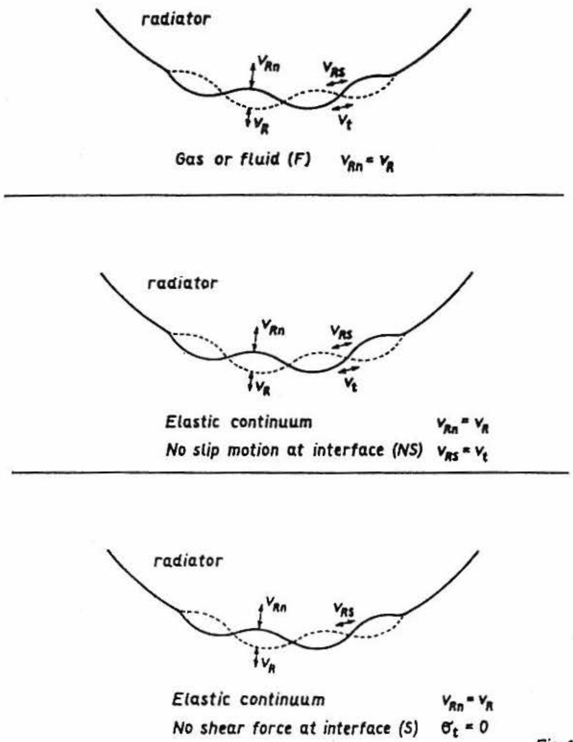
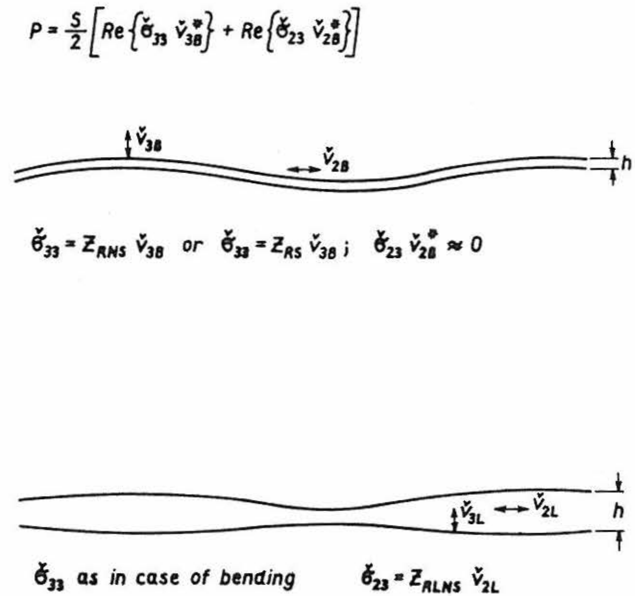


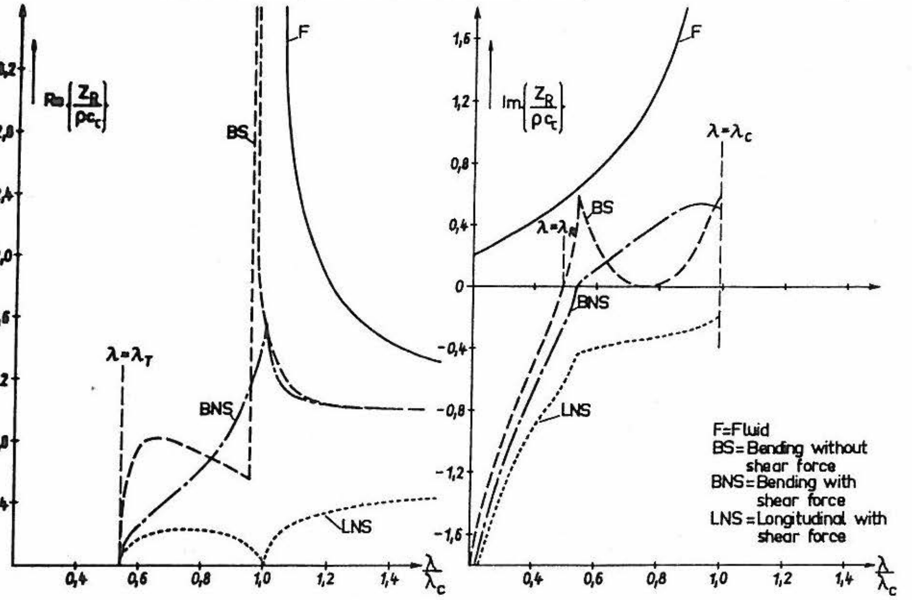
Fig. 1



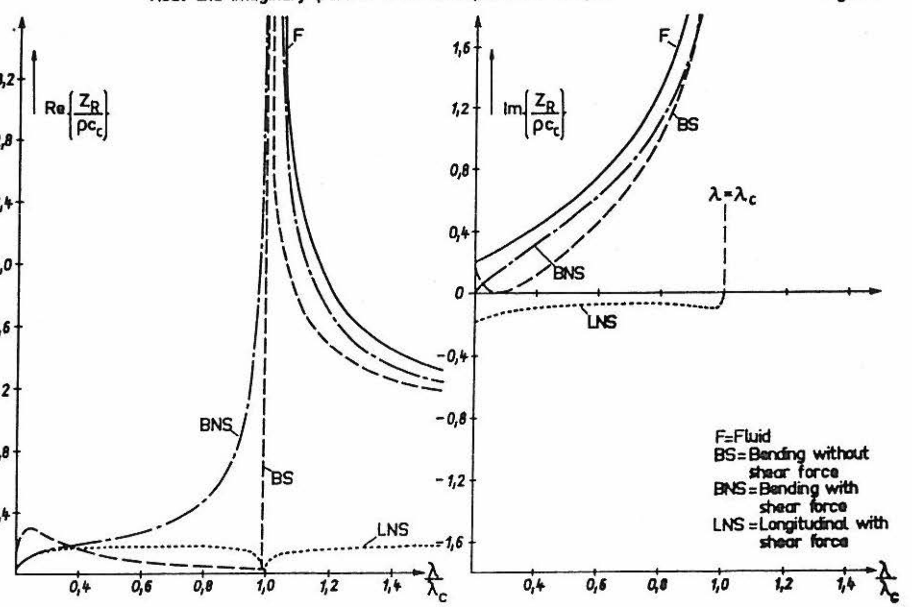
Radiation from plates in bending or longitudinal motion

Fig. 2

Real and imaginary part of radiation impedance V=0,3 *Fig. 3*



Real and imaginary part of radiation impedance. V=0,48 *Fig. 4*



Radiation efficiency of a finite velocity source, $V=0{,}3$

$v = v_o e^{-jk_B x}$

1: $l=0{,}5\lambda_{gc}$; 2: $l=\lambda_{gc}$; 3: $l=2\lambda_{gc}$; 4: $l=4\lambda_{gc}$; 5: $l=8\lambda_{gc}$

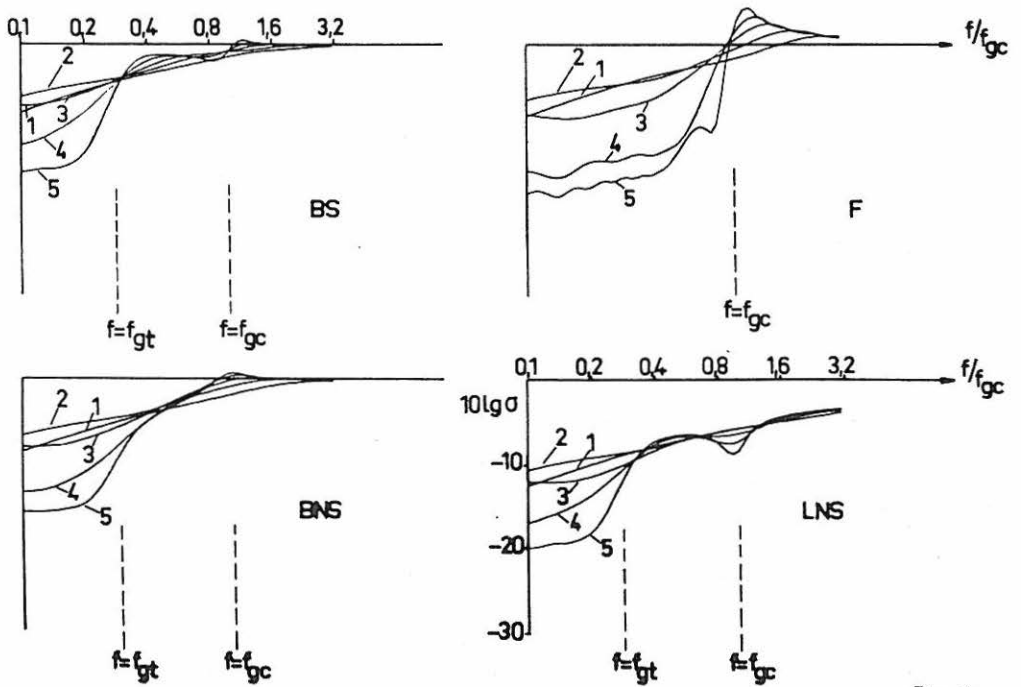


Fig. 5

Radiation efficiency of a finite velocity source, $V=0{,}3$

$v = v_0 \sin \frac{n\pi x}{l}$

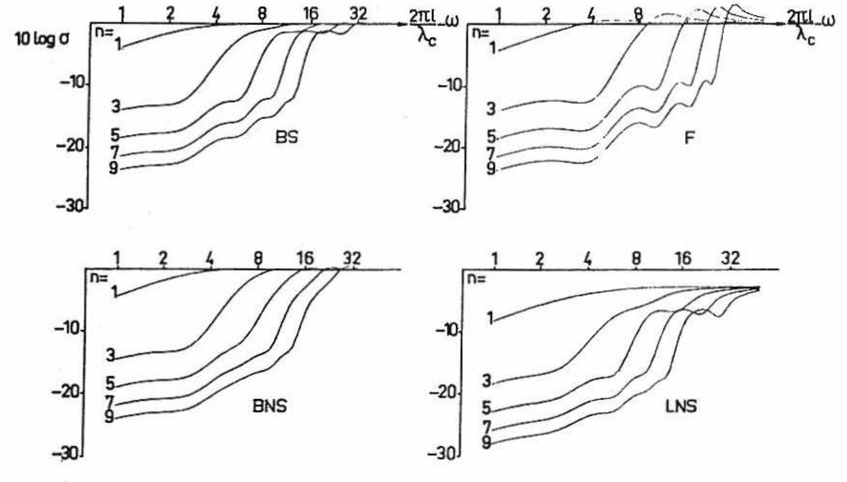


Fig. 6

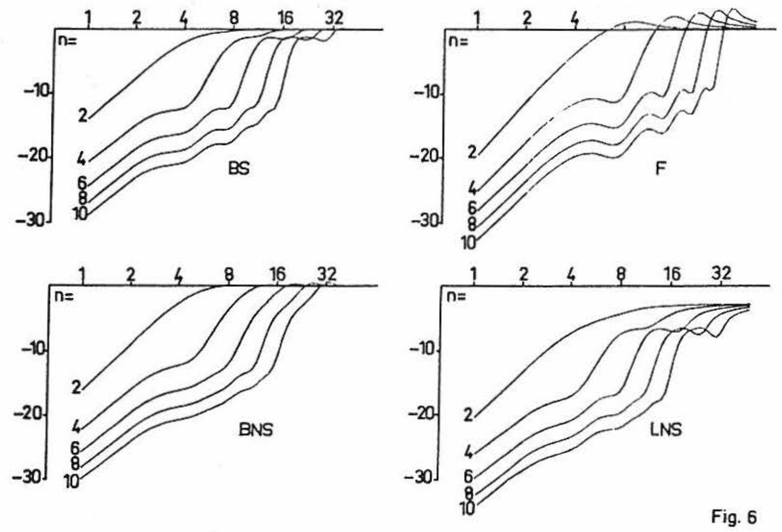


Fig. 6

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NORDTESTS VERKSAMHET INOM AKUSTIKOMRÅDET

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1. Allmänt om Nordtest

Nordtest är ett gemensamt nordiskt organ med uppgift att främja utvecklingen inom provningsområdet på ett sådant sätt att tekniska handelshinder undanröjs. Det bildades 1973 av Nordiska Ministerrådet på initiativ av Nordiska Rådet. Arbetet inom akustikområdet leds av fackgruppen för Akustik och Buller som bildades 1974. Fackgruppen, som har en representant från varje nordiskt land, arbetar inom områdena byggnadsakustik, buller, vibrationer och elektroakustik. I fackgruppen sitter för närvarande Hans Jonasson från Statens provningsanstalt i Borås, Fritz Ingerslev från DTH i Lyngby, Jens Trampe Broch från Lydteknik Senter i Trondheim, Juhani Parmanen från Statens tekniska forskningscentral i Helsingfors, Steindor Gudmundsson från Islands Byggforskningsinstitut samt Bo Lindholm från Nordtest.

Inom akustikområdet disponerar Nordtest ca FIM 300 000 per år för olika projekt. Ansökningar ställs till Nordtest. De behandlas i fackgruppen varefter Nordtests styrelse fattar beslut. Normalt utföres projekt av en projektledare som till sitt förfogande har en nordisk projektgrupp. Projekten avslutas ofta med ringprovning för slutlig kontroll av den föreslagna provningsmetoden.

Förutom inom akustikområdet arbetar Nordtest inom områdena bygg, brand, elektronik, kemi, mekanik, VVS samt NDT(icke förstörande provning). Alla dessa områden har egna fackgrupper. Därutöver finns två tvärvetenskapliga programgrupper som arbetar med harmonisering resp konsumentvaruprovning.

2. Uppnådda resultat

Nordtests verksamhet inom akustikområdet har hittills resulterat i 54 registrerade Nordtest-metoder av vilka 20 är utvecklade i egen regi. Flera av metoderna har kraftigt påverkat senare ISO-arbete inom området. Detta gäller t ex den första egenutvecklade metoden om intrimning av diffusorer vid ljudabsorptionsmätning i efterklangsrum. Denna återfinns nu som en del av den nyligen utkomna standarden ISO 354 "Measurement of sound absorption in a reverberation room". Två Nordtest-metoder, NT ACOU 036 och NT ACOU 037 för mätning av skärmdämpning hos kontorsskärmar resp ljudisolering hos små byggdelar utgör basen för två ISO-arbetsgruppers arbete för framtagning av nya ISO-metoder. Samtliga egenutvecklade Nordtest-metoder finns förtecknade i ANNEX A. Därutöver finns ett antal ISO- och IEC-metoder som antagits i oförändrat skick.

De flesta Nordtest-projekt har innehållit avsnitt om ringprovningar mellan olika nordiska laboratorier. Till följd av detta är det numera en självklarhet att mätresultat från något av de i Nordtest deltagande laboratorierna accepteras överallt i Norden. Detta har verksamt bidragit till att hålla nere provningskostnaderna och underlätta varuutbyte på den nordiska marknaden.

Nordtest har också utarbetat anvisningar för hur ett akustiskt laboratorium skall säkerställa kvaliteten på sina provningar. Detta har lett till att kvalitetshandböcker nu har utarbetats för tre av de ledande provningslaboratorierna i Norden. Fler väntas följa efter inom kort. Dessa kvalitetshandböcker förväntas leda till en allmän höjning av kvaliteten i provningsverksamheten samt att öka möjligheterna för att nordiska provningsresultat blir erkända på den internationella marknaden.

Ett fint exempel på hur framgångsrikt Nordtest har fungerat som katalysator för det nordiska samarbetet är ANNEX B som är en förteckning över samtliga tekniska rapporter som utarbetats i samband med hittills genomförda Nordtestprojekt. Rapporterna kan rekvireras från de utförande institutionerna.

3. Årets projekt

Som ett exempel på ett typiskt Nordtestprojekt kan nämnas "årets projekt", som årligen utses av fackgruppen. 1985 var detta "Bestämning av ljudeffektnivå med referensljudkälla".

Enligt internationell och modern nordisk praxis skall en bullrande maskin ljudmässigt klassificeras efter sin ljudeffektnivå. Denna mäts normalt enligt någon av standarderna ISO 3741-3747. Problemet med dessa är att de antingen kräver dyrbara speciallaboratorier eller har för dålig mätnoggrannhet. En korrekt bullerdeklaration kräver normalt mätningar med minst "engineering"-noggrannhet. Målet med Nordtestprojektet var därför att om möjligt ta fram en ny mätmetod som även under enkla förhållanden i fält gav denna noggrannhet. Enda möjligheten att uppfylla kraven var att arbeta med en referensljudkälla.

Utvecklingsarbetet finansierades av ett forskarråd, Arbetarskyddsfonden i Sverige, medan Nordtest finansierade de nordiska samarbetskostnaderna och en avslutande ringprovning för att slutgiltligt avgöra den föreslagna metodens mätnoggrannhet. 4 olika ljudkällor användes vid dessa mätningar som genomfördes efter 5 olika metoder. De olika källorna var en normal fläktreferensljudkälla med volymen 0,027 m3, en matrisskrivare med volymen 0,058 m3, en liten industridammsugare med volymen 0,236 m3 samt en högdirektiv låda med volymen 0,800 m3 och försedd med ett hål och en inre ljudkälla. Resultet av den nordiska ringprovningen sammanfattas i tabellen nedan.

Mätmetod	Källa				
	Referens-källa	Matris-skrivare	Damm-sugare	Låda med hål	ISO-krav
3741, jämförelse	0,04	0,07	0,03	0,06	0,2
3741, direkt	0,01	0,06	0,04	0,04	0,2
3744, direkt	0,04	0,03	0,09	0,03	0,2
3747	----	0,07	0,05	0,05	0,4
Nordtest	0,03	0,05	0,04	0,11	---

Tabell. Standardavvikelsen i B A-vägd ljudeffektnivå för de olika källorna och metoderna. Samtliga mätningar är utförda i oktavband. Resultaten baseras på mätningar i 4 olika laboratorier. För metoderna 3747 och Nordtest användes 3 olika vanliga rum per lab.

Av tabellen framgår att den nya, kraftigt förenklade metoden med marginal uppfyller ISO-kraven för "engineering"-noggrannhet även för den mycket direktiva lådan med hål. Den nya Nordtestmetoden kompletterar ISO 3747 eftersom den i motsats till denna föreskriver mätningar i efterklangsfältet. Det förutsättes dock att ljudkällan kan flyttas till ett rum där begränsningsytorna har begränsad ljudabsorption. Denna begränsning är nödvändig för noggrann mätning på högdirektiva källor. Samma goda resultat föreligger även i oktavband.

Erfarenheterna i projektet har förts vidare till ISO och bl a resulterat i förbättringar av ISO/DIS 3747 som inom kort föreligger reviderad som ISO 3747.

4. Pågående och planerade projekt

För närvarande pågår elva projekt. Dessa finns förtecknade i ANNEX C. För 1987 planeras fem nya projekt. Dessa avser mätning av vibrationer för bedömning av störningseffekt i byggnader, värdering av stegljudsförbättring på träbjälklag, mätning av stegljudsnivå på lätta bjälklag, bestämning av ljudeffektnivå från stora ljudkällor i fält samt mätning av insättningsdämpning hos kanalljuddämpare.

ANNEX A --- EGENUTVECKLADE NORDTESTMETODER

Metod nr	Titel	Beskrivning
NT ACOU 012	Reverberation room - suspended diffusers: absorption coefficients	Regler för intrimning av diffusorer i efterklangsrum vid mätning enligt ISO 354
NT ACOU 013	Doors and windows - sound reduction index	Komplettering till ISO 140. Monteringsanvisning samt regler för flanktransmissionskorrektion
NT ACOU 014	Hearing aid - induction loop systems: magnetic field characteristics	Provningsmetod för hörslingor i offentliga salar
NT ACOU 032	Acoustical screens - sound absorption	Komplettering till ISO 354. Montering samt redovisning av data för kontorsskärmar
NT ACOU 033	Bulkheads - sound radiation efficiency: laboratory measurements	Specifikation av provrigg samt mätning och beräkning av strålningsfaktor för fartygsskott
NT ACOU 034	Floor coverings - rating of impact sound improvement	Beskrivning av metod för entalsutvärdering av stegljudsförbättring
NT ACOU 035	Floating floors - insulation materials: dynamic stiffness	Bestämning av dynamisk styvhet hos det elastiska mellanskiktet hos flytande golv
NT ACOU 036	Acoustical screens - screen sound attenuation	Monteringsanvisning och metod för bestämning av kontorsskärmars skärmdämpning
NT ACOU 037	Small building elements - sound insulation	Komplement till ISO 140 för mätning av ljudisolering på små byggdelar som överluftdon och kabelgenomföringar
NT ACOU 038	Noise absorber pads - sound absorption	Komplement till ISO 354 för montering av bafflar och redovisning av deras ljudabsorptionsdata

NT ACOU 039	Road traffic - noise	Detaljerad mätmetod för vägtrafikbuller att använda vid t ex kontroll av beräkningsmetod
NT ACOU 040	Reverberant test rooms - sound absorption: reference sound source	Bestämning av absorptionskorrektion med referensljudkälla vid mätning av ljudisolering och ljudeffekt
NT ACOU 041	Sound level meters: verification procedure	Metod för regelbundet återkommande kontroll av ljudnivåmätare
NT ACOU 042	Rooms: noise level	Bestämning av ljudnivå i rum med byggnormskrav
NT ACOU 050	Floor coverings: reduction of transmitted impact noise - laboratory method	Komplement till ISO 140 vid mätning av stegljudsförbättring
NT ACOU 051	Comminuting machines: noise	Bestämning av ljudeffektnivå på granuleringskvarnar under specificerade driftsvillkor
NT ACOU 052	Sheet folding machines: noise	Bestämning av ljudeffektnivå på falsmaskiner under specificerade driftsvillkor
NT ACOU 053	Rooms: reverberation time	Bestämning av efterklangstid i rum med byggnormskrav
NT ACOU 054	Garden vehicles: operator's noise	Mätning av buller på operatörsplatsen på trädgårdsmaskiner
NT ACOU 056	Road traffic noise Simplified method	En förenklad version av NT ACOU 039
NT ACOU	Cabins and enclosures: sound insulation Part I: Sound protecting cabins Part II: Small enclosures	Bestämning av insatsisoleringen för hytter och inbyggnadssystem med olika metoder med och utan speciella ljudkällor

ANNEX B --- REFERENSER ÖVER UTFÖRDA NORDTESTPROJEKT

Rapporter, i kronologisk ordning, som helt eller delvis finansierats av Nordtest:

Hans Gerdien & Jarl Olofsson 1974, Internordisk jämförelsemätning av ljudabsorptionsfaktor, SP-RAPP 1974:30

Truls Gjestland 1975, Behovsanalys avseende buller, Rapport STF44 A75044, ELAB

Nic Michelsen 1976, Sammenlignende reduktionstalsmålinger for döre målt i laboratorium, Rapport nr 4, Lydteknisk Laboratorium

Rolf Ohlon 1977, Nordic Comparison Measurements of Absorption Coeffients, SP-RAPP 1977:13

Hans Jonasson 1977, Measurement of sound absorption of screens, SP-RAPP 1979:31

Nic Michelsen 1978, Måling af baflers absorption Rapport nr 15, Lydteknisk Laboratorium

Nic Michelsen 1978, Karakterisering af gulvbelägningers trilyddämpede egenskaper ved en enkelt talvärdi, Rapport nr 16, Lydteknisk Laboratorium

Matias Ringheim & al 1978, Maskinstöy. Veiledning for standardiserte målinger, Rapport STF44 A78063, ELAB

Nic Michelsen 1979, Måling i laboratorium af strålingsfaktor for skibsskod, Rapport nr 17, Lydteknisk Laboratorium

Ulf Kristiansen 1979, Seminar om dempning i lydfeller Rapport STF44 A79092, ELAB

Hans Jonasson 1980, Measurement of Insertion Loss of Screens SP-RAPP 1980:8

Knut Ulvund 1980, Målemetoder til bestemmelse av innredningsskotts reduksjonstall, Rapport 80-1125 från Det norske Veritas

Jan Arne Austnes 1980, Materialer for flytende golv. Akustiske egenskaper vid dynamisk påkjenning, NBI-rapport

Nic Michelsen 1980, Måling af äkvivalent absorption ved brug af en referencelydkilde, Rapport nr 20, Lydteknisk Laboratorium

Kaj Bodlund 1980, Mätning och redovisning av ljudisolering hos små byggnadselement, SP-RAPP 1980:22

Kaj Bodlund 1981, Mätning och redovisning av buller från avloppsinstallationer, SP-RAPP 1981:38

Hans Jonasson & Lennart Eslon 1982, Mätning av ljudisolering hos små inbyggnadssystem, SP-RAPP 1982:30

Knud Rasmussen 1982, Minimumskrav til laboratorier der udförer akustisk prövning, Publikation nr 17, Laboratoriet for Akustik, DTH

Jens Trampe Broch 1982, Seminar on impact sound insulation test methods, Rapport STF44 A82022, ELAB

Jörn Kjaer 1982, Databank for bygningsakustiske målinger. Förprojekt, Rapport fra Byggeriets Akustiske Målestation

Matias Ringheim 1982, Measurement of road traffic noise KILDE report 47

Torben Holm Pedersen 1982, Round Robin Test af lydtrykmålere, mikrofoner og kalibratorer Rapport 33, Lydteknisk Laboratorium

Hans Peter Wallin & Göran Gadefelt 1982, Studium av svängare för intern ljudfältsuppbyggnad i kapslar och huvar, Rapport TRITA-TAK-8202, Teknisk Akustik, KTH

Nic Michelsen & Birgit Rasmussen 1982, Laboratory effects on the measured sound reduction index of windows and glazings, Report no 34, Lydteknisk :Laboratorium

Hans Jonasson 1982, Bestämning av A-vägd ljudtrycksnivå samt efterklangstid i rum, SP-RAPP 1982:40

Kaj Bodlund 1983, Laboratory Measurement of the Improvement of Impact Sound. Insulation by Floor Coverings on a Standard Floor, SP-RAPP 1983:01

Jens Holger Rindel 1983, Måling af indbygningssystemers lydisolation, Publikation nr 20, Laboratoriet for Akustik, DTH

Hans Jonasson 1983, Bullermätningar på maskiner -- Granuleringskvarnar och falsmaskiner, SP-RAPP 1983:21

Torben Poulsen 1984, Nordic Round Robin Test on Hearing Protectors - Subjective Method, Internal Report No 21, The Acoustics Laboratory, DTH

Sören Damgaard Kristensen & Birgit Rasmussen 1984, Repeatability and reproduceability of sound insulation measurements, Report no 118, Lydteknisk Institut

Liljeroos 1984, Measuring method of driver's noise exposure for machine powered garden vehicles, Research note 343 from VTT

Kaj Bodlund 1984, Reverberation Time Measurements According to the Interrupted Noise Method, SP-RAPP 1983:35 och NT TECHN REPORT 026

Torben Poulsen 1984, Nordic Round Robin Test on Hearing Protectors - Objective Results, Internal Report No 22, The Acoustics Laboratory, DTH

Birgit Rasmussen 1984, Measurement of sound reduction index for glazings in a staggered test opening, Report no 119, Lydteknisk Institut

Hans Jonasson 1985, Accurate Sound Power Measurements Using a Reference Sound Source, SP-RAPP 1984:19

Juhani Parmanen 1985, A short test method for sound insulation measurements in dwellings, Report 158-85/LV17 from VTT

Hans Jonasson 1986, Bestämning av A-vägd ljudisolering hos fönster, SP-RAPP 1985:43

Fritz Ingerslev 1986, Retningslinjer for udarbejdelse af kvalitetshåndböger for laboratorier der udförer teknisk prövning inden for det akustiske fagområde, Rapport från Laboratoriet for Akustik, DTH

ANNEX C --- PÅGÅENDE PROJEKT

Montering av vibrationsgivare på handverktyg.

Mätning av bullerimmission från flygbuller.

Bestämning av trafikbullerskärmars insättningsdämpning.

Kalibrering av byggnadsakustiska hammarapparater.

Mätning av fönsters ljudisolering i fält.

Mätning av vibrationer i trädgårdsfordon.

Bestämning av ljudeffektnivå medelst intensitetsmätning.

Montering av rutor och fönster vid ljudisoleringsmätningar.

Objektiv mätning av hörselskydds dämpning.

Förenklad metod för mätning av stegljudsnivå i bostäder.

Kontroll av förenklad metod för mätning av luftljudsisolering.

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Pattern Recognition Approaches to Speech Recognition

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ABSTRACT

Algorithms for speech recognition can be dichotomized into two broad classes – namely pattern recognition approaches and acoustic phonetic approaches. To date, the greatest degree of success in speech recognition has been obtained using pattern recognition paradigms. Hence, in this paper, we will be concerned primarily with showing how pattern recognition techniques have been applied to the problems of isolated word (or discrete utterance) recognition, connected word recognition, and continuous speech recognition. We will show that our understanding (and consequently the resulting recognizer performance) is best for the simplest recognition tasks and is considerably less complete for large scale recognition systems.

I. Introduction

When one talks about the problem of speech recognition by machine, an image is conjured up of a machine like HAL in the movie 2001, or C3P0 in the movie Star Wars. These fictional machines had the ability to understand fluent, conversational speech, with unrestricted vocabulary, from essentially any talker. Although the promise of such a capable machine is as yet unfullfilled, the field of automatic speech recognition has made significant advances in the past decade [1-3]. This is due, in part, to the great advances made in VLSI technology, which has greatly lowered the cost and increased the capability of individual devices (e.g. processors, memory), and in part due to the theoretical advances in our understanding of how to apply powerful mathematical modelling techniques to the problems of speech recognition.

When setting out to define the problems associated with implementing a speech recognition system, one finds that there are a number of general issues that must be resolved before designing and building the system. One such issue is the size and complexity of the user vocabulary. Although useful recognition systems have been built with as few as two words (yes, no), there are at least four distinct ranges of vocabulary size of interest. Very small vocabularies (on the order of 10 words) are most useful for control tasks – e.g. all digit dialing of telephone numbers, repertory name dialing, access control etc. Generally the vocabulary words are chosen to be highly distinctive words (i.e. of low complexity) to minimize potential confusions. The next range of vocabulary size is moderate vocabulary systems having on the order of 100 words. Typical applications include spoken computer languages, voice editors, information retrieval from databases, controlled access via spelling etc. For such applications, the vocabulary is generally fairly complex (i.e. not all pairs of words are highly distinctive), but word confusions are often resolved by the syntax of the specific task to which the recognizer is applied. The third vocabulary range of interest is the large vocabulary system with vocabulary sizes on the order of 1000 words. Vocabulary sizes this large are big enough to specify fairly comfortable subsets of English and hence are used for conversational types of applications — e.g. the IBM laser patent text, basic English, etc. [4,5]. Such vocabularies are inherently very complex and rely heavily on task syntax to resolve recognition ambiguities between similar sounding words. Finally the last range of vocabulary size is the very large vocabulary system with 10,000 words or more. Such large vocabulary sizes are required for office dictation/word processing applications.

Although vocabulary size and complexity is of paramount importance in specifying a speech recognition system, several other issue can also greatly affect the performance of a speech recognizer. The system designer must decide if the system is to be speaker trained, or speaker independent; the format for talking must be specified (e.g. isolated inputs, connected inputs, continuous discourse); the amount and type of syntactic and semantic information must be specified; the speaking environment and transmission conditions must be considered; etc. The above set of issues, by no means exhaustive, gives some idea as to how complicated it can be to talk about speech recognition by machine.

There are two general approaches to speech recognition by machine, the statistical pattern recognition approach, and the acoustic-phonetic approach. The statistical pattern recognition approach is based on the philosophy that if the system has "seen the pattern, or something close enough to it, before, it can recognize it". Thus, a fundamental element of the statistical pattern recognition approach is pattern training. The units being trained, be they phrases, words, or sub-word units, are essentially irrelevant, so long as a good training set is available, and a good pattern recognition model is applied. On the other hand, the acoustic-phonetic approach to speech recognition has the philosophy that speech sounds have certain invariant (acoustic) properties, and that if one could only discover these invariant properties, continuous speech could be decoded in a sequential manner (perhaps with delays of several sounds). Thus, the basic techniques of the acoustic-phonetic approach to speech recognition are feature analysis (i.e. measurement of the invariants of sounds), segmentation of the feature contours into consistent groups of features, and labelling of the segmented features so as to detect words, sentences, etc.

To date, the greatest successes in speech recognition have been achieved using the pattern recognition approach. Hence, for the remainder of this paper, we will restrict our attention to trying to explain how the model works, and how it has been applied to the problems of isolated word, connected word, and continuous speech recognition.

II. The Statistical Pattern Recognition Model

Figure 1 shows a block diagram of the pattern recognition model used for speech recognition. The input speech signal, $s(n)$, is analyzed (based on some parametric model) to give the test pattern, T, and then compared to a prestored set of reference patterns, $\{R_v\}$, $1 \leqslant v \leqslant V$ (corresponding to the

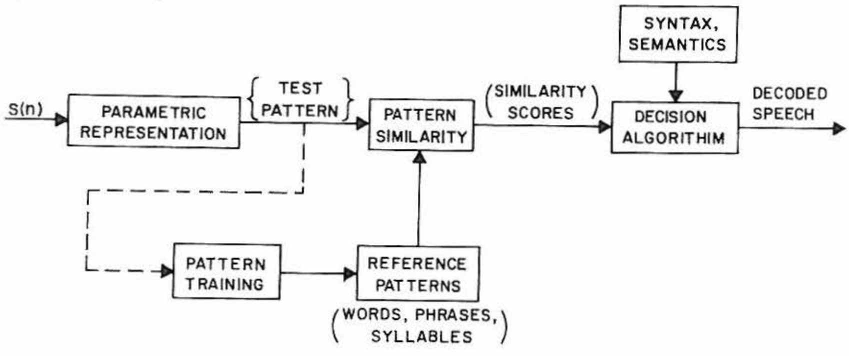


Fig. 1 Pattern Recognition Model for Speech Recognition.

V labelled patterns in the system) using a pattern classifier (i.e. a similarity procedure). The pattern similarity scores are then sent to a decision algorithm which, based upon the syntax and/or semantics of the task, chooses the best transcription of the input speech.

There are two types of reference patterns which can be used with the model of Fig. 1. The first type, called nonparametric reference patterns, are patterns created from one or more real world tokens of the actual pattern. The second type, called statistical reference models, are created as a statistical characterization (via a fixed type of model) of the behavior of a collection of real world tokens. Ordinary template approaches [6], are examples of the first type of reference patterns; hidden Markov models [7,8] are examples of the second type of reference patterns.

The model of Fig. 1 has been used (either explicitly or implicitly) for almost all commercial and industrial speech recognition systems for the following reasons:

1. it is invariant to different speech vocabularies, users, feature sets, pattern similarity algorithms, and decision rules
2. it is easy to implement in either software or hardware
3. it works well in practice.

For all of these reasons we will concentrate on this model throughout this paper. In the remainder of this paper we will discuss the elements of the pattern recognition model and show how it has been used for isolated word, connected word, and for continuous speech recognition. Because of the tutorial nature of this paper we will minimize the use of mathematics in describing the various aspects of the signal processing. The interested reader is referred to the appropriate references [e.g. 6-14].

2.1 Parametric Representation

Parametric representation (or feature measurement, as it is often called) is basically a data reduction technique whereby a large number of data points (in this case samples of the speech waveform recorded at an appropriate sampling rate) are transformed into a smaller set of features which are equivalent in the sense that they faithfully describe the salient properties of the acoustic waveform. For speech signals, data reduction rates from 10 to 100 are generally practical.

For representing speech signals, a number of different feature sets have been proposed ranging from simple sets, such as energy and zero crossing rates (usually in selected frequency bands), to complex, complete representations, such as the short-time spectrum or a linear predictive coding (LPC) model. For recognition systems, the motivation for choosing one feature set over another is often complex and highly dependent on constraints imposed on the system (e.g. cost, speed, response time, computational complexity etc). Of course the ultimate criterion is overall system performance (i.e. accuracy with which the recognition task is performed). However, this criterion is also a complicated function of all system variables.

The two most popular parametric representations for speech recognition are the short-time spectrum analysis (or bank of filters) model, and the LPC model. The bank of filters model is illustrated in Figure 2. The speech signal is passed through a bank of Q bandpass filters covering the speech band from 100 Hz to some upper cutoff frequency (typically between 3000 and 8000 Hz). The number of bandpass filters used varies from as few as 5 to as many as 32. The filters may or may not overlap in frequency. Typical filter spacings are linear until about 1000 Hz and logarithmic beyond 1000 Hz [9].

The output of each bandpass filter is generally passed through a nonlinearity (e.g. a square law detector or a full wave rectifier) and lowpass filtered (using a 20-30 Hz width filter) to give a signal which is proportional to the energy of the speech signal in the band. A logarithmic compressor is generally used to reduce the dynamic range of the intensity signal, and the compressed output is resampled (decimated) at a low rate (generally twice the lowpass filter cutoff) for efficiency of storage.

The LPC feature model for recognition is shown in Figure 3. Unlike the bank of filters model, this system is a block processing model in which a frame of N samples of speech is processed, and a vector of features is computed. The steps involved in obtaining the vector of LPC coefficients, for a

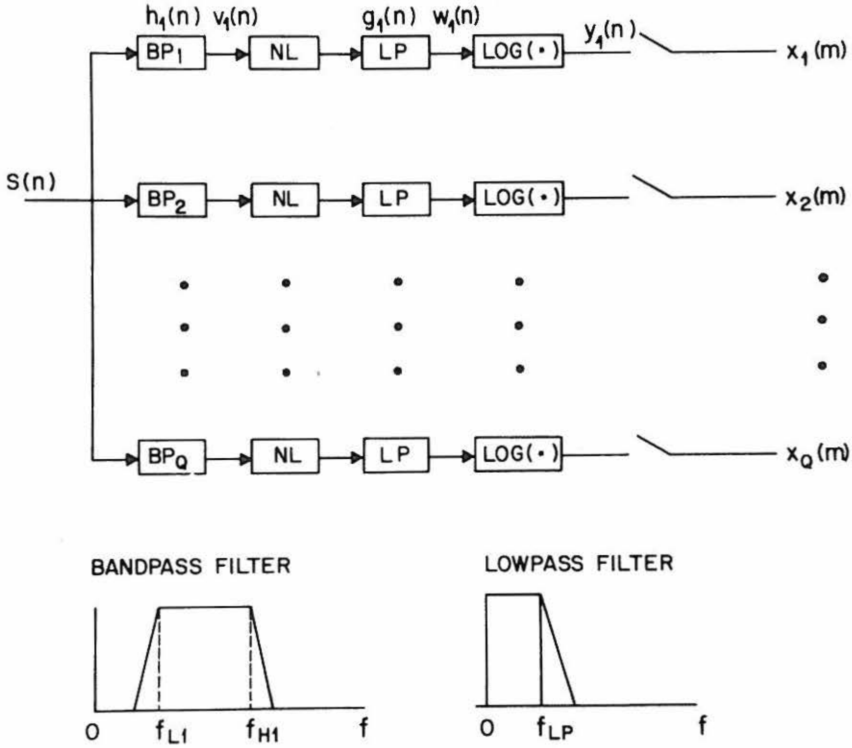
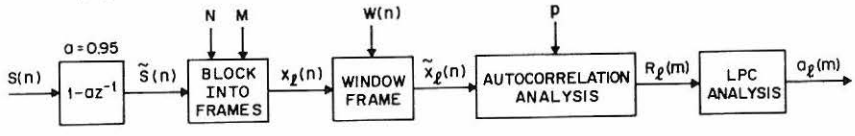


Fig. 2 Bank of Filters Analysis Model.



$$\tilde{s}(n) = s(n) - a s(n-1)$$

$$x_\ell(n) = \tilde{s}(M\ell + n), \qquad \ell = 0, 1, 2, \cdots\cdots, L-1$$
$$n = 0, 1, 2, \cdots\cdots, N-1$$

Fig. 3 LPC Analysis Model.

given frame of N speech samples, are as follows:

1. preemphasis by a first order digital network in order to spectrally flatten the speech signal
2. frame windowing, i.e. multiplying the N speech samples within the frame by an N-point Hamming window, so as to minimize the endpoint effects of chopping an N-sample section out of the speech signal
3. autocorrelation analysis in which the windowed set of speech samples is autocorrelated to give a set of $(p+1)$ coefficients, where p is the order of the desired LPC analysis (typically 8 to 12)
4. LPC analysis in which the vector of LPC coefficients is computed from the autocorrelation vector using a Levinson or a Durbin recursive method [10].

New speech frames are created by shifting the analysis window by M samples (typically $M < N$) and the above steps are repeated on the new frame until the entire speech signal has been analyzed.

The LPC feature model has been a popular speech representation because of its ease of implementation, and because the technique provides a robust, reliable, and accurate method for characterizing the spectral properties of the speech signal.

As seen from the above discussion, the output of the feature measurement procedure is basically a time-frequency pattern — i.e. a vector of spectral features is obtained periodically in time throughout the speech.

2.2 Pattern Training

Pattern training is the method by which representative test patterns are converted into reference patterns for use by the pattern similarity algorithm. There are several ways in which pattern training can be performed, including:

1. casual training in which each individual training pattern is used directly to create either a non-parametric reference pattern or a statistical model. Casual training is the simplest, most direct method of creating reference patterns.
2. robust training in which several (i.e. two or more) versions of each vocabulary entry are used to create a single reference pattern or statistical model. Robust training gives statistical confidence to the reference patterns since multiple patterns are used in the training.
3. clustering training in which a large number of versions of each vocabulary entry are used to create one or more reference patterns or statistical models. A statistical clustering analysis is used to determine which members of the multiple training patterns are similar, and hence are used to create a single reference pattern. Clustering training is generally used for creating speaker independent reference patterns, in which case the multiple training patterns of each vocabulary entry are derived from a large number of different talkers.

The final result of the pattern training algorithm is the set of reference patterns used in the recognition phase of the model of Fig. 1.

2.3 Pattern Similarity Algorithm

A key step in the recognition algorithm of Fig. 1 is the determination of similarity between the measured (unknown) test pattern, and each of the stored reference patterns. Because speaking rates vary greatly from repetition to repetition, pattern similarity determination involves both time alignment (registration) of patterns, and once properly aligned, distance computation along the alignment path.

Figure 4 illustrates the problem involved in time aligning a test pattern, $T(n)$, $1 \leqslant n \leqslant NT$ (where each $T(n)$ is a vector), and a reference pattern $R(m)$, $1 \leqslant m \leqslant NR$. Our goal is to find an alignment function, $m = w(n)$, which maps R onto the corresponding parts of T. The criterion for correspondence is that some measure of distance between the patterns be minimized by the mapping w. Defining a local distance measure, $d(n,m)$, as the spectral distance between vectors $T(n)$ and $R(m)$, then the task of the pattern similarity algorithm is to determine the optimum mapping, w, to minimize the total distance

$$D^* = \min_{w(n)} \sum_{i=1}^{NT} d(i, w(i)) \tag{1}$$

The solution to Eq. (1) can be obtained in an efficient manner using the techniques of dynamic programming. In particular a class of procedures called dynamic time warping (DTW) techniques, has evolved for solving Eq. (1) efficiently [6].

The above discussion has shown how to time align a pair of templates. In the case of aligning statistical models, an analogous procedure, based on the Viterbi algorithm, can be used [7,8].

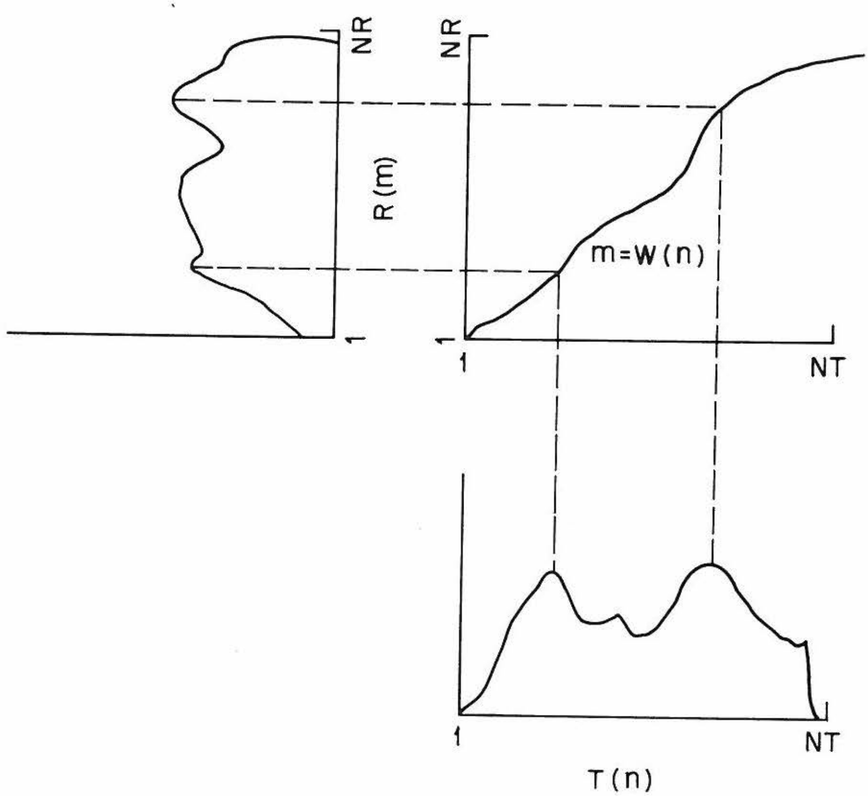


Fig. 4 Example of Time Registration of a Test and Reference Pattern.

2.4 Decision Algorithm

The last step in the statistical pattern recognition model of Fig. 1 is the decision algorithm which utilizes both the set of pattern similarity scores (distances) and the system knowledge, in terms of syntax and/or semantics, to decode the speech into the best possible transcription. The decision algorithm can (and generally does) incorporate some form of nearest neighbor rule to process the distance scores to increase confidence in the results provided by the pattern similarity procedure. The system syntax helps to choose among the candidates with the lowest distance score by eliminating candidates which don't satisfy the syntactic constraints of the task, or by deweighting extremely unlikely candidates. The decision algorithm can also have the capability of providing multiple decodings of the spoken string. This feature is especially useful in cases in which multiple candidates have indistinguishably different distance scores.

2.5 Summary

We have now outlined the basic signal processing steps in the pattern recognition approach to speech recognition. In the next sections we illustrate how this model has been applied to problems in isolated word, connected word, and continuous speech recognition.

III. Results on Isolated Word Recognition

Using the pattern recognition model of Fig. 1, with an 8^{th} order LPC parametric representation, and using the non-parametric template approach for reference patterns, a wide variety of tests of the recognizer have been performed with isolated word inputs in both speaker dependent (SD) and speaker independent (SI) modes. Vocabulary sizes have ranged from as small as 10 words (i.e. the digits zero-nine) to as many as 1109 words. Table I gives a summary of recognizer performance under the conditions discussed above. It can be seen that the resulting error rates are not strictly a function of vocabulary size, but also are dependent on vocabulary complexity. Thus a simple vocabulary of 200 polysyllabic Japanese city names had a 2.7% error rate (in an SD mode), whereas a complex vocabulary of 39 alphadigit terms (in both SD and SI modes) had error rates of about 21%.

Table I also shows that in cases where the same vocabulary was used in both SD and SI modes (e.g. the alphadigits and the airline words), the recognizer gave comparable performances. This result indicates that the SI mode clustering analysis, which yielded the set of SI templates, was capable of providing the same degree of representation of each vocabulary word as either casual or robust training for the SD mode. Of course the computation of the SI mode recognizer was comparably higher than that required for the SD mode since a larger number of templates were used in the pattern similarity comparison.

Vocabulary	Mode	Error Rate (%)
10 Digits	SI	1.8
37 Dialer Words	SD	
39 Alphadigits	SD	20.5
	SI	21.0
54 Computer Terms	SI	3.5
129 Airline Words	SD	12.0
	SI	9.0
200 Japanese Cities	SD	2.7
1109 Basic English	SD	20.8

Table 1

Performance of Template-Based Isolated Word Systems

The results in Table I are based on using word templates created from isolated word training tokens. Studies have shown that when adequate training data is available, the performance of isolated word recognizers based on statistical models is comparable to or better than that of recognizers based on templates. The main issue here is the amount of training data available relative to the number of parameters to be estimated in the statistical model. For small amounts of training data, very unreliable parametric estimates result, and the template approach is generally superior to the statistical model approach. For moderate amounts of training data, the performance of both types of models is comparable. However, for large amounts of training data, the performance of statistical models is generally superior to that of template approaches because of their ability to accurately characterize the tails of the distribution (i.e. the outliers in terms of the templates).

IV. Connected Word Recognition Model

The basic approach to connected word recognition from discrete reference patterns is shown in Fig. 5. Assume we are given a test pattern **T**, which represents an unknown spoken word string, and we are given a set of V reference patterns, $\{R_1, R_2, \ldots, R_V\}$ each representing some word of the vocabulary. The connected word recognition problem consists of finding the "super" reference pattern, $\mathbf{R}^s$, of the form

$$\mathbf{R}^s = R_{q(1)} \oplus R_{q(2)} \cdots R_{q(L)}$$

which is the concatenation of L reference patterns, $R_{q(1)}, R_{q(2)}, \ldots, R_{q(L)}$, which best matches the test string, **T**, in the sense that the overall distance between **T** and $\mathbf{R}^s$ is minimum over all possible choices of L, $q(1), q(2), \ldots, q(L)$, where the distance is an appropriately chosen distance measure.

There are several problems associated with solving the above connected word recognition problem. First we don't know L, the number of words in the string. Hence our proposed solution must

CONNECTED WORD RECOGNITION FROM WORD TEMPLATES

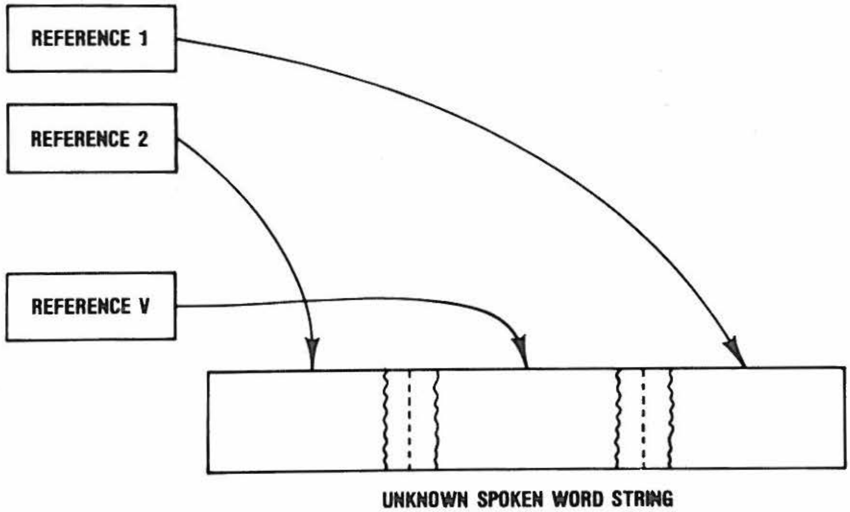


Fig. 5 Illustration of Connected Word Recognition from Word Templates.

provide the best matches for all reasonable values of L, e.g. $L = 1,2, \ldots, L_{MAX}$. Second we don't know nor can we reliably find word boundaries, even when we have postulated L, the number of words in the string. The implication is that the word recognition algorithm must work without direct knowledge of word boundaries; in fact the estimated word boundaries will be shown to be a byproduct of the matching procedure. The third problem with a template matching procedure is that the word matches are generally much poorer at the boundaries than at frames within the word. In general this is a weakness of word matching schemes which can be somewhat alleviated by the matching procedures which can apply lessor weight to the match at template boundaries than at frames within the word. A fourth problem is that word durations in the string are often grossly different (shorter) than the durations of the corresponding reference patterns. To alleviate this problem one can use some time prenormalization procedure to warp the word durations accordingly, or rely on reference patterns extracted from embedded word strings. Finally the last problem associated with matching word strings is that the combinatories of matching strings exhaustively (i.e. by trying all combinations of reference patterns in a sequential manner) is prohibitive.

A number of different ways of solving the connected word recognition problem have been proposed which avoid the plague of combinatorics mentioned above. Among these algorithms are the 2-level DP approach of Sakoe [11], the level building approach of Myers and Rabiner [12], the parallel single stage approach of Bridle et al. [13], and the nonuniform sampling approach of Gauvain and Mariani [14]. Although each of these approaches differs greatly in implementation, all of them are similar in that the basic procedure for finding $\mathbf{R}^s$ is to solve a time-alignment problem between $\mathbf{T}$ and $\mathbf{R}^s$ using dynamic time warping (DTW) methods.

The level building DTW based approach to connected word recognition is illustrated in Fig. 6. Shown in this figure are the warping paths for all possible length matches to the test pattern, along with the implicit word boundary markers $(e_1, e_2, \ldots, e_{L-1}, e_L)$ for the dynamic path of the L-word match. The level building algorithm has the property that it builds up all possible L-word matches one level (word in the string) at a time. For each string match found, a segmentation of the test string into appropriate matching regions for each reference word in $\mathbf{R}^s$ is obtained. In addition for every string length L, the best Q matches (i.e. the Q lowest distance L-word strings) can be found. The details of the level building algorithm are available elsewhere [12], and will not be discussed here.

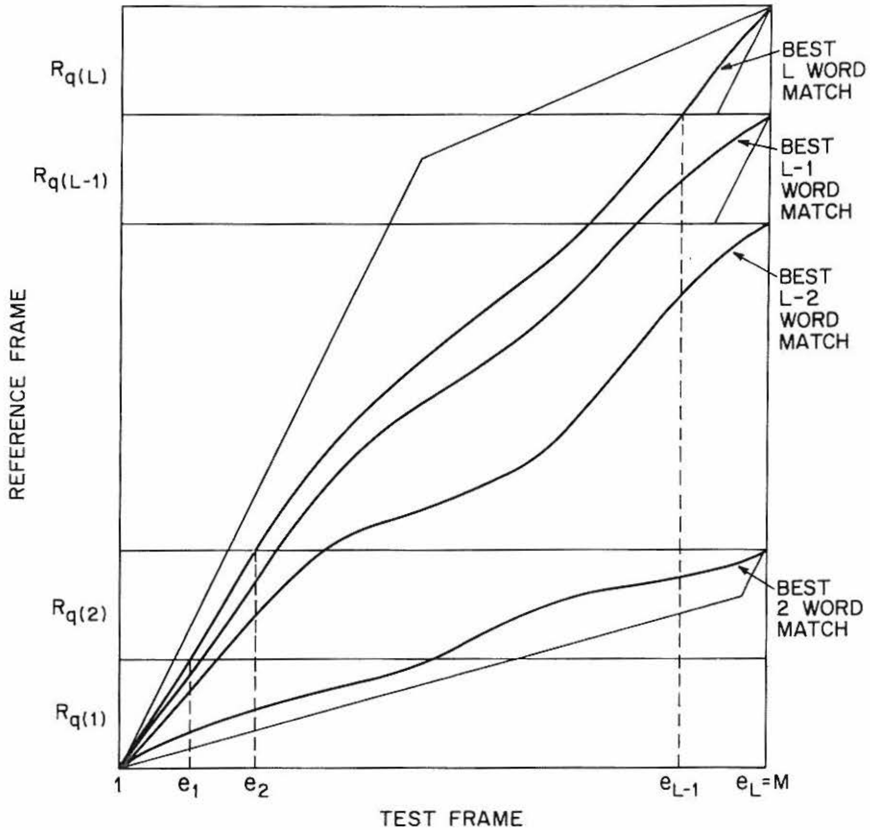


Fig. 6 Sequence of DTW Warps to Provide Best Word Sequences of Several Different Lengths.

Typical performance results for connected word recognizers, based on a level building implementation, are shown in Table II. For a digits vocabulary, digit string accuracies of from 91% to 99% have been obtained in different evaluations. For name retrieval, by spelling, from a 17,000 name directory, string accuracies of from 90% to 96% have been obtained. Finally, using a moderate size vocabulary of 127 words, the accuracy of sentences for obtaining information about airlines schedules is between 75% and 87%. Here the average sentence length was close to 10 words. Many of the errors occurred in sentences with long strings of digits.

V. Continuous, Large Vocabulary, Speech Recognition

The area of continuous, large vocabulary, speech recognition refers to systems with at least 1000 words in the vocabulary, a syntax approaching that of natural English (i.e. an average branching factor on the order of 12), and possibly a semantic model based on a given, well defined, task. For such a problem, there are three distinct sub-problems that must be solved, namely choice of a basic recognition unit (and a modelling technique to go with it), a method of mapping recognized units into words (or, more precisely, a method of scoring words from the recognition scores of individual word units), and a way of representing the formal syntax of the recognition task (or, more precisely, a way of integrating the syntax directly into the recognition algorithm).

For each of the three parts of the continuous speech recognition problem, there are several alternative approaches. For the basic recognition unit, one could consider whole words, half syllables such as dyads, demisyllables, or diphones, or sound units as small as phonemes or phones. Whole word units, which are attractive because of our knowledge of how to handle them in

VOCABULARY	MODE	WORD ACCURACY	TASK	STRING (TASK) ACCURACY
Digits (10 Words)	Speaker Dependent or Speaker Independent	99% SD 98%SI	2-5 Digit Strings	96% SD* 91% SI*
		> 99% SD	1-7 Digit Strings	99% SD**
Letters of the Alphabet (26 words)	Speaker Dependent or Speaker Independent	≈80%	Directory Listing Retrieval (17,000 Name Directory)	96% SD 90% SI
Airline Terms (129 words)	Speaker Dependent or Speaker Independent	97% SD 93%SI	Airline Information and Reservations	87% SD 75% SI

* Known string length.

** 5 talkers, known string length.

Table II

Performance of Connected Word Recognizers on Specific Recognition Tasks

connected environments, are totally impractical to train since each word could appear in a broad variety of contexts. Therefore the amount of training required to capture all the types of word environments is unrealistic. For the sub-word units, the required training is extensive, but could be carried out using a variety of well known, existing training procedures. A full system would require between 1000 and 2000 half syllable speech units. For the phoneme-like units, only about 30-100 units would have to be trained.

The problem of representing vocabulary words, in terms of the chosen speech unit, has several possible solutions. One could create a network of linked word unit models for each vocabulary word. The network could be either a deterministic (fixed) or a stochastic structure. An alternative is to do lexical access from a dictionary in which all word pronunciation variants (and possibly part of speech information) are stored, along with a mapping from pronunciation units to speech representation units.

Finally the problem of representing the task syntax, and integrating it into the recognizer, has several solutions. The task syntax, or grammar, can be represented as a deterministic state diagram, as a stochastic model (e.g. a model of word tri-gram statistics), or as a formal grammar. There are advantages and disadvantages to each of these approaches.

To illustrate the state of the art in continuous speech recognition, consider the office dictation system discussed in Reference [15]. This system uses phoneme-like units in a statistical model to represent words, where each phoneme-like unit is a statistical model based on vector-quantized spectral outputs of a speech spectrum analysis. A third statistical model is used to represent syntax; thus the recognition task is essentially a Bayesian optimization over a triply embedded sequence of statistical models. The computational requirements are very large, but a system has been implemented using isolated word inputs for the task of automatic transcription of office dictation. For a vocabulary of 5000 words, in a speaker trained mode, with 20 minutes of training for each talker, the average *word* error rates for 5 talkers are 2% for prerecorded speech, 3.1% for read speech, and 5.7% for spontaneously spoken speech [15].

VI. Summary

In this paper we have reviewed and discussed the general pattern recognition framework for machine recognition of speech. We have discussed some of the signal processing and statistical pattern recognition aspects of the model and shown how they contribute to the recognition.

The challenges in speech recognition are many. As illustrated above, the performance of current systems is barely acceptable for large vocabulary systems, even with isolated word inputs, speaker training, and favorable talking environment. Almost every aspect of continuous speech recognition, from training to systems implementation, represents a challenge in performance, reliability, and robustness.

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MODELLING OF BINAURAL INTERACTION - THE STATE OF THE ART

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1. Introduction

Although man can hear reasonably well with one ear only (monaural hearing), hearing with two properly functioning ears (binaural hearing) is superior to monaural hearing in many ways. The reason for this is that, when using both ears, the auditory system can pick up additional information which is contained in the differences of the input signals to the two ears in coded form. The auditory system is capable to decode part of this information and to evaluate it in the course of forming the auditory events, i.e., the final products of auditory perception.

Interest in the effects of binaural hearing and in understanding how interaural interaction within the auditory system really works, has increased again recently because of current progress in microprocessor technology. It now seems feasible to implement some of the auditory system's signal-processing algorithms on microprocessor hardware in real time. Consequently, a number of interesting technical applications have come into reach of today's technology, such as signal-processing pick-up devices and direction finders for sound signals in unfavorable acoustic environments; and - as the most general application - front-end devices for sound-signal processing computers and robots.

In order to be able to port signal-processing techniques of the auditory system onto today's hardware, the algorithms have to be put in mathematical form first, i.e., quantitative models have to be formulated. In the following, we report on the general structure of a model system developed in Bochum. Before that, a listing of some important binaural phenomena is given and briefly commented.

2. Binaural Phenomena

There is a great number of binaural effects which are described in the relevant literature (see, e.g., Durlach and Colburn 1978, Blauert 1983, Blauert 1984, for further references). Here we only mention those which are of major importance with regard to technical applications. The first group belongs to the field of spatial hearing.

When the two input signals to the two ears are sufficiently correlated, a single, precisely localized auditory event will most probably appear in the perceptual space of the listener - "binaural fusion". Differences of the input signals to the two ears (arrival-time and level differences) lead to a lateral shift of the auditory event from the median plane of the listener - "lateralization" -, an effect which is of crucial importance for determining the position of sound sources.

In enclosed spaces there is not only a direct sound signal arriving at the listener's ear directly from the source, but also delayed, single or multiple reflections of different order from the enclosing walls. It is not, however, that an individual auditory event is formed for the direct sound and each reflection - if it were so, we would be unable to determine the position of a sound source in rooms. Instead, most of the times one single auditory event in the direction of the first wavefront is perceived -"law of the first wavefront"-, provided that strong and distinct reflections do not come in earlier than about 1 ms and not later than about 50 to 80 ms after the direct sound - the exact time intervals depending on the level and the kind of the direct signal and its reflections. Reflections that come in very early (less than 1 ms) can co-determine the location of the auditory event -"summing localization". This effect is exploited in stereophony with two loudspeakers radiating highly correlated signals with small inter-loudspeaker time differences, among other differences. Strong, distinct reflections which arrive later than about 50 to 80 ms after the direct sound are perceived as an "echo".

We have mentioned above that in the case of the law of the first wavefront only one auditory event appears. However, this does not mean that the presence of reflections remains unperceived. Reflections, among possible effects on loudness and timbre, may cause an effect which is known as auditory "spaciousness" .It is very desirable with regard to the quality of spaces for musical performances. Auditory spaciousness means that the auditory event increases spatially, e.g., in a concert hall, occupies a larger space than is circumscribed by the visual contours of the orchestra. The effect is heavily dependent on the overall level; it can become so strong that the listener gets the impression of being enveloped by sound.

As these binaural effects in spatial hearing are very important for practical applications, their predictibility by a computer model is desirable. Yet, some binaural effects which play a role in binaural listening in noisy surroundings are even more important; especially those which concern discrimination between different sources and detection and understanding of desired sound signals. We comment on the ones that are most important for practical use.

In a concert, to take an example, we are capable to concentrate on a single instrument, disregarding the other ones at this instant. With one ear occluded, this is considerably harder, if not impossible. The orchestra, then, sounds much less "transparent" than with two open ears. Thus binaural hearing enhances discrimination between competing sound sources and detection of their respective sound signals. In a conversation where different people speak at the same time, such as a lively discussion, it is much easier for a listener to follow a particular speaker when both ears are well functioning. Unilateral hard-of-hearing people have severe problems with speech understanding in these situations, an effect well known as the "cocktail-party effect". Further, in binaural listening the perceptual reverberance is decreased compared to monaural listening - or listening over a monophonic system. Also, the timbre-distorting effect of certain early reflections is considerably smaller binaurally than monaurally ("binaural inhibition of reverberance and coloration", e.g., Danilenko, 1967).

3. Modelling Binaural Interaction

There have been quite a number of attempts in the past to develop conceptual as well as algorithmic models of binaural interaction. Colburn & Durlach (1978) have reviewed the relevant literature up to 1974, Blauert (1983, chapter 4.4.4) reports on later work also - up to 1982. A consense among model builders seems to be as follows: A model of binaural interaction to deal with the binaural phenomena listed in the preceding section must at least incorporate the following functional blocks:

(1) a simulation of the functions of the external ear, including those of the head (skull) and the pinnae;

(2) a simulation of the middle ear;

(3) a simulation of the inner ear, i.e. the cochleae including the receptors and first neurons;

(4) a set of binaural processors which identify interaurally correlated contents of the signals from the two cochleae and measure interaural arrival-time and level differences - within this functional block or in an additional one, algorithms are required which inhibit the directional information from signal components which arrive between 1 and 80 ms after the first wavefront;

(5) a set of algorithms which finally evaluate the information as rendered by the preceding blocks with

respect to the specific auditory task to be simulated.

The model of binaural interaction which is currently under development at the Ruhr University at Bochum - actually, a modular set of computer programs and some specialized hardware - follows this general lay-out.

3.1 External-Ear Simulation

The external ears are linear filters, the transfer functions of which depend on the directions and distances of the real and virtual sound sources. For more complex sound fields, we model the external ears using an artificial head which is exposed to the sound field. Its microphone signals - after appropriate equalization - serve as the input to the further stages of the model system. For sound fields with a limited number of real and virtual sources we also simulate the transfer functions of the external ear on the computer and modify the incoming signals accordingly. Specialized hardware allows for real-time processing, if required for the actual modelling task.

3.2 Middle-Ear Simulation

For the modeling of interaural interaction it seems to be sufficient, most of the time, to model the middle ear as a linear bandpass filter - what we actually do. Yet, research is carried out at our institute by H.Hudde and his associates with the aim of developing a model of the middle ear which allows for correct prediction of the behavior of this organ for frequencies up to 20 kHz. After completion, this model will be part of the system.

3.3 Inner-Ear Simulation

It is the purpose of the inner-ear simulation to model two primary functions, namely, that this organ performs a running frequency analysis of the incoming signals, and that in this organ the (analog) mechanical vibrations of the basilar membrane are transformed into (digital) nerve-firing patterns. In doing so, it has to be considered that both frequency selectivity and analog-to-digital conversion depend on the signal amplitude, i.e., behave nonlinearly.

The simplest approximation for the simulation of the frequency selectivity is by a bank of adjacent band-pass filters with, e.g., critical bandwidth each. We do so when computing speed is more relevant than preciseness. More precise modelling is achieved by modelling the selectivity at each point of the basilar membrane (Allen, 1977; Blauert & Cobben, 1978; Schlichthärle, 1981). In the near future, the amplitude dependence of selectivity will be included into the model by simulation of the active processes which are supposed to be part of the functioning of the cochlea.

A precise simulation of the physiological A/D conversion would require a stochastic receptor-neuron model to convert movement of the basilar membrane into nerve-spikes. We have implemented such models into our system for some simulations of binaural effects, e.g., the one of Duifhuis (1972). For practical applications of the model system, it is often not feasible to process individual nerve impulses. Instead, one can generate deterministic signals that represent the time function of the firing probability of a bundle of nerve fibers (Raatgever & Bilsen, 1977; Blauert & Cobben, 1978). To simplify further, we often assume a linear dependence of the firing probability on the receptor potential. The receptor potential is sufficiently well described for many applications by the time function of the movement of the basilar membrane, half-wave rectified and fed through of first order low-pass with a 800 Hz cut-off frequency. Thereby, among other things, we account for the fact that - in the frequency region above about 1.5 kHz - binaural interaction works on the envelopes rather than on the fine structure of the incoming signals .

3.4 Binaural Processors

In the binaural processors, as they currently stand in our model system, the following operations are performed. A modified, interaural running-cross-correlation function is computed on the two signals from the two inner-ear simulators which originate at corresponding points of the two basilar membranes -, i.e., points which respresent the same critical frequency. Cross-correlation has frequently been hypothesized as a principal for binaural processing (e.g., Danilenko, 1967; Blauert & Cobben; 1978; Stern & Colburn, 1978) and is physiologically evident (Yin et.al., 1986). The modification of cross-correlation consists in the employment of a binaural, contralateral inhibition algorithm (Lindemann, 1986). Further, monaural pathways are included in the binaural processors, to allow for the explanation of monaural-hearing effects. Some details of the binaural processors are given in the following paragraph.

The first stage of the processor is based on the well known coincidence-detector hypothesis by Jeffress (1948). It can, e.g., be illustrated by the assumption of two complementary tapped delay lines - one coming from each ear - the taps of which are connected to coincidence cells which fire on receiving simultaneous exitation from both side's delay lines (see Blauert, 1983, chapts. 3.2.1, 4.4.4 for more details). It can be shown that this stage renders a family of running interaural cross-correlation functions as output (Colburn & Durlach, 1978). Thus we arrive at a three dimensional pattern (interaural arrival-time difference, critical frequency, cross-correlation amplitude) which varies with time and can be regarded as a running binaural activity pattern.

The generation of running cross-correlation patterns is followed by the application of a mechanism of contralateral inhibition, as proposed in its present form by Lindemann. The basic idea of the inhibition algorithm is as follows: Once a wavefront has entered the binaural system through the two ears, it will consequently give rise to an activity peak in the binaural pattern. As of this occurrence, inhibition will instantly be applied to all other possible positions of activity in each band where excitation has happened. In each band were signals are received, the first incoming wavefront will thus more or less suppress possible activity created by later sounds with spectra similar to the first wavefront, such as reflections. The actual amount of inhibition is given by specific weights which vary as a function of position and time, in order to account best for the psychoacoustical data. Inhibition stays on for a couple of milliseconds and then gradually dies away until it is triggered again.

By this concept and specific algorithm of contralateral inhibition, in combination with the inclusion of monaural pathways into the processor, the processing of interaural level differences by the binaural system is properly modelled also. For certain combinations of interaural arrival-time and interaural level differences the model will render multiple peaks in the inhibited binaural activity pattern, thus predicting multiple auditory events - very much in accordance with the psychoacoustical data. See Lindemann (1986) for a detailed description of the algorithms.

3.5 The Final-Evaluation Stage

This stage of our system for modelling binaural interaction represents the more central functions of the auditory system with respect to binaural hearing. In a certain way, it can be thought of as a "multi-purpose computer" - as is the brain itself -, the function of which must be defined with respect to the actual, specific task required. Within the scope of our current modelling, we think of this stage as a pattern recognizer which works on the inhibited binaural activity pattern as delivered by the central processors.

This concept can be applied when the desired output of the model system is a set of parameters as evaluated from the sound field received, such as the number and the positions of the sound sources, the amount of auditory spaciousness, reverberance, coloration etc. Also, if the desired output of the model system is processed signals - such as a monophonic signal which has been improved with respect to its S/N ratio -, the final evaluative stage may produce a set of parameters which consequently serve to control further signal processing.

So far, we have defined pattern-recognition procedures for various tasks in the field of sound localization and spatial hearing, such as lateralization, multiple image phenomena, summing localization, the law of the first wavefront, and auditory spaciousness. Current work deals with binaural pitch effects, dereverberation, decoloration and the "cocktail-party effect". Input from other senses - like the visual and coordinative systems - as well as cognitive effects, could be dealt with in the final-evaluation stage by defining the pattern- recognition procedures appropriately, namely, by making them controllable by relevant hetero-sensual and/or cognitive information. A realization of such processes has not yet been attempted in Bochum.

4. Discussion

The work on a complex, modular software model of binaural interaction has some great advantages for university research. In the first place, it provides the framework for long-term research on the auditory system, helps to structure the achieved psychophysical and physiological results and creates the need for specific new research projects. These attributes are of extreme importance for a university laboratory with its frequent exchange of experienced staff and with students working on their theses.

Since the model is modularly organized, different versions of the individual moduls can easily be written and tried out. For specific modelling tasks, only those modules have to be used which are necessary, thus optimizing the trade-off between required precision and calculation time. Also, practical applications are possible as the system progresses, though the system as a whole may be far from being complete - and may probably never be so.

However, technical problems constantly arise since the progress of the model is slower than the progress in hardware. Currently, we use an HP-1000 system, the CPU and operation system of which had to be exchanged more than once during the years. At present, the computer system is antiquated again and may have to be replaced once more - a task which will again occupy a lot of effort and retard our research.

The obvious power of the general concept of our model system is promising. In the field of spatial hearing we have been successful so far in modelling almost all phenomena which do not require specification of internal noise in the model. Binaural pitch effects are obviously within the reach of the model system, as are - after inclusion of internal noise - just noticeable differences (JNDs) in various binaural auditory tasks.

A goal for the near future is to expand the model system in such a way that it can process sound signals in a similar way as the natural auditory system does it. Improvement of the signal-to-noise ratio relative to monophonic recording should be possible as well as the enhancement of speech intelligibility in noisy, echoic or reverberant surroundings.

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Our work on binaural models will be continued. We greatfully acknowledge continuous support from the DFG. Mrs. Kugler-Kruse kindly checked the English of this article.

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The Effects of Room Noise, Overall Loss, and Sidetone on the Subjective Opinion of Telephone Connection Quality.

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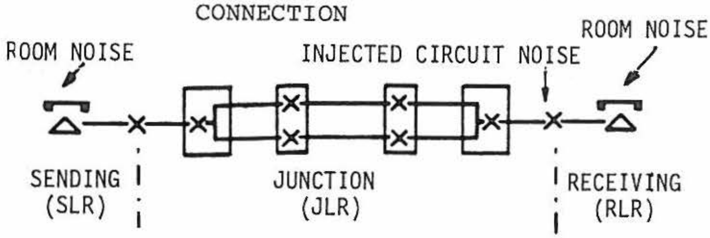
1. Summary

CCITT have for a number of years been studying the problem of telephone sidetone (the extent to which a subscriber hears his own voice during a conversation). It has been argued that some sidetone is necessary but when present it will transmit any room noise to the earphone, thus masking the incoming speech signals. Studies have only comparatively recently begun to address this latter problem, termed 'listener sidetone', as a result largely of recent problems with new electronic telephones and digital exchanges. The main paper will review the very latest results of subjective and acoustic tests aimed at understanding the problem and making recommendations for the control of listener sidetone. What follows is of necessity background information.

2. Introduction

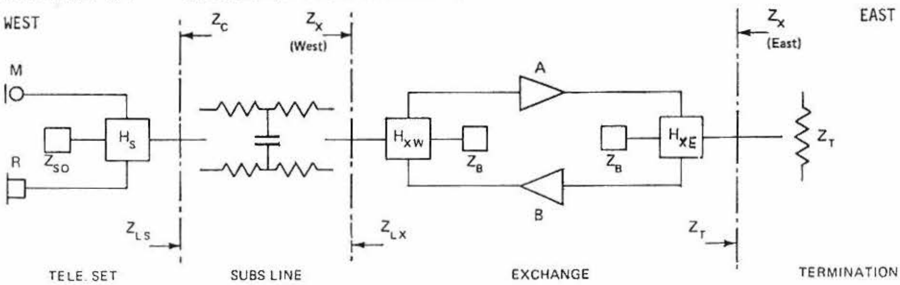
When two subscribers converse over the telecommunication network there are a number of factors which influence their assessment of the connection quality and their ability to communicate. Probably the most important factors are overall loss between the mouth of the talker and the ear of the listener (L_{me}), attenuation frequency distortion, echo, circuit noise and sidetone. All these factors are controlled by International Recommendations. The paper will confine attention to the effects of overall loss, sidetone and noise, including the effects of room noise heard via the sidetone path.

FIGURE 1: A TYPICAL TELECOMMUNICATION CONNECTION



A typical telephone connection is illustrated in Figure 1 in which the overall speech path is broken down into three sections, the two subscribers loops at East and West containing the subscribers instrument and local line up to the local exchange, and the transmission path between these two points. This central portion can contain anything from a pair of wires, for example if the subscribers are on the same local exchange, to a complete international circuit including satellite links. For the purposes of characterising the connection for overall loss L_{me}, which is a function of frequency, is transformed into a single figure, the overall loudness rating (OLR) by the use of a suitable algorithm [1]. This overall path may be broken down into three sub-sections: a sending end, a receiving end, and the electrical transmission path termed the junction in the centre. These sub-sections are characterised according to CCITT practice into a send loudness rating (SLR), a receive loudness rating (RLR) and a junction loudness rating (JLR) and these quantities may each be calculated from the appropriate electro-acoustic sensitivities [2]. Circuit noise can enter the connection at a number of points but is normally assumed for the purposes of any calculations to be entering at the local exchange (Fig. 1). Room noise enters the circuits from the telephone instruments themselves and this will be dealt with in the next section.

FIGURE 2: FACTORS AFFECTING SIDETONE PERFORMANCE



3. Sidetone and Sidetone Paths

The sidetone path, i.e. that acoustic-to-acoustic path through which the subscriber hears his own voice arises as a result of unbalance between the line impedance seen by the telephone set, Zl, and the impedance that the telephone set would like to see in order to reduce sidetone to zero over the whole frequency range, Zso. The complete expression for determining the sidetone sensitivity from the mouth position to the ear is

$$S_{meST} = S_S + S_R - 20 \log_{10} R_{ST} \qquad (1)$$

where $$R_{ST} = (|Z_L + Z_C| \, |Z_C + Z_{SO}|) / (2|Z_C| \, |Z_L - Z_{SO}|) \qquad (2)$$

and S_S and S_R are respectively the sending and receiving sensitivities into matched impedances. All quantities are functions of frequency.

In (2), the quantity Zc is the terminal impedance of the telephone set. Figure 2 shows these quantities together with some other factors that can have an important effect on sidetone performance.

Sidetone sensitivity S_{meST} is normally measured acoustically, [3], with the telephone instrument connected to appropriate line, feed circuit and exchange terminations. S_{meST} may be transformed in a single figure quantifying its apparent loudness to the talking subscriber using another algorithm [2]. This algorithm makes use of the natural sidetone path occuring in the human head as a threshold against which to measure the telephone sidetone path and the resulting figure is the SideTone Masking Rating (STMR). STMR was developed as a result of some subjective tests carried out in the USA [4], and the re-examination of these at a later date [5]. Various subjective tests have shown that STMR is very satisfactory as a rating method for sidetone if one is considering the subscriber as a talker and if the sidetone sensitivity is typical. [6, 7, 8]. However these same subjective tests also suggest that when the subscriber is considered as a listener STMR on its own is not sufficiently accurate as a rating method but needs to be supplemented with a factor that takes into account the effects of local room noise.

4. Room Noise

When a subscriber is listening to a distant party the speech signals from the far end of the connection can be masked by the effects of the room noise reaching the same ear. Room noise reaches the ear via the telephone sidetone path but also by the leakage path under the earcap. For room noise signals reaching the ear this leakage path has been designated L_{RNE} whereas in the direction from the telephone earphone to the ear, the path by which the distant party is heard, it becomes the coupling loss factor L_e. Both L_{RNE} and L_e are variables, depending on how the handset is held, and indirectly on the conversational conditions and factors such as overall loss and the level of the room noise. Little is known about the relationship between L_{RNE} and L_e although some measurements have been reported [9]. These quantities are very difficult to determine and involve the use of probe microphones positioned in the ear of the subjects.

5. Room Noise Sending Sensitivity

When considering the transmission of room noise either to the far end of a connection or to the local earphone via the sidetone path one cannot use the normal sending sensitivity frequency characteristic measured with artificial mouth, S_{mJ}. It is necessary to determine the room noise sensitivity S_{mJ}/RN, using a diffuse sound field. The quantity DELSM, is often used to express the room noise performance of a telephone handset, and is given by,

$$\mathrm{DELSM} = S_{mJ}/RN - S_{mJ} \qquad (3)$$

Generally handsets having the more negative DELSM are more satisfactory under noisy room conditions. Here carbon microphones usually have an important advantage due to their non-linear transfer function (9,10). For linear microphones DELSM is determined mainly by handset length.

6. Conversational Effects

The human feedback mechanisms which operate during a telephone conversation are many and mostly complex. The proper control of sidetone, particularly on high loss connections can be crucial to good conversational conditions. High levels of sidetone tend to cause a reduction in vocal effort on the part of the talker, and this is often accompanied by an increased leak at the earphone/ear interface, thus allowing room noise to enter that ear, but also reducing the received level of the distant party's speech. Obviously this is much more serious for long distance conversations than for local calls where speech signals are high. The interaction between these various factors will be discussed in the light of recent contributions to the study of the subject.

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THE SOUND FIELD FROM A POINT SOURCE ABOVE AN INFINITE IMPEDANCE BOUNDARY AND ABOVE A BOUNDARY WITH AN IMPEDANCE DISCONTINUITY

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It is wellknown since the first decades of this century, by papers of A. Sommerfeld [1] and H. Weyl [2], that the wave field above an infinite homogeneous impedance boundary can be calculated by solving Helmholtz' equation for the velocity potential of the fluid motion with appropriate boundary conditions. For a point source this solution consists of three terms, given for example by A. Wentzel 1974 [3] and S.I. Thomasson 1976 [4]. The first term is independent of the impedance of the boundary and decays as R^{-1}, where R is the distance between the point of observation and the source. The second term is the surface wave, which occurs if the impedance fulfils a certain condition. It decays as $r^{-\frac{1}{2}}$ along the boundary, where r is the horizontal distance between the point of observation and the source, and decays exponentially with the height above the boundary. The third term is given as an integral of which no evaluation in terms of elementary functions is known. This term is referred to as the integral contribution of the field.

Recently, papers [5],[6],[7],[8] have appeared on the calculation of the sound field from a point source above a surface with an impedance discontinuity. Some of these papers treat the case of an impedance discontinuity as a perturbation of the case of a homogeneous boundary surface. In other papers a more fundamental approach is made, but they have another serious shortcoming: if the impedances of both sides of the discontinuity are put equal, the wellknown abovementioned three terms of the sound field from a point source above a homogeneous impedance boundary are not obtained in a recognizable form. This is because these three terms, in papers dealing with a homogeneous impedance boundary, are calculated by means of cylindrical coordinates, which are not useful in the case of an impedance discontinuity.

Thus it is desirable to calculate the sound field from a point source above a boundary with an impedance jump by a method which directly gives the results of Wentzel and Thomasson if the impedances are put equal. The study of this problem showed the necessity of doing the calculation of the sound field from a point source above a homogeneous impedance boundary in a new way, using cartesian instead of cylindrical coordinates. In Ref. 9 it is found in this way that the integral contribution of the field is given by a new expression, the identity of which with the formerly known expression is not at all apparent. It turns out that the new expression has some definite advantages in comparison to the old one. The first advantage is that the surface wave term is separated from the integral contribution in a more direct way with use of the new expression. The second advantage is that the new expression can be expanded asymptotically with coefficients in closed form for large distances (compared to the wavelength) from the source. All hitherto known asymptotical and/or approximate expressions for the old integral term are special cases or first few terms of this expression for the new term. The third advantage is that the condition, given e.g. by Thomasson [4], on the admittance (= inverse of impedance) ν for the existence of a surface wave can be replaced by a weaker and simpler condition, namely $\mathrm{Im}\nu < 0$, using the new integral term. It is also showm why the old more complicated condition is too restrictive.

Because of the use of cartesian coordinates on the boundary surface this new treatment of the homogeneous impedance boundary problem is easily generalized to the case of a point source above a boundary with an impedance discontinuity [10]. By means of Fourier transforms the boundary value problem of Helmholtz' equation is transformed into a singular integral equation, which can be solved exactly by a method given by N.I. Muskhelishvili. The exact solution for the sound field is then given by triple integrals and double integrals. For the case when the source and the observer are situated on the ground and the distance Y from the discontinuity to the source is many wavelengths and much smaller than the distance y from the discontinuity to the observer these integrals can be evaluated approximately and give results which are qualitatively listed below. If the source is placed where the admittance is ν_1 and the observer where the admittance is ν_2 the result of the calculation registered by the observer can be qualitatively described in the following way:

1. $\mathrm{Im}\nu_1 > 0$, $\mathrm{Im}\nu_2 > 0$. In this case there is no surface wave. The discontinuity is a weak source of a wave which decays as $r^{-\frac{3}{2}}$ (if there is no discontinuity the wave decays as r^{-2}).

2. $\mathrm{Im}\nu_1 > 0$, $\mathrm{Im}\nu_2 < 0$. The observer registers a surface wave of the same kind as if the surface were homogeneous with admittance ν_2, weakly modified by a correction term which is dependent of the direction and and proportional to $Y^{-\frac{1}{2}}$.

3. $\mathrm{Im}\nu_1 < 0$, $\mathrm{Im}\nu_2 > 0$. The observer registers a surface wave of the same kind as if the surface were homogeneous with admittance ν_1, weakly modified by a correction term which is dependent of the direction and proportional to $Y^{-\frac{1}{2}}$.

4. $\mathrm{Im}\nu_1 < 0$, $\mathrm{Im}\nu_2 < 0$. The observer registers a surface wave created by the ν_2 surface of the same kind as in p. 2 and a surface wave created by the ν_1 surface and strongly dependent of the direction.

These results are quantitatively accounted for in Ref. 10. They suggest a number of experiments with point sources above discontinuity surfaces with and without surface waves.

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SCALE MODEL SIMULATION OF OUTDOOR SOUND PROPAGATION UNDER THE INFLUENCE OF SOUND SPEED GRADIENTS.

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Introduction.

Sound waves in the atmosphere above the ground are refracted due to the increase or decrease of the sound speed with height. Already at 25 m distance from the source, effects on the sound level have been observed, [1].

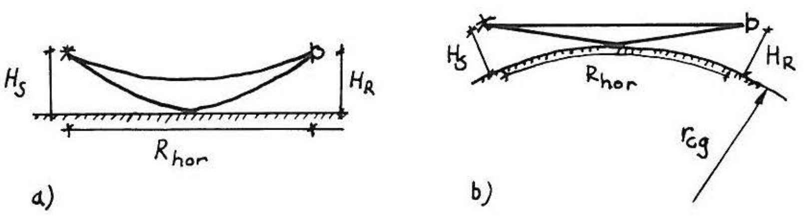

Figure 1. (a) If the sound speed e.g. decreases with height, the sound rays will bend upward. Will a sound pressure measurement according to (b) give the same result as in (a)? The definitions of the source height, receiver height, and horizontal distance are shown for the two cases.

At Chalmers University of Technology, it is at present investigated if it is possible to simulate the influence of a sound speed gradient on outdoor sound propagation, by curving the ground in a scale model instead of the sound rays, see figure 1. The method has previously been rejected by others.

Firstly, a ray-theoretical study was made in order to find out if any geometrical or propagational errors are made with the method. Secondly, measurements of the sound pressure relative to free field above a curved hard surface were made and compared with what is expected theoretically above a flat hard surface with a constant sound speed gradient in the medium. Later, it is planned to do the comparison with a soft surface. By choosing a hard surface the problem of determining the impedance of the ground is separated from the test of the curved ground method. The reflection from a hard smooth surface is governed by the acoustic boundary layer impedance, which can be calculated, [2] and [3].

Ray-theoretical investigation.

In [4] the results of a ray-theoretical treatment of the problem is presented. As a first approximation it is assumed that the sound-speed-gradient is constant with height, which leads to circularly curved sound rays. Rasmussen [5] achieved relatively good agreement between measured (in full scale) and calculated sound pressure relative to free field. Larsson et al., [1], have classified their measured data under a constant gradient assumption.

The radii of the curved rays vary with the angle of emission from the source and it was decided to select the radius of the curved ground, r_{cg}, equal to the radius (in scale 1:s) of the limiting ray which grazes the ground, i.e.

$$r_{cg} = c_0/(s\,dc/dz) \qquad (1)$$

c_0 is the sound speed at ground level and dc/dz is the sound speed gradient. A negative radius in (1) means that the surface shall be convexly curved, which is the case shown in figure 1.

The systematic errors inherent in the curved ground method were examined with ray theory. They were found to be negligible for the geometries and gradients which are important for noise propagation problems outdoors.

Calculation of the sound pressure relative to free field above a flat impedance surface in a temperature-stratified medium.

If it can be shown that the measured sound pressure relative to free field above a curved surface is approximately equal to the corresponding calculated sound pressure above a flat surface in a medium with a constant sound speed gradient, then the hypothesis that the curved ground method gives a good prediction is further strengthened.

Rasmussen, [5], presents the theory from Pridmore-Brown and Pierce [6]. The sound pressure relative to free field can be calculated by numerical integration of

$$\left|\frac{p}{p_{ff}}\right| = \left|-2\ R_{hor} \int_0^{\infty} J_0(\kappa R_{hor}) P(\kappa)\kappa d\kappa\right| \qquad (2)$$

where J_0 is the Bessel function of first kind and zero order. R_{hor} is the source to receiver distance and P is a complicated function of the integration variable kappa, the source and receiver height, the sound speed at ground level, the sound speed gradient and the impedance (or its inverse, the admittance) of the ground. Rasmussen performed the numerical integration of equation (2) and compared with the results of an approximate theory.

<u>Measurement of the sound pressure relative to free field above a curved surface.</u>

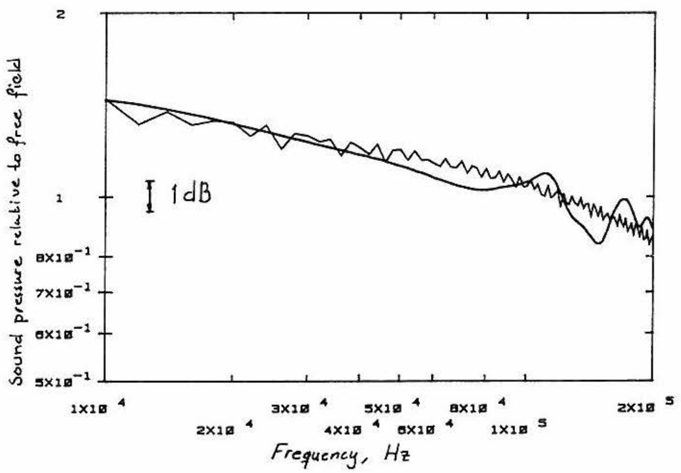

Figure 2. Thick line: measured sound pressure relative to free field from a spark source above a cylindrically curved 4 mm perspex sheet. $1/R_{cg} = -0.1471\ m^{-1}$. Thin line: calculated sound pressure relative to free field from a point source above a solid flat surface in air with a linear decrease of sound speed with height. $(1/c_0)dc/dz = -0.1471\ m^{-1}$. The wiggling of the calculated graph is caused by truncation errors.

Using the measurement system described in [3], the sound pressure relative to free field from a spark source above a cylindrically curved surface was determined. The scale model ground was the surface of a 4 mm perspex sheet which was convexly curved by placing the sheet on curved supports with a precalculated curvature. The actual radius of curvature was checked and the inverted radius was $-0.1471\ m^{-1}$ for the case presented here. If the scale is 1:100 this corresponds to the sound speed gradient $-0.5\ s^{-1}$. This radius was selected because it corresponds to a rather extreme upwind case, [1]. For details concerning the experimental procedure and the calculations the reader is referred to [7].

In figure 2 is shown the measured sound pressure relative to free field for a case when the microphone is just above the shadow zone border (i.e. it is not in the shadow zone). The calculated response is shown in the same figure.

Conclusions

The measured and calculated response in figure 2 as well as for other source and receiver combinations above the same curved surface (not presented in this paper) agrees rather well. So still the method of curving the ground instead of the sound rays is very promising.

Acknowledgements

The curved ground method has been suggested by Professor Lindblad in Lund, Sweden. My colleagues at the department of Building Acoustics are thanked for their interest and advice. The study has been financially supported by the National Swedish Environment Protection Board.

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OPTIMERING AF STØJAFSKÆRMNINGERS UDFORMNING

Civilingeniør Ole Rasmussen
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8000 Århus C

I takt med den stigende vejtrafik og skærpede krav til miljøforhold øges behovet for effektive miljøforbedrende foranstaltninger.

En af de mest åbenlyse miljøfaktorer er støj. Støjen dæmpes ofte ved opsætning af støjskærme langs vejene. Disse støjskærme fremstilles ofte af meget tunge og gedigne materialer, hvilket resulterer i høje anlægsudgifter.

Akustisk set er støjskærmene voldsomt overdimensioneret, idet lydisolationen for skærmvæggen er meget høj. Sammenlignet med, at den maksimalt opnåelige indsatsdæmpning for en støjskærm er 15-20 dB, er det nærliggende at anvende lettere materialer.

Der findes en lang række materialer, der normalt anvendes i andre sammenhænge, men kan anvendes til enkle skærmkonstruktioner. De akustiske data for disse materialer er imidlertid ikke kendte, og egnetheden kan derfor være svær at vurdere.

Laboratoriemålinger

Da der savnedes akustiske data for aktuelle materialer, måtte disse fremskaffes.

Skærmkonstruktionerne med stolper og fittings monteredes i en ca. 10 m^2 stor måleåbning i laboratoriet. Monteringen blev foretaget så "naturtro" som muligt uden ekstra tætning af utætheder og med et realistisk antal stolper og stivere.

Selve reduktionstalsmålingen blev foretaget ved anvendelse af vort sædvanlige målesystem. Der er således anvendt to højttalerpositioner i senderummet og roterende mikrofoner i såvel senderum som modtagerum.

Beregningsprincip

Ved anvendelse af traditionelle skærmtyper bestemmes indsatsdæmpningen af skærmens højde, placering i forhold til støjkilde og modtager samt skærmens tykkelse.

Ved anvendelse af de lette skærmmaterialer vil den lyd, der passerer gennem skærmen, ofte sætte en øvre grænse for den opnåelige indsatsdæmpning. Dette kan beskrives ved formel (1):

$$\Delta L_{sti} = 10 \log \left(10^{-\Delta L_{si}/10} + 10^{(-R_i + 3)/10} \right) \qquad (1)$$

hvor:

ΔL_{sti} er den totale skærmdæmpning

ΔL_{si} er den geometrisk bestemte skærmdæmpning

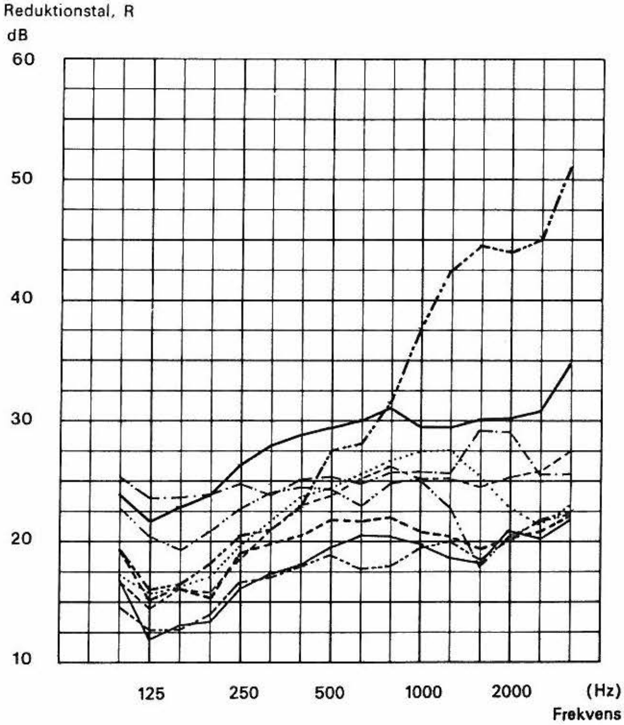
R_i er skærmmaterialets reduktionstal

Formel (1) er gældende for det i'te frekvensbånd.

Både ΔL_{si} og R_i er afhængige af frekvensen, og den endelige vurdering må derfor foretages på grundlag af et standardiseret trafikstøjspektrum.

Skærmtyper

Undersøgelsen omfatter elleve forskellige skærmmaterialer. De målte reduktionstalskurver for otte af disse er vist i figur 1.



Figur 1.

----- 1,5 mm profileret stålplade

—— 2 mm profileret aluminiumplade

—— 1,5 mm stålplade belagt med tyndt lag beton

–--- stålplade + mineraluld + perforeret stålplade

–··– bræddekonstruktion, én på to

–··– flade af spunsjern

–·– 10 mm bølgeeternit

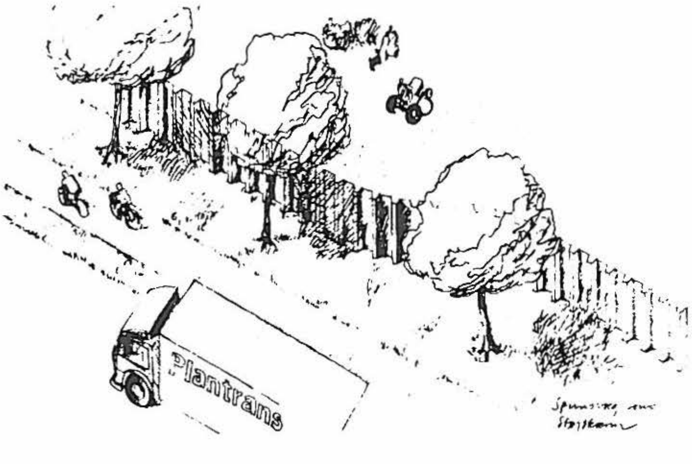
·········· 5/8" ACX krydsfiner

---- flade af luftfyldte plastplanker

Andre faktorer

Foruden de rent akustiske faktorer er der foretaget en vurdering af:

- holdbarhed
- økonomi
- udseende



Resultat og konklusion

Mens dette skrives, er undersøgelsen ikke afsluttet, og nogen endelig konklusion kan ikke gives.

Det er imidlertid åbenlyst, at der findes anvendelsesmuligheder for alle de behandlede skærmkonstruktioner. Hvor den ønskede skærmdæmpning er mindre end 10 dB, og hvor skærmens fysiske dimensioner er små, er det udelukkende økonomiske og æstetiske krav, der bør afgøre valget af skærmmateriale.

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Henrik Møller and Per Rubak

ACOUSTICAL PLANNING OF THE OUTDOOR WARNING SYSTEM IN HELSINKI

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Introduction

In Finland, the outdoor warning systems are maintained not only for alerting of air raids, but also for warning of other catastrophes such as environmental hazards. The city of Helsinki has two outdoor warning systems in operation. The older one dates from World War II and consists of some 60 electrical sirens. This network is not considered fully reliable. The other system has eleven high-power units, which were installed in the 1960's.

It is common practice that planning is included in the offer of devices. In Helsinki, complexity of both the communications network and acoustical properties led to an independent planning project. Our company acts as subconsultant to the telecommunications consultant Kupari Consulting Ltd. As to the present knowledge, planning project pays back its own costs richly.

The work has two objectives, the first one being to reach all outdoor areas with the basic signal, i.e. a simple siren or horn signal. The second objective is an emphasized local warning at selected locations with signal and voice messages.

Signals

The basic signal types are defined nationally by the Ministry of the Interior. Each commune may decide whether it uses horn-like or siren-like devices. The signal messages are defined for both (see any telephone catalogue in Finland). The messages carried by the signals are slightly different for civil defence and during normal times. The two most important for horns are:

Intermittently 2 s sound + 2 s pause,
duration 1 minute.
Constant 1 minute sound.

Comparison measurement

A comparison measurement was arranged to acquire data of different devices and operation principles. Besides this, information was obtained on suppliers' ways to deliver, install, operate and even repair their warning devices in the field.

The test was made on 27 September, 1985, in the centre of Helsinki. The sound propagation environments included both built-up and park areas.

Test arrangement The goal was to obtain comparison data of large units. Measurements were made at large distances (800 m to 1500 m). The mechanism of sound propagation was not of interest and the measurement positions were therefore selected to have good audibility.

To achieve impartiality, all suppliers were invited to participate and the order of sounding and placing of devices were determined by lot. Besides this, the measurement positions were kept secret before the measurements.

Eleven devices of different types participated. Artist's view of these is given in Figure 1. Pintsch is a member of the existing system.

During the test the weather was favourable to sound propagation. The wind was unperceivable and thin clouds covered the sky. Sounds were heard up to a distance of 6 km.

Signals were recorded at 11 positions. Two positions were fixed to enable checks of repeatability of the emitted sounds.

To minimize the effect of changes in sound propagation conditions, one device (the old one) was chosen as a reference. It was sounded between every other device enabling normalization to a near-by reference.

Analysis The tape recordings were analysed with a narrowband filter to separate the signals from background noise. The filters were synthesized in a signal analyser. The "filters" were individually formed for each device so

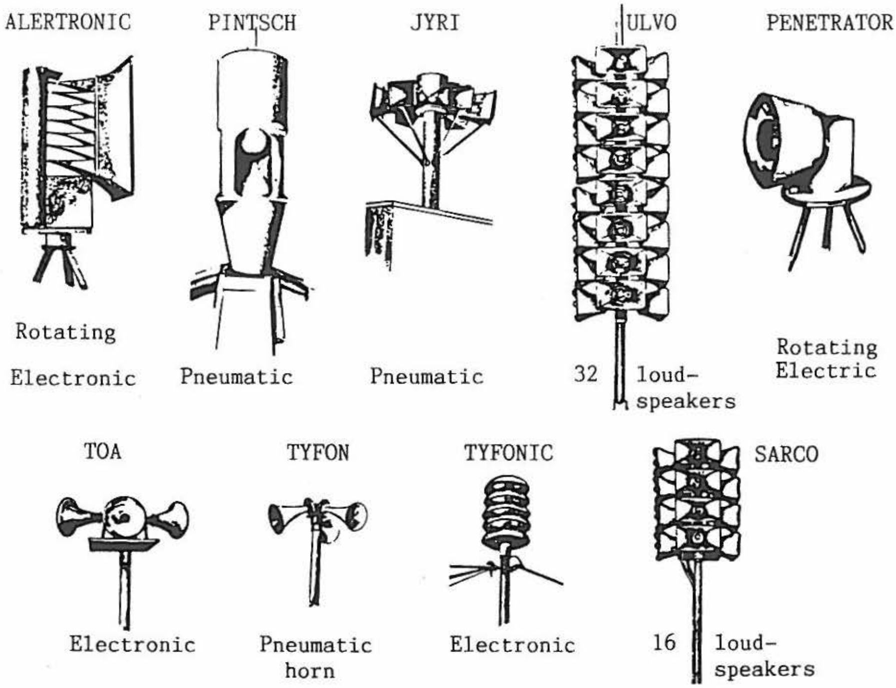

Figure 1. Collection of alarms that participated in the test.

that it captured the sound heard in a near-by tower 800 m from the devices. This goal was achieved to within 1 dB.

The reduction of pink noise in these "filters" was 9 - 13 dB(A). An example of the filtering of the spectra is given in Fig. 2.

Results The A-level difference between the loudest and quietest device was 11 dB. The highest 2 dB class contained six devices. Three of these were electronic and three based on chopped air flow /1/.

The new system

As the message in the basic signal is carried by the on/off sequence, an extra effort is made to avoid confusion of intermittent and continuous sounds: the tone combinations and on/off periods are required to differ in adjacent alarms. There will totally be three different types. The intermission periods are chosen so that during a sounding (one minute) the sounds are in co-phase at least twice.

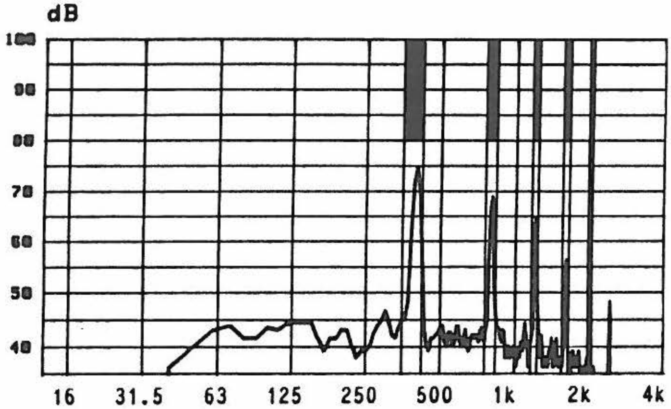

Figure 2. Sample spectrum (Reference device). Black areas show the "filter" used for its analysis. A-weighting included.

Propagation calculations are being made with the joint Nordic calculation model for external noise /2/. This model is adjusted to be valid in slightly favourable sound propagation conditions. These parts of the model need to be converted to correspond to unfavourable cases.

Basic signals Large concentrated units (12 to 14) will be used. Work has started with the renewal of the present devices. Each unit will consist of 6 - 9 horns and use the air supply system of the present ones. The main network is first evaluated and the need for extra units will be decided after this.

Voice messages Locations where local warning is needed are determined by the probability of environmental hazards. They will include road crossings, railyards and harbours where hazardous chemicals are handled.

The decisive factor for voice broadcasting is intelligibility. Large units suffer from garbling by echoes and reverberation. If several units must be used to cover the desired area, they must be connected to the signal so that one avoids two adjacent devices operating simultaneously.

References

/1/ Comparison measurement of high power warning devices 27.9.1985, Final report. Finnish Acoustics Centre Ltd, Technical Report TR 1153-5, Espoo, December 1985.

/2/ Kragh, J., Andersen, B. & Jakobsen, J., Environmental noise from industrial plants. General prediction method. Lydteknisk Laboratorium, Report 32, Lyngby 1982.

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Range of Air Raid Sirens

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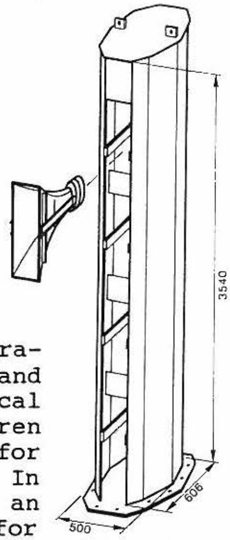

Fig. 1
M84 Siren

1. Introduction

The Danish Acoustical Institute has in cooperation with Danish factories (Vifa, Amplidan and Nea-Lindberg) developed a new electroacoustical siren for the Danish Civil Defence. The siren will be part of a new public alert system for warning a population under indoor conditions. In the horisontal plane the siren will produce an omnidirectional SPL of 138 dB re 20 μPa/1m for 600 W electrical input at frequencies from 200 to 1200 Hz. The system is described in [1] and [2].

In connection with this project computations of the range, within which 80% of the population could hear the sirens, were made. The computations were based on theoretical and practical investigations of sound propagation, sound insulation, internal background noise, and audibility and perception of signals. A result of the computations was, that the SPL outside buildings with sealed double glazings should be higher than 65-68 dB re 20 μPa at frequencies in the range 200-400 Hz. As a difference of approx. 3 dB (e.q. in the estimates of the average sound insulation or the average background noise) would halve or double the necessary number of sirens, an experiment was carried out to determine the range of the sirens in practice.

2. The Practical Experiment

Saturday, december 7th 1985, four prototypes of the new siren were placed in Helsingør (50 km north of Copenhagen). The distances between the sirens were 1400-2500 m and the effective heights were 8-33 m above ground level (fig. 2).

At 2^{15} p.m. a signal (duration 45 seconds) with the fundamental tone sweeping in the range 400-800 Hz (one sweep every two seconds) was sent out. In the subsequent 3 hour period an institute for marketing research (AIM) made telephoneinterviews of approx. 500 persons in 10 selected areas

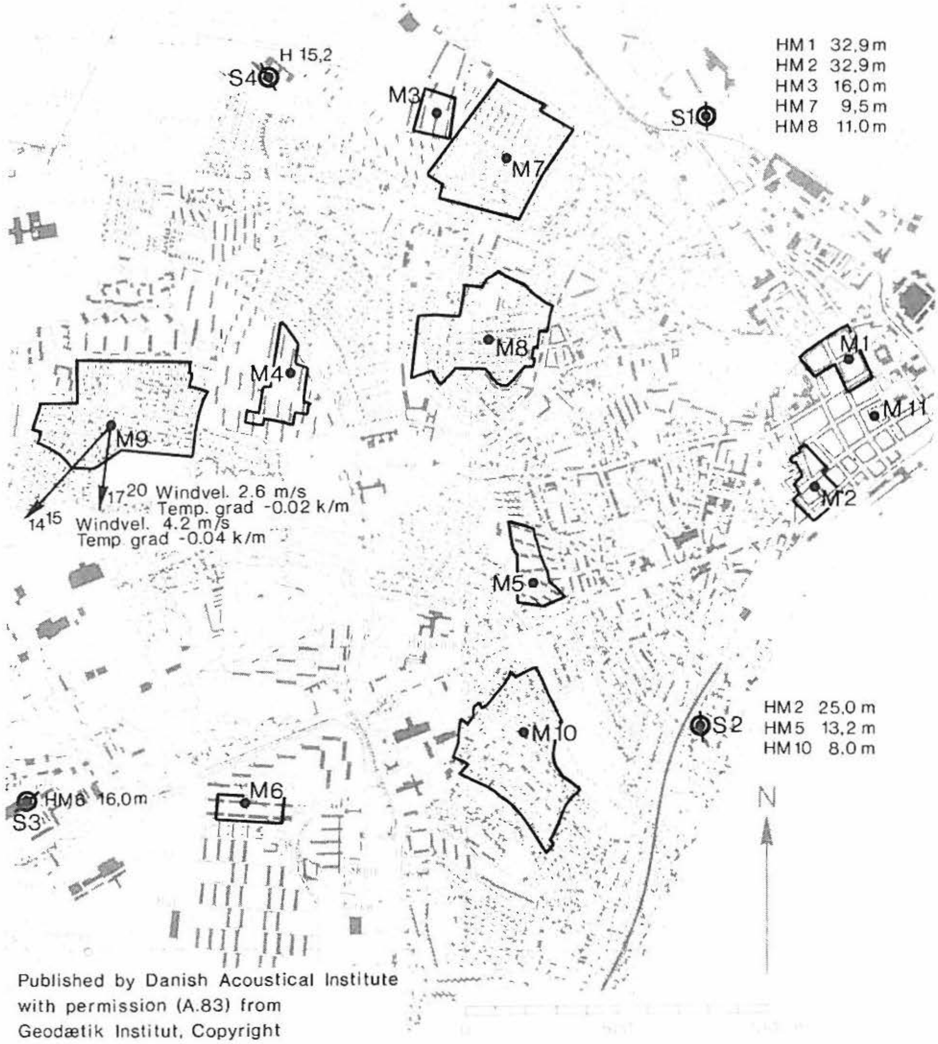

Fig. 2 S1-S4 Siren positions. The figures state the effective heights of the sirens. For some sirens the effective heights are different seen from different areas.

S1

M3 M1-10 Measuring positions.

Limits for interview-areas.

M1-M2 City, 2-3 stories, densely build up
M3-M6 3-storeyed buildings, open build up
M7-M10 1-storeyed buildings, open build up

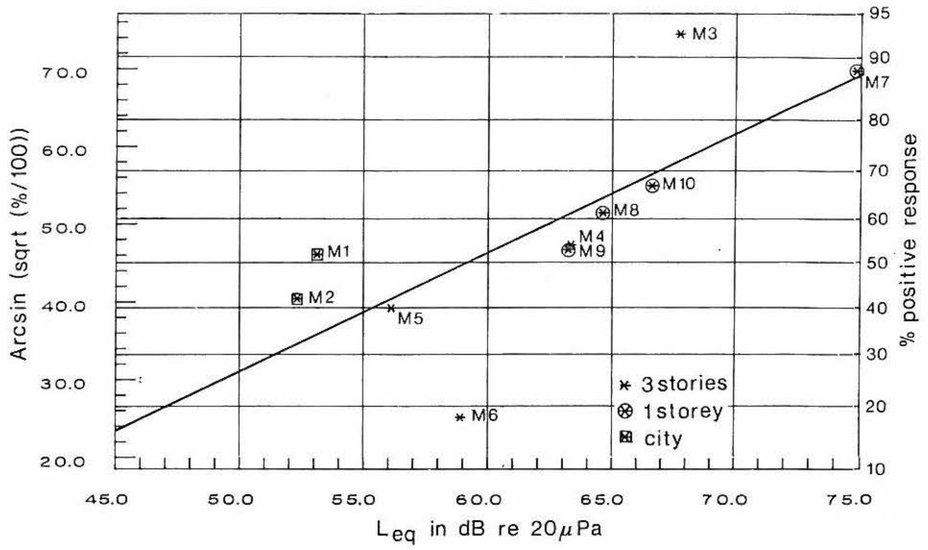

Fig. 3 200-400 Hz signal. 70% ≈ L_{eq} = 66.8 dB re 20μPa

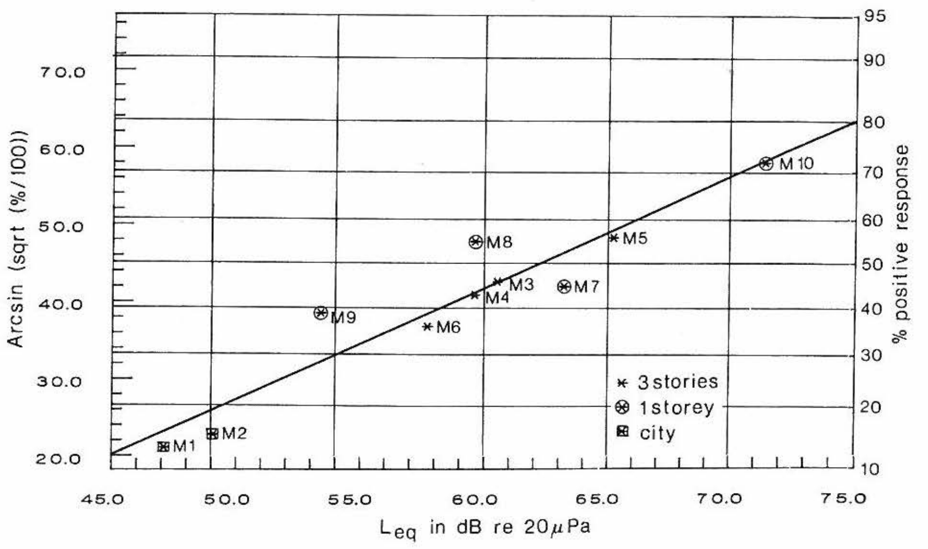

Fig. 4 400-800 Hz signal. 70% ≈ L_{eq} = 70,3 dB re 20μPa

Fig. 3 and 4 The figures show the correlation between the outdoor measured SPL's and the response from persons under indoor conditions with normal background noise. Each figure represents the answers from approx. 200 persons.

between the sirens. The main question was if the person had heard the siren, but also questions concerning background noise, windows (type, open/closed), residence-floor, age and sex were included.

In the same areas calibrated tape recordings of the signal were made in 23 positions for later analyses in the laboratory.

At 5^{20} p.m. another signal, with the same acoustical power, was sent out. This time the fundamental tone was sweeping in the range 200-400 Hz, and the procedure with interviews and tape recordings was repeated. This time the "neighbours" to the first group were interviewed.

3. Results

Statistical analysis (multiple linear regression) was made on the data from the interviews and the tape recordings analyses (L_{eq}, L_{max} and L_{10} of the total signal and L_{eq} of each of the harmonics). The best correlation with the interview data was found with L_{eq} of the total signal, (strongly dominated by the fundamental).

The main results of the analysis is shown in figures 3 and 4. It is seen, that to obtain e.g. 70% positive response for a group of indoor placed persons with normal background noise the outdoor SPL of the 400-800 Hz signal must be 3.5 dB higher than the SPL of the 200-400 Hz signal. Furthermore the attenuation during propagation was higher for the "high-frequency" signal.

The analysis also showed, that listening to radio or TV decreases the probability of hearing the siren with 21% and opening the windows increases the probability with 13%.

It can be computed, that to obtain 80% response (indoor with normal background noise), within the total coverage of the sirens, the outdoor SPL of the 200-400 Hz signal must be 66-67 dB in one-storeyed areas and approx. 68 dB in three-storeyed areas in the border areas.

From these results a model for dimensioning networks of sirens has been set up.

[1] Akustisk varsling, baggrund og principper.
Lydteknisk Institut, rapport 116, 1984.

[2] New design of a long distance public alert system.
Internoise 84 proceedings.

[3] Sireneforsøg - december 1985, Hovedresultater.
Lydteknisk Institut 1986. (Property of the Danish Civil Defence).

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Beregning af ekstern støj fra store virksomheder.

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Introduktion

Miljøstyrelsen accepterer, som noget relativt nyt, beregning af støjimmissionen fra virksomheder på lige fod med målinger.

For en virksomhed er der følgende fordele ved en beregning.

- Komplicerede og dermed kostbare målinger i virksomhedens grundgrænse, som dokumentation for virksomhedens støjimmission, kan undgås.

- Beregningerne skaber oversigt over støjkildernes betydning i valgte immissionspunkter.

- Det er enkelt at opdatere beregningerne ved nedlægning, ændring eller etablering af nye støjkilder.

De data der skal behandles efter "den nordiske beregningsmodel", er meget omfattende og kræver beregninger på en datamat for at blive overkommelige. I det følgende redegøres der for programarkitekturen for et EDB-program, som kan håndtere data for mange enkeltstøjkilder. Programmet er specielt udviklet m.h.p., at alle støjkilder og transmissionsveje bliver behandlet korrekt.

Programbeskrivelse

Det bærende element i programmet er et (x, y, z)-koordinatsystem. I praksis indlægges et (x, y)-koordinatsystem på et stamkort og z vælges som en kote.

Udgangspunktet for beregningerne er en række data opdelt i blokke, som i store træk omfatter:

1. Fabrikdata. Her inddateres luftabsorptionskoefficienten ten, virksomhedens navn, nr., sagsbehandler og oprettelsesdato.

2. Immissionspunktdata, koordinater, jordabsorptionen, navn og nr. fastlægges og inddateres.

3. Støjkildedata for op til 999 støjkilder hver bestående af op til 99 delkilder.

   Dataene omfatter:

   - Støjkildens betegnelse (nr.) og navn, samt beskrivelse af støjkilden.
   - A-vægtet lydeffekt pr. 1/1 oktav i området 63-8000 Hz.
   - Koordinater og jordabsorptionskoefficient.
   - For ikke symmetriske støjkilder også støjkildens orientering til x-aksen og korrektionsværdier for udstrålingsretning (4 hovedretninger).
   - Driftstid i dag, aften og natperioden.

4. Standardstøjkilder. Lydeffektniveauet for en række støjkilder som lastbiler, dozere, ventilatorer, el-motorer m.m. Data herfor ligger i en databse, som A-vægtet lydeffektniveau pr. 1/1-oktav for 63-8000 Hz.

5. Reduktionstal. Reduktionstallet for en række bygningskonstruktioner ligger i en database, som R pr. 1/1-oktav for 63-8000 Hz.

6. Støjafskærmninger. Bygninger m.m., som ventes at yde skærmning, indlægges i en database, idet (x, y, z) for skærmenes højre og venstre kant inddateres.
   Data for støjafskærmninger vælges forsigtigt. Eksempelvis vælges den laveste taghøjde for bygninger med varierende taghøjde.

På grundlag af ovennævnte oplysninger kan støjniveauet i valgte immissionspunkter beregnes for dag, aften og natperioden. Beregningerne udskrives, sorteret efter betydning for hvert immissionspunkt.

Da der er tale om meget store datamængder, typisk 2000 transmissionsveje, er det nødvendigt at benytte statuskoder for at sikre, at transmissionsvejene for alle betydende støjkilder er beregnet med den fornødne nøjagtighed.

Beregning af støjniveauet i immissionspunkter kan opdeles i følgende trin med tilhørende statuskode.

a. Støjniveau beregnet ud fra afstandsdæmpning, luftabsorption og orientering alene. I vor model en statuskode 02-beregning.

b. Som a., men effekten af indlagte støjskærme og spejlkilder indregnes også. Det er en statuskode 05-beregning.

c. Som a., men effekten af skærme, beregnes bedst muligt. Det vil her være nødvendigt at afprøve flere skærmkombinationer for at vælge den "rigtige". Øvrige data for kote til reflektionsplan, jordabsorption, udstrålingskorrektion for den aktuelle transmissionsvej rettes om nødvendigt, og endelig beregning foretages. Det er en statuskode 10-beregning.

For at reducere tidsforbruget til beregningerne mest muligt, er det væsentligt kun at betragte de betydende støjkilder.

Kriteriet for om en støjkilde er væsentlig kan lettest illustreres ved et eksempel.

Der er 400 støjkilder, og kravet til støjniveauet i immissionspunktet er 45 dB.

En støjkilde er væsentlig, når støjniveauet i immissionspunktet overstiger: 45-10 (log 400)-10 = 9 dB.

For dette eksempel er en statuskode 02-beregning tilstrækkelig, når støjniveauet i immissionspunktet er mindre end 9 dB. Er dette ikke tilfældet, foretages en statuskode 05-beregning, og er støjniveauet stadig større end 9 dB afsluttes med en statuskode 10-beregning.

Der vil ofte være behov for at foretage støjdæmpende foranstaltninger for at nedbringe støjniveauet i immissionspunkter.

Beregningerne er opdelt i 2 niveauer. For en første gennemregning kan effekten af støjdæmpning beregnes ud fra valgte dæmpninger i dB(A). Der vil typisk være behov for at dæmpe ca. 50 støjkilder. På dette stade deltager bygherren i valg af støjkilder, som skal dæmpes, idet økonomi, fremtidige produktionsforhold m.m. er afgørende faktorer. Når de orienterende beregninger er afsluttet, udarbejdes et projekt for hver støjkilde, og beregningerne gentages ud fra aktuelle dæmpningsværdier pr. 1/1-oktav.

Afslutning

Programudviklingen kan ikke anses for helt afsluttet, da programmerne løbende udvikles/ændres i takt med erfaringer og klientønsker.

Vore erfaringer med de udviklede sorteringsrutiner, samt beregningsmetoder er gode og må anses for at være et nødvendigt værktøj for at beregne støj fra store virksomheder.

Programudviklingsomkostninger? ca. 500.000,- kr.

Figur 1

Detailberegning for en transmissionsvej.

```
FABR. nr.:    1                                                    13      00/B
NAVN      : Valby Busserviceanlæg
IMM.PKT. :            B : (x,y,z) :   7.2 176.5   3.5    Kote reflekt. plan:   0.0
KILDE nr.: 13        00 : (x,y,z) :  12.0  48.0   8.5    Kote reflekt. plan:   7.5
Kilde-navn: Tagventilator fab. Novenco type ADE-800-K4               K.type:     3
Jordabs. koeff.v. immissionspkt.:   0.0                      Kilde, længde:   0.0
                  midterområde  :   0.0                             bredde:   0.0
                  kilde         :   0.0                    Vinkel t. X-akse :   0.0
Ret.korr.: 0°:   0.0 dB      90°:   0.0 dB    180°:   0.0 dB     270°:   0.0 dB
Frekvens, Hz                    :    63   125   250   500    1k    2k    4k    8k
LwA dB. Std.kilde nr.:      ADE:  69.8  72.1  84.5  94.3  94.1  91.6  82.7  70.0
Redukt.tal bygn.del, R nr.:  0  :   0.0   0.0   0.0   0.0   0.0   0.0   0.0   0.0
Udstrålings korrektion          :   0.0   0.0   0.0   0.0   0.0   0.0   0.0   0.0
Anden korrektion                :   0.0   0.0   0.0   0.0   0.0   0.0   0.0   0.0

Skærm nr.1:           4  Venstre: -19.0  63.5  12.5   Højre  : 190.0  63.5  12.5
Skærm nr.2:              Venstre:   0.0   0.0   0.0   Højre  :   0.0   0.0   0.0

Vegetation, midte (x, y, z)     :   0.0   0.0   0.0     Vegetationsbredde:   0.0
Tidskorrektion (dag,aften,nat)  : 100.0 100.0 100.0 % Bebyg.dæmpn. tr.vej :   0.0
Bygn.reflekt.koeff.,antal flader:   1.0     0
Antal kilder                    :     1
Egensk. bygn. vandret   (x,y,z) :   0.0   0.0   0.0 m
Egensk. bygn. lodret    (x, y ) :   0.0   0.0
```

	LpA	63	125	250	500	1K	2K	4K	8K
Total lydeff. LwA	98.6	69.8	72.1	84.5	94.3	94.1	91.6	82.7	70.0
Reduktionstal	98.6	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
Udstrålingskorr.	98.6	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
Afstandsdæmpning	45.4	-53.2	-53.2	-53.2	-53.2	-53.2	-53.2	-53.2	-53.2
Luftabsorption	44.9	0.0	0.0	-0.1	-0.3	-0.5	-0.9	-2.2	-7.2
Terrænkorrektion	47.9	3.0	3.0	3.0	3.0	3.0	3.0	3.0	3.0
Skærm 1: 4	31.0	-8.4	-10.5	-12.9	-15.6	-18.5	-20.0	-20.0	-20.0
Skærm 2:	31.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
Reflektion	31.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
Vegetation	31.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
Korrektion antal	31.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
Bebyggelses-dæmp.	31.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
Anden korrektion	31.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
Egensk. bygninger	31.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
Kilde-støjdæmpning	31.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
Resultat LpA	31.0	11.2	11.5	21.3	28.3	25.0	20.5	10.3	-7.4

```
Lydtrykniveau LpA (dag,aften,nat) :      31.0          31.0          31.0 dB
------------------------------------------------------------------------------
Eff.sk.højde 1,beregn. og geometr.:      3.75    4.60 m   V.kant-tr.v.:  30.4 m
Afst. kilde - sk.1 - immiss.pkt.  :     15.51  113.08 m   H.kant-tr.v.: 178.5 m
Længde af skærm 1                 :    209.00 m
Skærings-koor. f. sigtelinie/skærm:     11.42   63.50 m

Eff.sk.højde 2,beregn. og geometr.:      3.75    0.00 m   V.kant-tr.v.:   0.0 m
Afst. kilde - sk.2 - immiss.pkt.  :      0.00    0.00 m   H.kant-tr.v.:   0.0 m
Længde af skærm 2                 :      0.00 m
Skærings-koor. f. sigtelinie/skærm:      0.00    0.00 m

Afst.vandr.fra kilde t.immiss.pkt.:    128.59 m
Bemrk.:
```

Figur 2

Deludskrift af beregningsresultat for et immissionspunkt.

Sorteret efter : DAG　　　　Kilde-statuskode >= : 01

Støjkilde nr.		Støjkilde-betegnelse	Status kode	Støjdæmpning dB(A)	Beregnet L(A)eq dB Dag	Aften	Nat
VACU.PM6	00	VACUUMPUMPE PM6	10	0	36.5	36.5	0.0
R0G3.KED	00	RØGGASVENTILATOR KEDEL 3	10	0	36.1	36.1	0.0
VACU.PM7	00	VACUUMPUMPER PM7	10	0	34.7	34.7	34.7
SK-1.KED	00	SKORSTEN,KEDEL 1	30	10	34.1	31.1	28.1
138-.PM7	00	ABEN EXHAUST	10	0	33.3	33.3	33.3
137-.PM7	00	EXHAUST NR 2	30	15	33.3	33.3	33.3
128N.PM7	00	EXHAUST 1 NORD. MASKINRUM PM7	30	10	33.2	33.2	33.2
FATN.PM6	00	NORDFACADE PM6. TILBYGNING	02	0	30.5	30.5	30.5
139-.PM7	00	EXHAUST NR 3	30	10	26.8	26.8	26.8
127N.PM7	00	VAD EXHAUST 3 NORD. PM7	10	0	26.3	26.3	26.3
126S.PM7	00	VAD EXHAUST 2 SYD. PM7	10	0	26.2	26.2	26.2
125S.PM7	00	VAD EXHAUST 1 SYD. PM7	02	0	26.1	26.1	26.1
Total lydniveau for valgt immissions-punkt		1			43.9	43.7	41.6

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PREDICTION OF SOUND LEVELS IN DAIRIES

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Introduction

On a grant from the Swedish Work Environment Fund, a project has been carried out concerning the reduction of noise from pumps and pipings within the food & dairy industry. Included in this work is the development of a computer program for the prediction of noise inside industries, particularly considering the special noise sources of dairy processes. Apart from pumps, separators and homogenizers, a large amount of pipes and valves contribute to the total noise level. In early noise predictions line sources were not included. The new program is capable of handling noise propagation from pipes with the aid of an algorithm for line sources. Effort has also been paid to the reduction of noise from two common types of dairy pumps: a centrifugal pump and a liquid ring pump. Improvements in the order of 4-9 dBA are achieved. Sound intensity measurements are used for the rating of partial sound sources and the follow-up of noise reduction steps.

Noise prediction by computer aid

A statistical model for the sound propagation in factories is described in ref. 1 . The model is simplified to a point rating procedure for a number of parameters affecting the sound propagation, see ref. 2 . This method is applied in the computer program NOISE-CALC and further improved by adding first order image sources to account for noise sources close to walls and corners. Another important feature is the use of a directivity model for line sources, ref. 3 . This is used for pipes, which are divided into straight lines between bends.

Source positioning, directivity and distance calculation is three-dimensional and can be applied to any right-angled room shape. Calculation of resulting A-weighted sound levels is carried out in a square mesh at any level above the floor. Internally, the program calculates in octave bands or, for fast overviews, in dBA. Input data is compiled from a data base of sound power levels for a variety of equipment and source-connected pipes. Finally, iso-level-curves are plotted using a developped interpolation routine. The program has been tested by predicting the noise levels of a dairy production hall and measuring the actual levels in the same square mesh, 1.5 m above the floor. 79 sources were accounted for. The result is shown in figures 1 and 2. Levels are accurate within 2 dB. The importance of including the piping is verified by comparing figures 1 and 3, yielding a difference of about 4 dB.

Pump noise abatement

The use of sound intensity measurements for this work is reported in ref. 4 . Results of different noise reduction steps are listed in Table 1. The pump itself radiates little noise in the absence of cavitation. However, structure borne noise is transmitted from the pump to the casing, which becomes the most important partial source for the liquid ring pump. Also for the centrifugal pump the casing raises the total sound power level. New casings are constructed, using vibration isolation, damping layer, sealing and absorption. Improvement is 9 dB for the liquid ring pump, 4 dB for the centrifugal pump.

Table 1. Noise reduction of a liquid ring pump

Quarter wave stub at pressure side outlet	-3 dBA
Smooth edges of pump house outlet	-2 dBA
Damped casing	-4 dBA
Damped and sealed casing with sound trap	-6 dBA
Damped, vibration isolated casing with absorption	-9 dBA

Random shovel spacing of the pump wheel instead of equal 22.5 degrees proved no sound reduction but was designed to eliminate the pure-tone character of the noise from the liquid ring pump. This was achieved, thereby eliminating a demand for 5 dB lower A-weighted sound level. The original narrow band spectrum is shown in figure 4, the spectrum from the random-spaced wheel is seen in figure 5.

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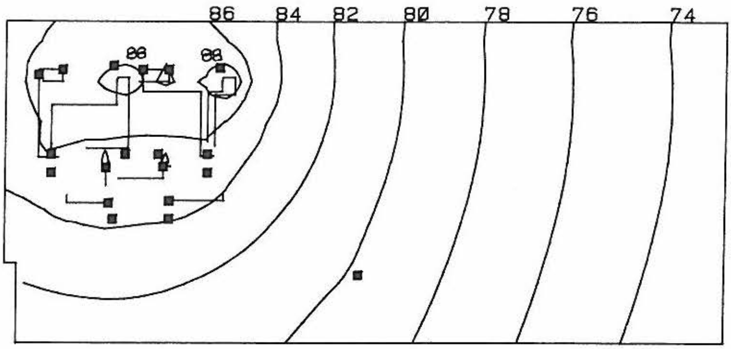

Fig. 1. Calculated sound level contours in dBA for 19 point sources and 60 line sources

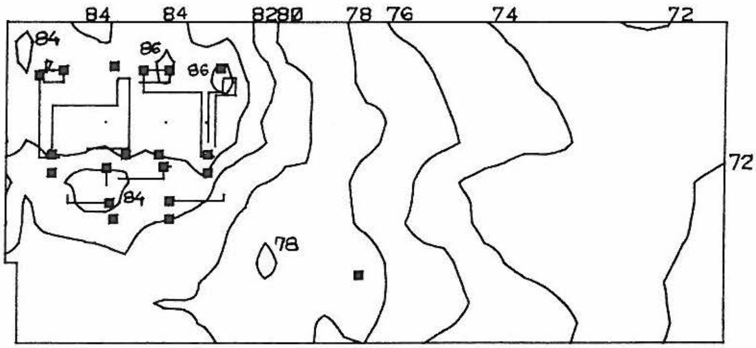

Fig. 2. Measured sound level contours in dBA for sources above

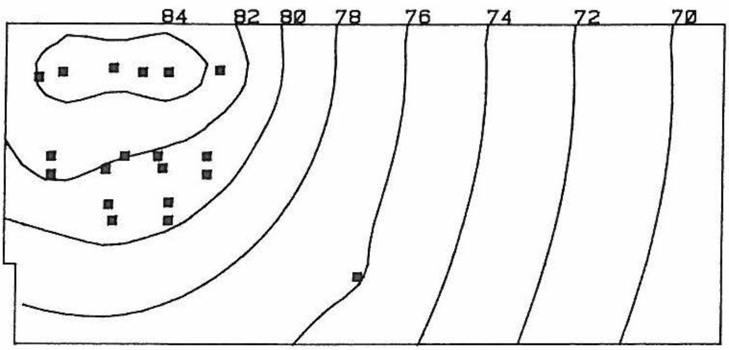

Fig. 3. Calculated sound level contours for 19 point sources, line sources excluded

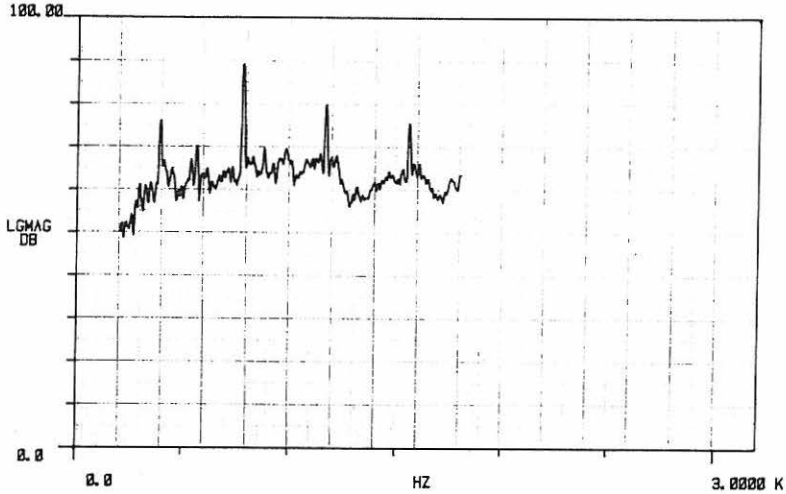

Fig. 4. Pure tone spectrum from liquid ring pump with standard pump wheel. Frequency resolution 3.125 Hz. Pure tone penalty 5 dB.

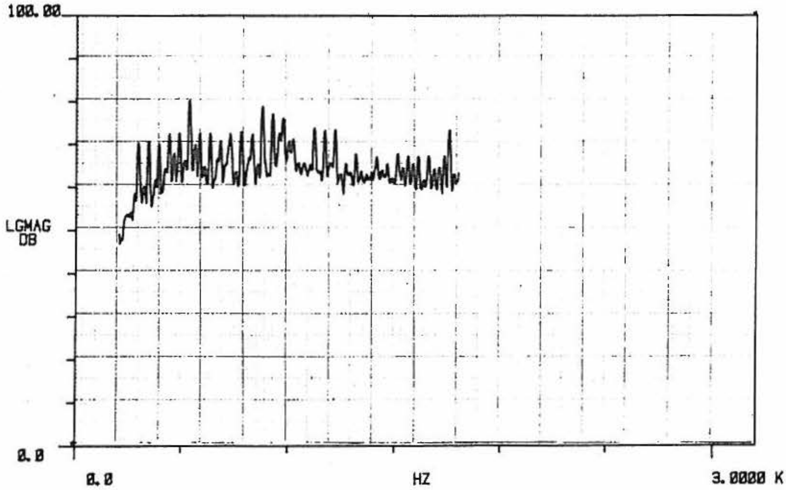

Fig. 5. Spectrum from liquid ring pump with modified pump wheel, random shovel spacing. Experienced by listener as a continuous spectrum. Pure tone penalty 0 dB.

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Noise in offices.

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Introduction

During the last two years the Building Acoustics Section of the Danish Building Research Institute has investigated the noise in a number of offices. The investigations are parts of two research projects. The first project , the Sick Building Project, is a broad investigation of the influence of the physical indoor climate parameters on mans health. The project is initiated by physician Peder Skov at the University Hospital of Copenhagen and the purpose is to try to discover the relations between the socalled Sick Building Syndrome, a complex of symptoms that are significantly higher in some buildings than in others. The project involves a questionnaire with 69 questions concerning work, diseases, symptoms, family and living habits, medical examination of face and hands on the employees and measurement of approximately 30 physical parameters. The project is divided in two parts. In the first part one office in each of 14 buildings is investigated and in part two offices in each of four buildings chosen from the 14 in part one is investigated. The offices are chosen as typical for the buildings. The other project, Information Technology Equipment in Offices, is carried out by the Division of Environmental Design of the Danish Building Research Institute. The purpose is to find recommendations for the design of offices. The project includes case studies of some offices with office automation equipment. The employees in the offices have been interviewed and measurements of thermal and acoustic parameters have been made in some of the offices.
In the following the acoustical measurements and a summary of their results are described. A few results from the questionnaires are also given.

Acoustical Measurements

Statistical distribution of noise:
The A-weighted noise level distribution was measured with a Brüel & Kjær statistical noise analyzer type 4426 with the microphone placed in a single position not close to walls or working places. The distribution was measured in one-hour periods over at least 24 hours. From this the A-weighted equivalent noise level, $L_{A,eq}$, and the N-fractional levels, L_N, in a working day could be calculated. This method was used in part one of the sick building project and in the information technology project. In part two of the sick building project the statistical analysis was carried out with a frequency analyzer.
Frequency distribution of noise:
In the sick building project the frequency distribution in 1/3-octaves was measured using a Brüel & Kjær digital frequency analyzer and a Hewlett-Packard microcomputer with the microphone placed on a rotating boom. In part one of the project the spectrum was measured in a 15 minute period with normal activity in the office. In part two the statistical distribution of the level in each 1/3-octave band and in a linear weighted and a A-weighted channel was calculated in one-hour periods as long time as possible in one working day.
Reveberation time:
In part one of the sick building project the reverberation time, T_{60}, in the 100-3150 Hz 1/3-octaves was measured using the frequency analyzer and the microcomputer. The mean reverberation time was calculated from the 1/3-octave values.
Noise from individual machines:
In part two of the sick building project and in the information technology project the noise from office equipment that emmits noise in longer periods (typewriters, computer terminals etc.) was recorded on tape. The recordings was analysed in the laboratory for contents of pure tones using 1/12-octave or 3% bandwith filters. In some cases the taperecording was supplemented with a measurement of the SLOW-averaged and the PEAK-level of the A-weighted sound pressure level. These measurements were made with the microphone placed at the ear position of the operator.

Results of the measurements

In table 1 the minimum, average and maximum values of the measured values of $L_{A,eq}$, L_{95} and T_{60} (mean value) are given

		minimum	average	maximum
$L_{A,eq}$	(dB rel. 20µPa)	51.2	55.7	60.3
L_{95}	(dB rel. 20µPa)	25.4	35.3	44.1
T_{60}	(s)	0.28	0.40	1.05

Table 1. Minimum, average and maximum values of measured $L_{A,eq}$, L_{95} and T_{60}.

Generally speaking the low levels of L_{95} are measured in offices with little or no office automation equipment. The higher levels measured in offices with this kind of equipment is due to the noise emitted from cooling motors, disc-drive motors and the like. In some cases where the equipment is turned off for some time of the day the working day value of L_{95} is low, but the level measured in the one-hour periods where the equipment is high. In fig. 1 the statistical distribution of L_A is shown for an office with and an office without office automation equipment.

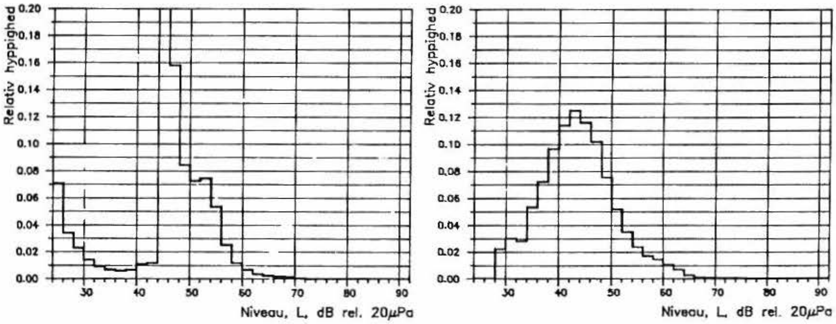

Figure 1. The statistical distribution of the A-weighted sound pressure level for an office with two text processing system terminals (left) and an office with no office automation equipment.

In fig. 2 a frequency-level-time plot of the noise levels in an office investigated in the sick building project is shown. The frequency distribution and the little time variation that is shown in this plot is typical for the offices that has been investigated.

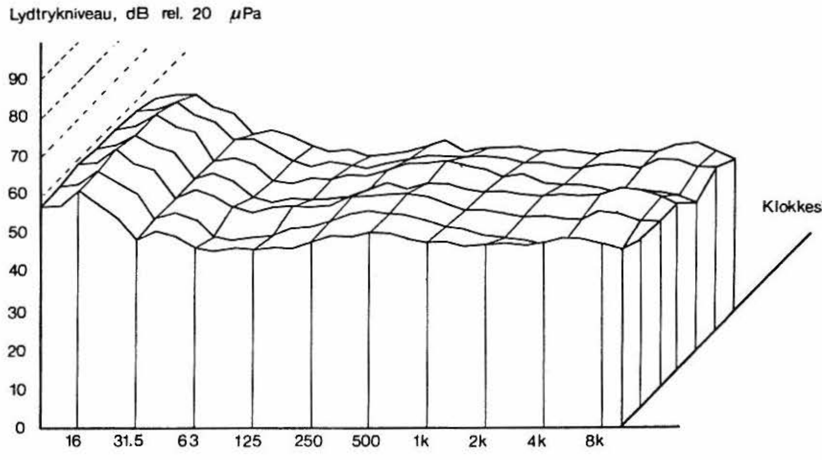

Figure 2. Frequency-level-time plot of the sound pressure level in an office. The time- (z-) axis scale is 1 h/div.

In table 2 the ranges of the values of SLOW-averaged and PEAK values of the A-weighted sound pressure level at the ear position of the operator of different kinds of office equipment are shown. It is also stated if the sound contents single frequencys above the hearing level in the actual noise (pure tones). In fig. 3 a 1/12-octave spectrum from a text-processor with attached disc drive is shown.

	$L_{A,SLOW}$	$L_{A,PEAK}$	Pure tones
Typewriters (el.)	68-70	85-91	No
VDU's without cooling	<41	-	Yes (16 kHz)
Microcomputers (text processors)	37-57	-	Yes
Printers (matrix)	59-74	78-97	No

Table 2. Range of levels in dB rel. 20µPa of slow-averaged and peak value measured from different office equipment.

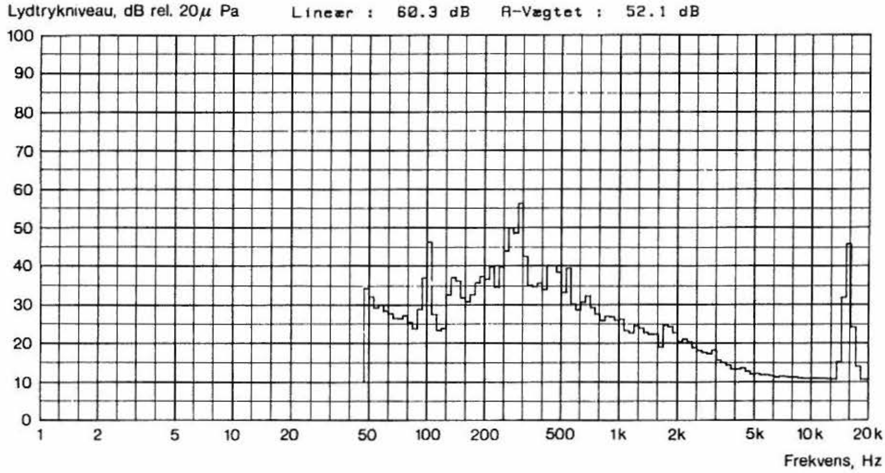

Figure 3. 1/12-octave spectrum measured from text processor with attached disc-drive.

In table 3 the answers on questions concerning noise and a few other parameters for comparance is given. The statistical material from the sick building project has not yet been investigated in detail, but preliminary analysis shows no significant correlation between the measured acoustical parameters and the answers to the questionnaire. On the other hand an often heard answer in the interviews from the information technology project is "I am not annoyed by the noise, but it is very nice when it stops".

Draught	30
Shifting temperature	33
Noise in the room	23
Noise from other rooms	11
Noise from outdoors	8

Table 3. Percent of asked persons that were often annoyed by the mentioned parameter.

Discussion

The results from the investigation described here indicates that the overall noise level in offices only show little variation from office to office. The statistical distribution of the noise differs with the machines that are installed in the offices. There seems to be a trend that with the office automation equipment constant noise sources are introduced that raises the stationary noise level and the pure tone contents of the noise in the offices. There were only little automation equipment in the offices covered by the questionnaire in the sick building projekt.
The litterature only gives a little information about the background noise in offices. Noise begins to interfere with conversational speech at L_{pA} around 40 dB. An A-weighted maximum level of 35 dB is often recommended for installations and ventilation systems. The "Nice when it stops"-answers indicated that stationary noise levels even below 40 dB might be a stress factor in office environments.

Noise should be reduced at the source. Guidelines for the noise emitted by office equipment should be made. Especially attention should be payed to the pure tone contents of the noise. A good noise climate provides good planning.

Litterature

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Støjdæmpning af mobile flishuggere

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INTRODUKTION

Mobile flishuggere anvendes til at sønderdele træaffald fra skove, parker, frugtplantager m.v. I arbejdsområdet omkring en mobil flishugger vil støjniveauet ofte være over 100 dB(A).

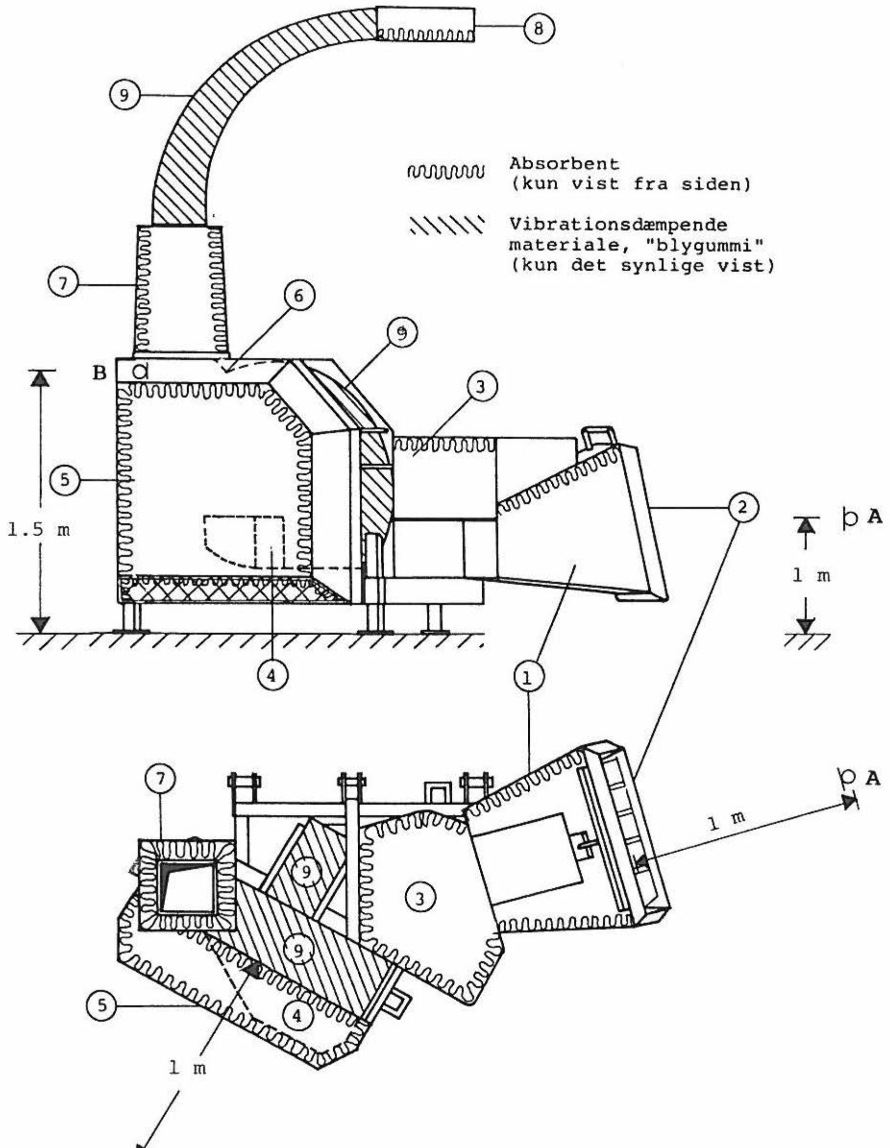
Lydteknisk Institut har i samarbejde med Skovteknisk Institut og en dansk flishuggerfabrikant gennemført et støjdæmpningsprojekt, som blev finansieret af Arbejdsmiljøfondet.

STØJDÆMPET PROTOTYPE

Der blev udviklet en støjdæmpet prototype, og for den bedst støjdæmpede version er den samlede reduktion af de A-vægtede lydtrykniveauer 15-20 dB - afhængigt af måleposition og driftstilstand. Dette betyder, at støjen fra den kraftigt belastede traktor herefter ofte vil bestemme det samlede støjniveau.

På figuren er med nr angivet de gennemførte støjdæmpende foranstaltninger. Ved den efterfølgende kortfattede beskrivelse af disse er desuden angivet målte/anslåede reduktioner af de A-vægtede støjniveauer for de to viste målepositioner A og B.

Det bemærkes, at der er væsentlig forskel på tomgangsstøj og bearbejdningsstøj, idet tomgangsstøj er domineret af luftturbulensstøj med rentoner, medens bearbejdningsstøjen i højere grad skyldes støjudstråling fra de vibrerende flader over et bredt frekvensområde.


8
9
Absorbent
(kun vist fra siden)
Vibrationsdæmpende
materiale, "blygummi"
(kun det synlige vist)
7
6
9
B
3
5
1.5 m
2
A
1 m
4
1
7
A
9
1 m
3
9
5
4
1 m
B

STØJDÆMPENDE FORANSTALTNINGER

1 Lydabsorbenter i indmadningstragtens sider og overpart. Disse er beskyttet af kraftig og afstivet perforeret plade. Reducerer indmadningstragtens "trompetvirkning".

Støjreduktion: A: 5 dB, B: Ingen.

2 Lamelgardin i tragtåbning.

Støjreduktion: A: 2-3 dB, B: Ingen.

3 Tætning af huller og sprækker ved hus omkring hydraulikmotor samt lydabsorbent indvendig i hydraulikhus. Mindsker den direkte udstråling af luftlyd fra skærezonen gennem åbninger omkring fremføringsvalser.

Støjreduktion: A: 2 dB, B: 1 dB.

4 Ekstra lufttilgangsåbning og ledeplader forbedrer blæseevnen og formindsker, især ved tomgang, luftindsugningsstøj ved indmadningstragt. (Se foranstaltning 5).

Støjreduktion: A: 5-6 dB, B: -2 - 1 dB.

5 Lydfælde og afdækning af side af blæserhus reducerer støjudstråling fra lufttilgangsåbninger.

Støjreduktion: A: Ingen, B: 7-10 dB.

6 Afrunding af "tunge" (kant) i blæserhus reducerer, især ved tomgang, tonestøj stammende fra blæservingers passage i lille afstand.

Støjreduktion: A: 1 dB, B: 7 dB.

7 Lydfælde i start af afkastkanal formindsker luftlydens forplantning op gennem afkastkanalen.

Støjreduktion: A: 1 dB, B: 1 dB.

8 Afskærmning og lydabsorbent ved afkaståbning ved vippetud mindsker den direkte udstråling af luftlyd, stammende fra blæserstøj og flis-transportstøj.

Støjreduktion: A: 1 dB, B: 2-5 dB.

9 Vibrationsdæmpende materialer på alle frie tyndplader formindsker især støjudstråling fra vibrerende overflader, når der er træ i maskinen.

Støjreduktion: A: 1-4 dB, B: 1-4 dB.

OVERSLAGSPRISER

Omkostningsniveauer kan kun overslagsmæssigt angives:

- Støjdæmpning af indmadningstragt og hus over hydraulik. Ca. kr. 6.000.
- Ekstra luftindtag og lydfælde på indtagsside plus vibrationsdæmpende materiale på blæserhus. Ca. kr. 4.5000.
- Lydfælder og vibrationsdæmpende materiale på afkastkanal. Ca. kr. 4.000.

Priserne er excl. moms og beregnet af IKAS A/S pr. dec. 1985. Der kan ikke forventes større prisforskelle, om det er støjdæmpning af en stor eller lille mobil flishugger. Ved serieproduktion vil væsentlige besparelser kunne opnås.

GENERELT ANVENDELIGE METODER

De anvendte støjdæmpningsmetoder er generelt anvendelige for de fleste typer af mobile flishuggere. Foranstaltningerne kan gennemføres ved ibrugværende flishuggere, men metoderne kan også anvendes ved udvikling af nye flishuggere.

Projektet er afsluttet med en detaljeret projektrapport og en praktisk rettet pjece. Pjecen udgives af Arbejdsmiljøfondet.

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STØJDÆMPNING AF 167 kW GUMMIHJULSLÆSSER

Morten Skands
AS IKAS, København Danmark

Indledning

Opgaven bestod i at støjdæmpe en fabriksny Fiat Allis gummihjulslæsser type FR 20, således at støjniveauet målt i en afstand af 7 m fra maskinen maksimalt måtte være 76 dB(A).

Udgangspunkt

Støjniveauet fra den fabriksleverede gummihjulslæsser var 86 dB(A) incl. et påmonteret støjdæmpningskit. Kittet bestod i et baffel-lydslusearrangement monteret efter køleren samt lydabsorberende plader med et samlet areal på ca. 1.5 m^2 installeret i motorrummet.

Målinger foretaget med støjdæmpningskittet afmonteret viste, at støjniveauet steg ca. 3 dB(A).

Støjkilder

Til detektering af de enkelte støjkilder samt opstilling af disse i en rangorden blev anvendt følgende målemetoder:

A: Intensitetsmåling.
B: Afskærmning af støjkilder og lydtryksmåling i 7 m's afstand.

Ad A. Intensitetsmålingerne blev udført af Lydteknisk Institut.

Der blev foretaget kortlægningsmålinger på højre og venstre side, underside, bageste ende og top over motor. Endvidere blev der målt på støjudstrålingen fra udstødning og luftindsugning til motor.
Lydeffektniveauerne blev grafisk fremstillet i et gråtoneplot. Ved at inddele gråtoneplottet i passende områder blev fordelingen af lydeffekten procentvis repræsenteret i et pinde-diagram.

Ad B. De praktiske forsøg med at "isolere"/afskærme de enkelte støjkilder blev udført på følgende måde: Lydtrykket blev målt i den kontrollerende måleposition (7 m fra maskinen), samtidig med at områder af gummihjulslæsseren blev afskærmet (støjdæmpet) ved stålplader, absorptionsmateriale m.m.

Støjdæmpende tiltag

Ved en vurdering af de føromtalte målemetoder samt en analyse af måleresultaterne kunne de enkelte støjkilder rangordnes med nødvendige dæmpningsværdier, som er vist i tabellen nedenfor.

STØJKILDER	NØDVENDIG DÆMPNING
1. Køleluftindtag på sider af motorinddækning samt diverse utætheder i topplade	5 dB(A)
2. Støjudstråling fra "knæk"	4 dB(A)
3. Køleluft afkast, bagende	3 dB(A)
4. Støjudstråling fra bund	3 dB(A)
5. Udstødningssystem	3 dB(A)

Tabel 1. Rangorden af støjkilder og nødvendig dæmpning.

For at opnå de ønskede dæmpningsværdier blev følgende tiltag udført.

Ad 1. Den eksisterende motorindkapsling blev udskiftet med en nyudviklet type, der har integrerede oplukkelige lydsluser. Indersiden af motorinddækningen blev overalt forsynet med en 50 mm tyk lydabsorbent.

Ad 2. Utæthederne i "knækket" blev lukket med nøjagtigt tilpassede stålplader omkring hydraulikcylindre samt kardanled.

Ad 3. Luftafkastet fra køleren blev forsynet med en lydabsorberende tragt, der ledte luftstrømmen opad.

Ad 4. Diverse utætheder i bunden blev lukket med stålplader, der på indersiden var pålimet en speciel olieafvisende lydabsorbent af mineraluld.

Ad 5. Udstødningssystemet blev monteret med en 0,9 m lang absorptionslyddæmper efter den fabriksleverede lyddæmper.

Afsluttende bemærkninger

De afsluttende kontrolmålinger viste, at gummihjulslæsseren støjede 78 dB(A) i 7 meters afstand. De sidste 2 dB, der manglede, stammede fra et såkaldt "ørkenfilter", der var placeret oven på motorinddækningen.

Ved at anbringe filteret inde i motorindkapslingen blev slutresultatet 76 dB(A) i 7 m's afstand.

De anvendte støjdæmpningsteknikker er alle kendte. Vanskelighederne ligger i at udvælge de relevante og især i at tilpasse og modificere disse, så de kan indgå som en integreret del af en gummihjulslæsser, der skal kunne arbejde til daglig - og ikke kun ved støjmålinger.

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OFFSHORE NOISE CONTROL

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INTRODUCTION

The project NOISE REDUCTION ON OFFSHORE FACILITIES co-ordinated at the Acoustics Research Center is sponsored by Norsk Hydro, Saga Petroleum and Statoil. The project's main purpose is to reduce noise on offshore platforms both existing and planned, to achieve this end guidelines are being written to cover most of the equipment that is used offshore. The guidelines are being written in co-operation with Veritec, Lund & Aass and Miljøplan.

The guidelines consist of:

(i) OFFSHORE NOISE CONTROL GUIDE;
which is a design guide.

(ii) OFFSHORE EQUIPMENT NOISE AND NOISE CONTROL DATA BASE;
which is a data base containing high quality noise data and noise control measures.

(iii) OFFSHORE MATERIALS ACOUSTIC DATA BASE;
which is a data base containing the acoustic properties of materials that can be used offshore.

The necessity of the guidelines has been brought about by the growing number of noise regulations for offshore platforms. These regulations have been introduced to combat the problems associated with noise offshore:

- Hearing damage due to over exposure to noise.
- Masking of warning signals.
- Safety aspects associated with lack of speech intelligability

(communication on the drill floor for example).
- Annoyance/sleep disturbance in the accommodation.

Surveys on offshore installations have shown that the limits for noise are exceeded on a large number of platforms. On a representative sample of 30 drilling rigs, the general work area limit of 83 dB(A) L_{eq} 12 hours, was exceeded in 70% of the measurement positions.

DESCRIPTION OF THE GUIDELINES

The methods and data contained in the three parts of the guidelines can be used at the design stage and for noise control when retrofitting or replacing equipment.

The Offshore Noise Control Guide is intended for use during the early stages of design, when any changes required for noise control purposes are most likely to be accepted. The problems associated with noise control at the design stages are well described by S. Johansen [1]. The purpose of the Guide is to awaken the interest of the discipline engineer, give guidance on equipment selection and serve as a reference for the design stages before detailed design. The noise data contained in the Guide are consistant with the quality required for conceptual designs.

The Offshore Equipment Noise and Noise Control Data Base contains good quality sound power level (SWL) data for equipment. The data is mainly being collected using in situ surveys while the equipment is running under its normal conditions. The surveys are carried out using intensity measurements. The measurement method is described below. The Equipment Data Base consists of data sheets, figure 1, containing the data for unmodified and noise modified equipment. The Equipment Data Base will be up dated as new equipment is produced.

The Offshore Materials Acoustic Data Base will contain data for materials that can be used offshore. This will consist, for example, of absorption coefficients and sound reduction indices. The data will be both for general materials (steel plates, glass panes, etc.) as well as for specific items (windows, wall elements, etc.).

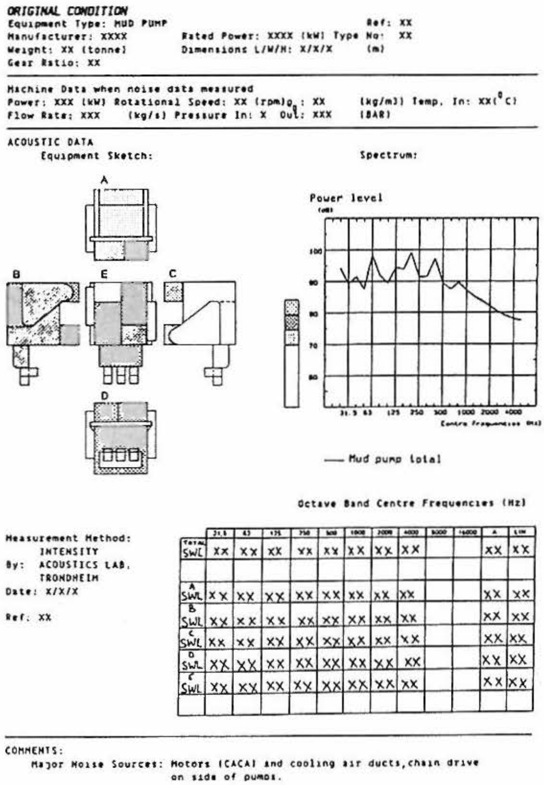
INTENSITY MEASURING METHOD

The intensity measuring method, which has mainly been used to collect data for the Equipment Data Base, was developed in the early phases of the project. The standardised method, as described in ISOs 3740-46, for use when determining the radiated power level from equipment were found to be very difficult to apply to offshore measurements, due to background noise. The intensity measuring method has been found to give good results with background noise up to 15 dB higher than the source to be measured [2].

The method used can be characterised by the following stages:

(i) Evaluate the noise characteristics of the source and the extraneous noise (time dependence, etc.).
(ii) Choose the measuring surface.
(iii) Divide the surface into logical and practical sub-areas.
(iv) Scan each sub-area to obtain the sound power radiated through that sub-area.
(v) Sum the sound power radiated through each sub-area, to obtain the total radiated sound power radiated by the source.

The method was developed for offshore application but can be used in many other situations, though its advantages are only shown when background noise is intrusive. The method is more fully described in [3].



ORIGINAL CONDITION
Equipment Type: MUD PUMP Ref: XX
Manufacturer: XXXX Rated Power: XXXX (kW) Type No: XX
Weight: XX (tonne) Dimensions L/W/H: X/X/X (m)
Gear Ratio: XX

Machine Data when noise data measured
Power: XXX (kW) Rotational Speed: XX (rpm) ρ_0: XX (kg/m3) Temp. In: XX(°C)
Flow Rate: XXX (kg/s) Pressure In: X Out: XXX (BAR)

ACOUSTIC DATA
Equipment Sketch: Spectrum:

Octave Band Centre Frequencies (Hz)

Measurement Method: INTENSITY
By: ACOUSTICS LAB. TRONDHEIM
Date: X/X/X
Ref: XX

	31.5	63	125	250	500	1000	2000	4000	8000	16000	A	LIN
TOTAL SWL	XX	XX	XX	XX	XX	XX	XX	XX			XX	XX
A SWL	XX	XX	XX	XX	XX	XX	XX	XX			XX	XX
B SWL	XX	XX	XX	XX	XX	XX	XX	XX			XX	XX
C SWL	XX	XX	XX	XX	XX	XX	XX	XX			XX	XX
D SWL	XX	XX	XX	XX	XX	XX	XX	XX			XX	XX
E SWL	XX	XX	XX	XX	XX	XX	XX	XX			XX	XX

COMMENTS:
Major Noise Sources: Motors (CACA) and cooling air ducts, chain drive on side of pumps.

Figure 1. Example data sheet for Offshore Equipment Noise And Noise Control Data Base.

STATUS OF THE PROJECT

OFFSHORE NOISE CONTROL GUIDE:

The final editing of the Guide is being carried out at present. As new methods and processes are put into use, the Guide will be revised.

OFFSHORE EQUIPMENT NOISE AND NOISE CONTROL DATA BASE:

Noise data for a range of equipment has been collected. To date the data is for types of equipment where no data existed previously, such as: shale shakes, mud pumps, cementing units, etc. For other equipment where good noise data already exists, efforts are being made to release the data to the project.

The possibility of using a computerised data base to store the noise data is being investigated. It may be possible in the future to use an expert system to choose the quietest equipment for a given duty. At present the data base is stored on a manual system, figure 1, shows an example.

OFFSHORE MATERIALS ACOUSTIC DATA BASE

The blank data sheets for the data base have been produced. The data should be more available than for equipment. In the next phase of the project requests for data will be sent out. Similar to the Equipment Data Base, the materials data will probably be placed in a computerised data base.

CONCLUSION

The guidelines produced by the project provide a useful and flexible reference for controling noise on offshore installations. The gathering of noise emission data has only been made possible by the developments that have recently occured in acoustic intensity measurements.

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NOISE FROM PROCESS AND UTILITY PIPING, OFFSHORE OIL INSTALLATIONS

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INTRODUTION

Noise control on offshore oil and gas installations has grown to an accepted feature during the last ten years.

Several "Guidelines" and Specifications with address both to mechanical and noise control engineers have been prepared. Examples are the Norwegian (SINTEF) "Offshore noise control guide" and The English (UEG) "Noise and vibration control; Guidance for designers of offshore installations". However, all these "Guidelines" and Specifications are very general and direct recommendations to design solutions for common installations are not given.

Advice on where and when to use acoustic insulation of process and utility piping are generally lacking.

Based on our experience from engineering of new installations, successive review of the result and improvements on already operating platforms, we have made a recommendation for accoustic insulation of process and utility piping.

SYSTEMS AND NOISE SOURCES

The systems of interest are the process piping, covering oil/condensate and gas compression lines and the utility piping, covering cooling medium and water injection lines.

The main noise sources in the systems referred above are pumps, compressors and valves. Except for safety relief lines (to flare header) the noise generated by the flow in pipes (included bend, branches etc.) can be neglected.

The noise data for pumps and compressors are normally given as sound pressure levels at one metre distance from pump and compressor casing/body. No information is given on (gas-/fluid-/structure-borne) noise transmission to pipes. Although, in many occations, the sound power emission from pipes can exceed the power levels for pumps and compressors with a significant amount. Consequently, basis for decisions on acoustic pipe insulation is missing.

For valves, the noise data is given as sound pressure levels at one metre distance from a standard pipe, one metre downstream of valve. Noise emission from valve body is not given as this is of minor importance. Hence, the valve noise data gives an adequate base for decisions on acoustic pipe insulation.

NOISE REDUCTION BY DISTANCE

The noise energy generated in pumps, compressors and valves transmit to the surroundings by emission direct from casings, through supports and through pipes, both as gas-/fluid- and structure-borne sound.

On offshore istallations (or other light structures), the sound radiation from structures, transmitted from pipes and equipment through supports, may be significant.

The interaction between gas-/fluid- and pipe wall and successive impedance changes along a pipe system, reduces the noise level as the distance from the source increase.

For fluid lines (cooling medium/injection water) the noise reduction along the lines is quite low. Consequently, the requirement to acoustic sound insulation for pipes may cover the total system. Further, all supports should be of resilient type to reduce "secondary" noise emission from structures.

For gas lines the noise reduction may be quite large even for a short distance, dependent on the amount of flanges, branches, valve bodies (open valves) etc.

MEASUREMENTS

Measurements have been performed on production platforms both with air- and water process coolers.

Both compressor/pump noise radiation and noise radiated from pipes and connected structures (supports) have been measured. Further, both gas systems with and without recycling/antisurge operation haved been covered.

A typical production platform with gas to air-coolers can be characterized by long pipe runs and airfan coolers on top of the platform. A typical production platform with gas to fluid-coolers require less process piping, but the cooling medium pipes come in addition.

The investigations have shown that the gas pipes in a gas compression system with no recycling/antisurge in operation are rather quiet. Only close to compressors, acoustic insulation will be required.

With antisurge operating, the extent of acoustic pipe insulation depend on the antisurge valve selection. Standard valve selection may lead to conditions difficult to solve just by resilient supports and pipe insulation (Flanges, valve bodies etc. can not be tight insulated.)

With gas to fluid cooling systems, the noise from the cooling medium control valves may give a contribution to area noise levels exceeding the noise originated in compressors and control/recycling valves.

Figure 1 shows the noise levels for a common compression system with gas to fluid cooling system and with/without antisurge operating (standard valve selection).

Figure 2 shows the required extent of pipe insulation, supposing that all valves are below 95 dBA. Acoustic insulation class A (according to OCMA NWG5) which consist of 50 mm resilient material and a 0.5-1 mm metal jacketing, will be sufficient.

All supports should be of resilient type if not fixed to main structure members and part of support welded to pipe should be insulated together with pipe.

RESULTS AND RECOMMENDATIONS

As a summary of the results of measurements and evaluations, the following design objectives for gas service lines can be stated:

a) Sound radiation from flanges is approximately 10 dBA lower than for pipes.
b) Sound radiation from valve body is approximately 10 dBA lower than for pipes.
c) Sound radiation from pump and compressor body is approximately 5 dBA lower than for pipes.
d) Sound radiation from support beams welded to pipes is approximately 10 dBA <u>higher</u> than for pipes.
e) Structure borne sound reduction by resilient supports to light structural members is 5-10 dBA.
f) Noise reduction by straight pipe runs is 0.1-0.2 dB/metre.
g) Noise reduction by standard flanges is 1-2 dBA.
h) Noise reduction by open valves and main branches is 2-3 dBA.

The points I to IV below state the required noise reduction measures for different pipe systems.

I. Insulation of gas compression systems. See figure 2. Recycling valves to meet 95 dBA. Resilient mounts.

II. Insulation of relief system. Valve noise levels above 110 dBA not accepted unless pipe insulation class A.

III. Insulation of cooling medium system. Insulation class A. Control valves to meet 95 dBA. Resilient mounts.

IV. Insulation of water injection system. Insulation class A from feed pumps to injection manifold. Resilient mounts.

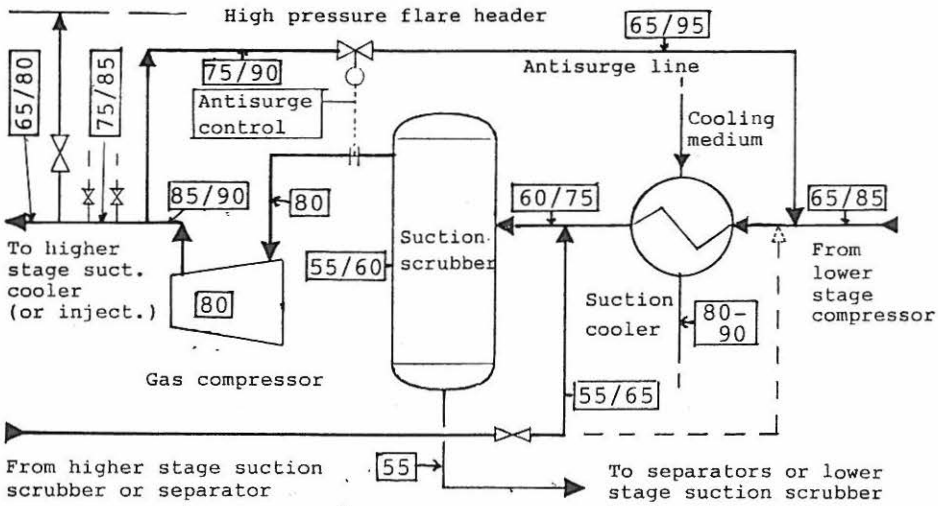

FIGURE 1. Noise levels, 1m. No antisurge/antisurge op.

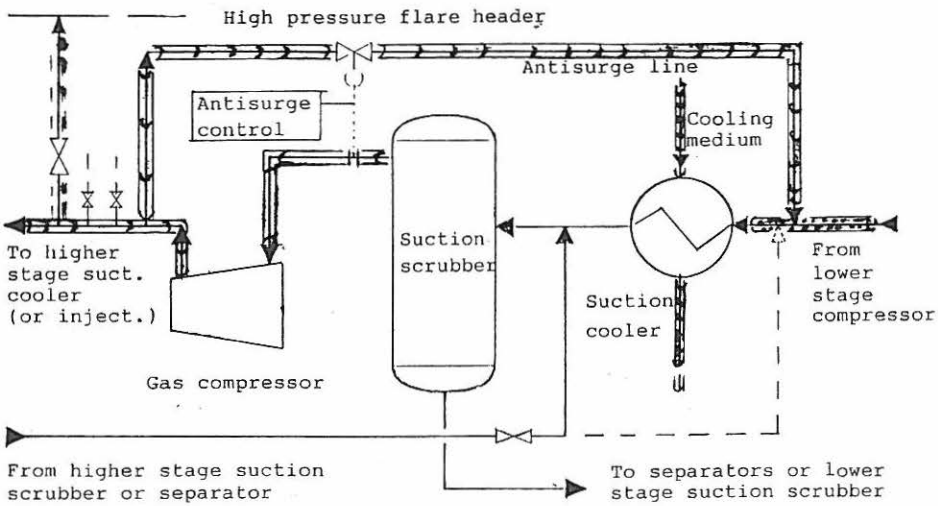

FIGURE 2. Extent of acoustic insulation. (50mm)

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PROPOSAL FOR A PROGRAMME OF ACTION TO REDUCE NOISE FROM ROAD TRAFFIC.

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In spite of the fact that 225 mill. NOK (1984 kr) has been used to reduce road traffic noise by facade insulation and building barriers, noise from road traffic is one of the environmental problems affecting the largest number of people in Norway today. We must expect it to take many years before it will be possible, from the purely practical point of view, to reduce the problem. For example, it will take up to 15 years to replace today's vehicle park. Therefore, if a satisfactory situation is to be achieved by the year 2000, we must act now.

In the light of this fact the State Pollution Control Authority (SPCA) began in 1983 to prepare a Programme of Action to reduce noise from road traffic. The perspective was to be the year 2000.

Nielsen and Solberg have given a review of the prognosis of the situation in Norway 1970-2000 (1) and the model study (2) which was the basis of the proposal of the Programme of Action.

The social cost of road traffic noise.

In spite of the lack of knowledge about the effects of noise, we have tried to indicate the extent of the problem of noise from road traffic in kroner. It must be emphasized that calculations of this involve a large degree of uncertainty, But, with this reservation, we estimate that the damage caused by noise from road traffic costs at least 400 million kroner per year.

The negative value people place on the disamenity caused by noise can be calculated on the basis of the reduction in the value of property as a result of the noise. The calculations are based on investigations which are methodologically difficult, and economists disagree somewhat as to whether the methods used are sufficiently accurate. It is emphasized that the estimate does not include the reduction in value of business property or public buildings.

It seems reasonable to suppose that disturbance to sleep will affect the capacity to work and the ability to concentrate, and therefore productivity in the working life. No certain data are available to show to what degree this is a correct assumption. As an illustration of the possible extent of the problem we have based our calculations on a reduced productivity of 1-5 % for the part of the actively employed population whose sleep at home is disturbed as a result of noise from road traffic.

Valuation of the disamenity caused by the noise

- Reduced value of property
  indicates that the disamenity caused by the noise is valued at

  300 million kr/yr.

- Loss of productivity
  due to reduced work performance as a result of disturbed sleep may amount to

  100 – 500 million kr/yr.

- In addition there are other effects which we have not attempted to quantify
  - Effects on health
  - Reduction in the value of business premises, public buildings, and schools
  - Effects on well-being in the street environment

<u>The necessary investments to reach different goals.</u>

What we have done is to concentrate on two main goals, and for each of these goals, work out different methods of achieving them.

The first goal is that indoor noise in dwellings should be reduced so that no-one is exposed to noise levels exceeding 35 dBA. This can be achieved by insulation of facades alone, or by as far as possible to reduce the outdoor noise levels to 65 dBA by decreasing the noise emission from vehicles, combined with some insulation of facades. 30 % of the population state that they are "very annoyed" by noise levels higher than this. Therefore a goal of 35 dBA cannot be said to be very ambitious.

The second and more ambitious goal is to reduce indoor noise to 30 dBA, alternatively outdoor noise to 60 dBA. This can also be achieved by insulation of facades alone, or by a combination of measures.

The following table shows the annual investments and increased costs for vehicles if these goals are to be achived. Two alternative combinations of measures are shown for each of the two goals.

Estimate of the total need for investments in insulation of facades/noise barriers and on measures to vehicles for the different goals during the period 1986 – 2000 (mill. 1984 kr)

Goal for yr 2000	Measure	Annual investments for 15 yr		
		Insulation of facades/ barriers	Requirements reg. vehicles	Sum
35 dBA indoors	Insulation of facades (A35)	76		76
65 dBA outdoors	Requirements reg. vehicles Insulation of facades Barriers (B35)	19	188	207
30 dBA indoors	Insulation of facades/ barriers (A30)	354		354
60 dBA outdoors	Requirements reg. vehicles/ barriers/facades (B30)	142	188	330

Cost-effect data for different goals.

When choosing between the four different alternatives we must first ask ourselves which will give the greatest improvement to the present situation per kroner spent.

The following table shows how many persons are no longer "very annoyed" and whose sleep are no longer "frequently disturbed" in the year 2000 per million kroner invested.

Reduction in the number of persons disturbed by noise in the year 2000 per million kroner used annually on different combinations of measures against noise from road traffic

Combinations of measures	No. of residents	
	Very annoyed	Sleep frequently disturbed
A 35	225	114
B 35	443	84
A 30	273	95
B 30	293	81

Alt. A: Insulation of facades
Alt. B: More stringent requirements regarding noise from vehicles + insulation of facades

For the sake of simplicity, in the continuing discussion we have chosen to ignore the solution giving an indoor noise level of 30 dBA by insulation of facades alone. The reason, somewhat simplified, is that we are able to obtain much more per invested kroner by achieving this result through more stringent requirements regarding noise from vehicles, and because the alternative is of little relevance for other reasons as well.

Cost-benefit calculations.

When defining our goal, we cannot only look at what we get back for each invested kroner. We must also consider the costs and benefits of the relevant alternatives, as shown in the following table.

Calculated cost and benefit to the society of using the most relevant combinations of noise-abatement measures (mill. 1985 kr per yr)

Combination of measures	Cost to the Society	Benefit to the Society
A 35	45	53 – 97 (42)
B 35	200	174 – 294(144)
B 30	275	238 – 380(202)

Low benefit alt: price of housing + 1 % loss of productivity

High benefit alt: price of housing + 5 % loss of productivity

The figures in brackets are calculated by the housing price method.

This table needs some explanation. In the first place the figures given for costs are not identical with the figures for investments used earlier. One of the reasons is that in the cost of insulating facades is included the maintenance cost and the benefit gained from energy saving. The benefit from energy saving means that the actual cost of insulating facades is lower than indicated by the investment figure considered in isolation.

The figures representing benefit are the sum of how people value the improved situation calculated on the basis of increased value of dwellings, and increase in productivity. The figures do not include less noise in business premises and offices etc., reduced health damage and increased well-being in the street environment. As stated already, the figures involve a large degree of uncertainty.

Conclusions

The conclusion drawn by SPCA is that the goal 65/35 dBA should be regarded as a minimum of what should be achieved by the year 2000. Probably the right goal to aim for would be somewhere in between the minimum goal and the more ambitious goal defined above.

As regards the best measures to reach the goal, above all the noise emission of the vehicles should be reduced, to 75 dBA (measured in the second gear) for private cars, 80 dBA for heavy vehicles. SPCA recommends the introduction of noise related charges in Norway, as the best instrument to introduce such comparatively low noise vehicles.

Besides this, it is recommended to spend at least 300 million kroner on the insulation of buildings, to expand the use of traffic management measures in the largest towns and to introduce stricter planning practises to avoid the building of new noise-exposed housing.

Finally it is recommended to invest 15 million NOK on research and development during the next 5 years. One important topic will be the development of sound absorbing road surfaces.

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Erfaringer med anvendelse af den fællesnordiske metode til beregning af støj fra jernbaner og forslag til forbedring.

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Den fællesnordiske metode til beregning af støj fra jernbaner blev offentliggjort i 1985.

Den har nu været anvendt et par år. I det følgende nævnes et par erfaringer med metoden og på baggrund heraf gives forslag til forbedringer af visse led i metoden.

Fjerntrafik ved Knudshoved/Nyborg

Ved overvejelserne om sammenlægning af bil- og jernbanefærgerne over Storebælt blev de nuværende og fremtidige støjforhold beregnet. Som kontrol af beregningerne gennemførtes målinger. Målingerne blev dels foretaget ved et rangerterræn, dels ved en strækning med persontogstrafik med høj hastighed. Målingerne ved trafik med høj hastighed viste god overensstemmelse med metodens beregningsresultater. Modellen kunne ikke anvendes ved rangerterræn.

Godstogstrafik i København

Som led i en forbedring af godstrafikken på jernbane mellem Sverige, Danmark og Tyskland indgik SJ, DSB og DB aftale om etablering af DanLink. DanLink baserer sig på forbedrede færgeruter, med øget kapacitet og reduceret rejsetid. Som del af DanLink skal DSB etablere et nyt færgeleje i Københavns Frihavn. Lokalplanproceduren i forbindelse med disse anlægsarbejder gav anledning til ganske voldsom debat om først og fremmest støjforholdene. Fra Frihavnen må jernbanetrafikken passere gennem tætbebyggede boligområder. Afstand fra spor til nærmeste bebyggelse er enkelte steder nede på 7-10 m. Samtidig er der stærke stigninger (op til 10 o/oo) og relativt skrappe kurver. Derfor er støjniveauet flere steder højt, selvom jernbanetrafikken ikke er særlig voldsom.

Støjreducerende foranstaltninger er derfor blevet medtaget i projektet. DSB pegede bl.a. på mulighederne for at reducere støjen ved kilden ved at indsætte de mest støjsvage lokomotiver og reducere max. hastigheden fra 60-70 km/t til 40 km/t.

Vi fik til opgave at vurdere og belyse den støjmæssige effekt af disse foranstaltninger samt generelt belyse situationen i dag og efter etableringen af DanLink ved forskellige driftsoplæg.

Til dette formål opbyggedes et edb-program baseret på den fællesnordiske beregningsmetode. Vi løb dog hurtigt ind i forskellige problemer. Metoden er således primært baseret på målinger på hurtigtkørende el-drevne persontog. Vi skulle belyse langsomtkørende lange godstog med forskellige typer diesellokomotiver. Og vi skulle belyse støjforholdene ved selektivt lokomotivvalg og forskellige, lave hastigheder. Vi delte problemet op i følgende delproblemer:

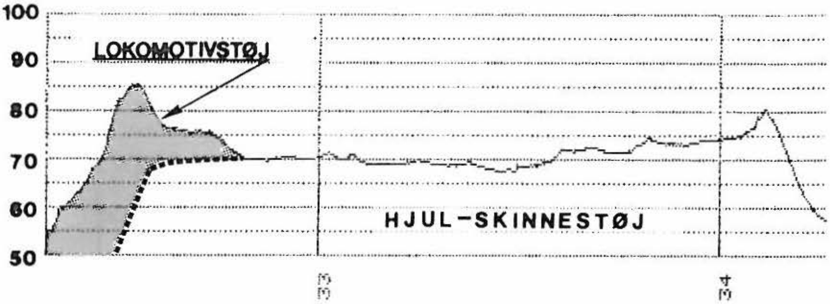
Støjbidrag fra lokomotivet

I beregningsmetoden gives en korrektion for lokomotivtype uanset togantal/døgn og det enkelte togs længde. Dette er naturligvis ikke korrekt. Vi delte derfor støjen fra et langt godstog op i

- o hjul-skinnestøj
- o lokomotivets støjbidrag.

Hjul-skinnestøjen lader sig direkte bestemme i beregningsmetoden som funktion af toglængde og hastighed.

Vi valgte at beregne lokomotivets støjbidrag udfra støjniveau ved given effektafgivelse (motoromdrejning på dielselmotoren) normalt benævnt ved en kontrollerstilling mellem 0 og 8, og lokomotivets hastighed, hvorved varigheden af en lokomotivpassage bestemtes. På figur 1 er dette princip vist.



Figur 1. Støjniveau ved togpassage, opdelt på lokomotivstøj og hjul-skinnestøj.

Støj fra forskellige lokomotivtyper/effektafgivelse

Beregningsmetoden indeholder ikke korrektionsværdier for de forskellige typer af diesel-elektriske lokomotiver, som DSB anvender. Vi måtte derfor for de enkelte lokomotivtyper indhente måledata for lokomotivstøjen som funktion af kontrollerstilling. I nedenstående tabel er disse værdier angivet

KST	MX	MY	MZIV	MH*
8	92	96	89	90
7	90	93	87	87
6	88	89	84	83
5	87	88	82	82
4	83	81	81	75
3	83	79	78	73
2	81	75	77	69
1	79	75	77	69

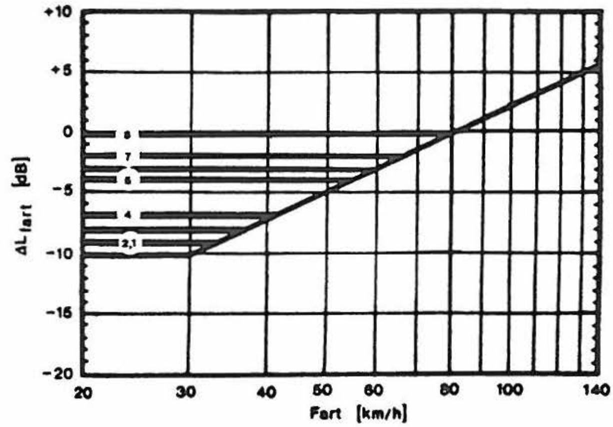
Tabel 1. A-vægtet lydtrykniveau i 10 m's afstand.

* MH-loko har trinløs regulering.

Med disse informationer er det derefter muligt at beregne de togtypekorrektioner, modellen skal operere med, og de korrektionsled, der skal anvendes ved forskellige hastigheder og kontrollerstillinger.

Nedenfor er en enkelt af disse korrektionsled vist.

De ydre rammer (ingen henholdsvis fuld acceleration) for korrektionsleddet er i fuld overensstemmelse med beregningsmetoden, som den foreligger.



Figur 2. Hastighedskorrektion for et 200 m langt persontog ved forskellige kontrollerstillinger.

Vort forslag til modelkorrektion

Vi vil foreslå, at metoden opdeler støjen i et støjbidrag fra lokomotivet og et bidrag fra hjul-skinnestøj. Dette vil gøre det muligt at belyse støjen mere korrekt i nærheden af stationsområder, hvor støjproblemerne i øvrigt ofte er størst.

Søren Rasmussen

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VEGTRAFIKKSTØY - LAVE KJØREHASTIGHETER

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INNLEDNING

Nordisk beregningsmetode for vegtrafikkstøy (NBV) er basert på utgangsverdier v/10 m for ulike trafikkmengder og hastigheter. For hastighetsområdet over 50 km/h finnes gode bakgrunnsdata for metoden, men for lave kjørehastigheter - under 50 km/h - har en måttet foreta antakelser. Dette arbeidet har hatt som hovedsiktemål å supplere NBV med utgangsverdier for hastighetsområdet 20 - 50 km/h.

METODE

På 7 målesteder er det tilsammen innsamlet data for rundt 3500 enkeltpasseringer av kjøretøy. Disse er fordelt på 15 ulike trafikksituasjoner, avhengig av kjøreretning og kjøretøytype. Trafikkforholdene viser stor spredning over de ulike målestedene, bl.a. inngår områder med fartshindrende humper, flat veg, fra/til kryss og stigning. Det er målt med fri flyt og "glissen" trafikk. Hastigheten er målt sammen med forbipasseringsnivået (Fast), og dette har gitt grunnlag for omregning til ekvivalentnivå. Omregningsmetodikken er kontrollert ved direkte ekvivalentnivåmålinger, og typisk nøyaktighet er innenfor 1 dB. Alle data er omregnet til 10 m avstand, fritt felt. Det er vurdert 10 trafikksituasjoner for lette kjøretøy og 5 for tunge kjøretøy. Resultatene fra disse vurderes samlet med hensyn til forholdet mellom støynivå og kjørehastignet 20-50 km/h.

RESULTATER

Maksimalnivå: Resultatene viser at midlere forbipasseringsnivå varierer mellom ca. 2-4 dBA pr. 10 km/h hastighetsøkning for lette kjøretøy. For tunge kjøretøy er forbipasseringsnivået stort sett uavhengig av hastigheten.

Det maksimale støynivået er ca. 80 dBA for lette kjøretøy og ca. 90 dBA for tunge kjøretøy - som i NBV pr. idag. Maksimalnivået synes uavhengig av hastigheten både for lette og tunge kjøretøy. I trafikksituasjoner med "forsiktig kjøring" tyder resultatene på at maksimalnivået for tunge kjøretøy kan settes ned til 85 dBA, mens en slik konklusjon ikke er entydig for lette kjøretøy.

Ekvivalentnivå: Regresjonslinjer for ekvivalentnivå vs. kjørehastighet for de ulike trafikksituasjonene viser at områder med "forsiktig kjøring" (f.eks. humper i vegbanen jevn hastighet, lite aksellerasjon) ligger 3-6 dB lavere enn områder med mere "aggressiv" kjøring, (f.eks. fra kryss, betydelig aksellerasjon). Dette gjelder både tunge og lette kjøretøy.

NBV skiller ikke mellom ulike trafikksituasjoner i utgangsverdien - utover kjørehastigheten. I figur 2 er således resultatene midlet og midlingskurver inntegnet. En får følgende samband for lette og tunge kjøretøy i hastighetsområdet 20-50 km/h:

Lette : L = 32.0 + 0.11 v (dBA) (20-50 km/h)
Tunge : L = 49.8 - 0.11 v (dBA) (20-50 km/h)
(L = ekvivalentnivå pr. kjt. pr. time, 10 m, fritt felt)
(v = kjørehastighet km/h).

Midlingskurvene gjelder for enveiskjørte veger, og for veger med lik kjøremåte i begge retninger. For vanlige veger med kjøring i begge retninger vil midlingskurvene ligge i datamengdens øvre sjikt.

Overstående formler er benyttet til supplering av diagrammet for utgangsverdien i NBV, se figur 1. Figur 1 viser følgende: - 100% lette kjt : Støynivået øker 3 dB fra 20 til 50 km/h. - 100% tunge kjt : Støynivået avtar 3 dB fra 20 til 50 km/h. - 10 % tungtrafikkandel (vanlig på riksveger): Støynivået er uavhengig av hastigheten mellom 20 og 50 km/h. - I forhold til NBV's nåværende utgangsverdier for 50 km/h ligger måledataene lavere, økende differanse med økende tungtrafikkandel. Ved 10 % tunge er differansen ca. 2 dB, et tall også tidligere studier er kommet fram til.

Denne undersøkelsen har primært fokusert på forholdet mellom støynivå og kjørehastighet. Men andre forhold spiller også avgjørende rolle for støynivået. Det er bl.a. kjent at kjøring ved lavt turtall gir en betydelig reduksjon av støyen. Dette kommer blant annet til uttrykk i figur 2, der ekvivalentnivået viser en spredning på opptil 6 dB ved lave hastigheter. Vi vet idag ikke nok til entydig å definere slike områder som gir "rolige kjøreforhold", f.eks. som en korreksjon i beregningsmetoden. Allikevel er det en viktig oppgave å avklare forhold som gir rolig kjøring, og identifisere typiske områder, bl.a. som bakgrunn for støyreduksjonstiltak i tettbygde strøk.

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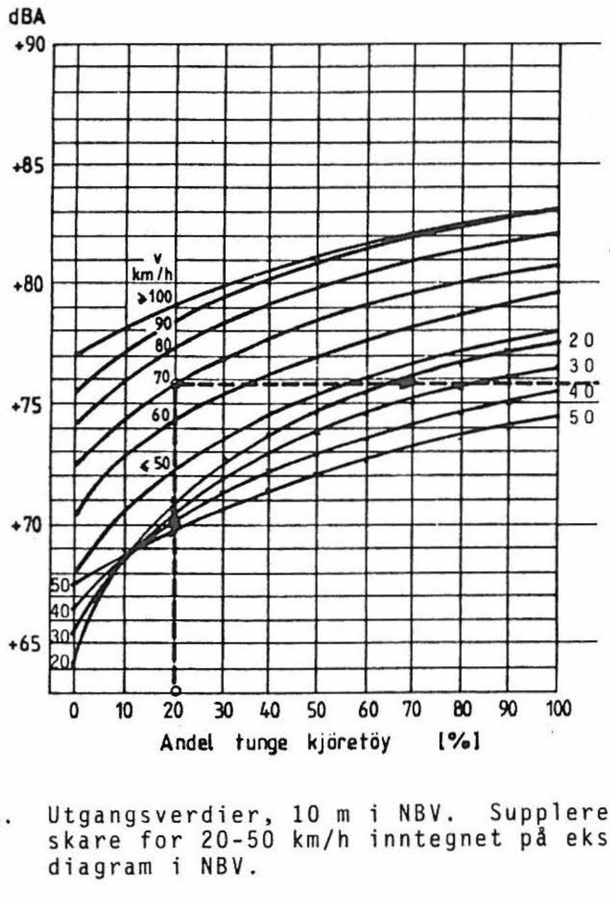
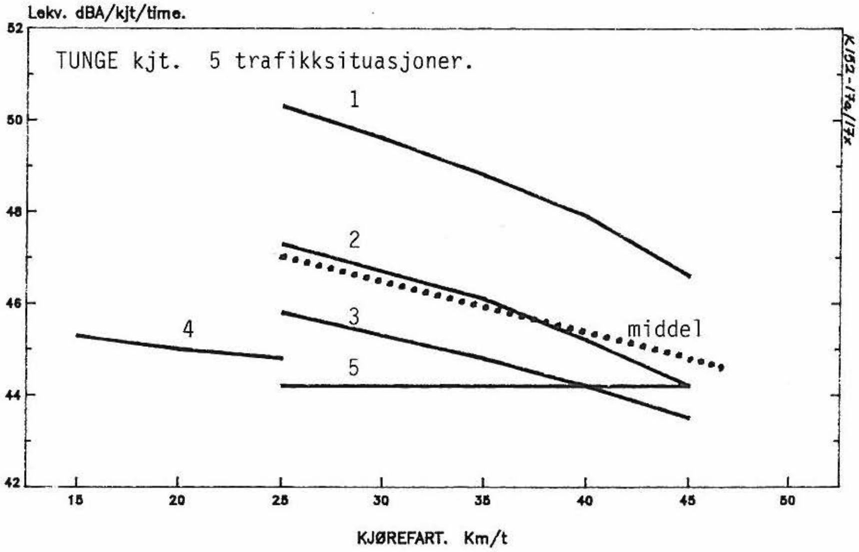
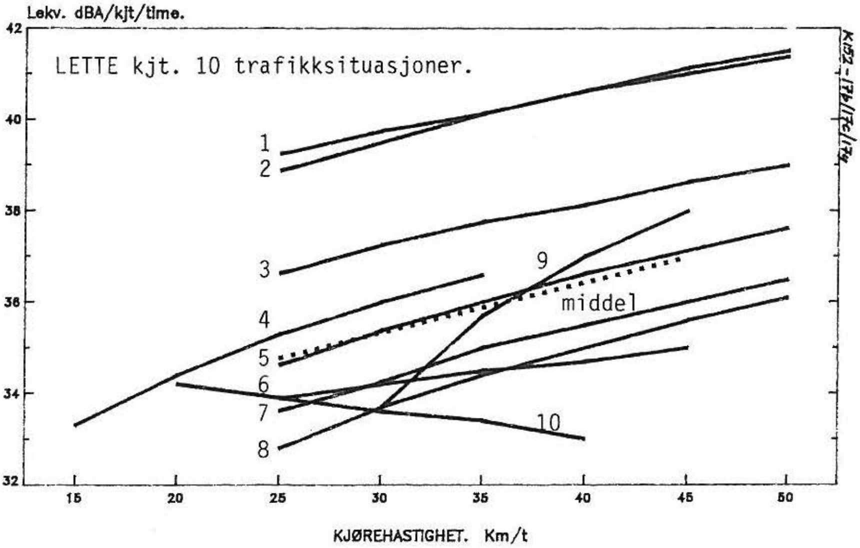


Figur 1.. Utgangsverdier, 10 m i NBV. Supplerende kurveskare for 20-50 km/h inntegnet på eksisterende diagram i NBV.



Figur 2. Ekvivalentnivå vs. hastighet.
Regresjonslinjer på energibasis.

a) Tunge kjøretøy
b) Lette kjøretøy



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PLANE-WAVE RADIATION: A MECHANISM FOR TIRE/ROAD NOISE AT LOW FREQUENCIES.

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INTRODUCTION

Further reduction of road traffic noise is still an urgent task. Tire/road noise is an increasing problem, since engine noise is constantly being reduced by vehicle manufacturers.

Both low and high frequencies contribute to the roadside tire/road noise. However, the low frequencies dominate behind screens and behind or inside buildings. Most R&D efforts on tire/road noise mechanisms have focused on high frequencies (above 800 Hz). The present study, therefore, sought an explanation of the low-frequency component of tire/road noise.

PLANE BENDING WAVES IN THE TIRE TREAD BAND

Leading and trailing edge of the tire/road contact patch each create a narrow, horn-shaped geometry. Two things occur:

1. High acoustic impedances result near the contact edges (Fig. 1).
2. Plane bending waves in the tire tread band cause high particle velocities close to the contact edges (Fig. 2).

As bending waves in the tread form a constant velocity source, the combination of high acoustic impedances and high particle velocities cause high levels of radiated acoustic power (Eqn. 1):

$$L_W = 10 \log \left(u^2 \cdot Z_r \cdot A_c \right) + 120 \quad (\mathrm{dB}) \quad \text{...........(1)}$$

where for a given radiation area between road and tire:

L_W = radiated sound power level (dB)
u = particle velocity (m/s)
Z_r = resistive part of the acoustic impedance (Ns/m^3)
A_c = horn mouth area (sample area in calculations) (m^2)

<u>The tread band as a wave-guide.</u>

Just as with air borne sound in a duct, bending waves in a tire tread can be either plane or of higher-order. At low frequencies, below the cut-off frequency for the higher-order modes, only plane waves can exist.

At higher frequencies, the randomly textured surface roughness of normal tire/road systems causes moment excitation of the tread (1). Most of the wave energy is then contained in higher-order modes. They are poor radiators of far-field sound since "hydrodynamic short-circuiting" (1) neutralizes radiation.

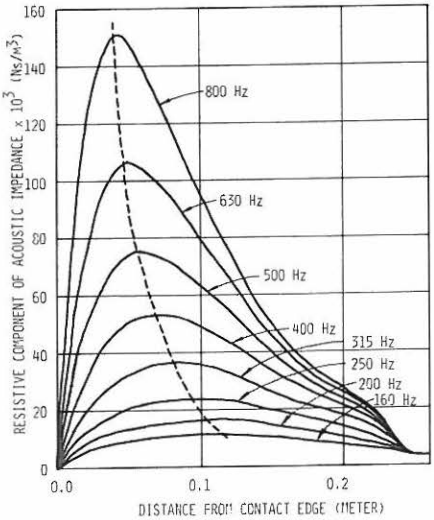

Fig. 1. Acoustic impedance at various positions near the contact patch. As the frequency rises, the peak in impedance approaches the contact edge. Bald tire and smooth surface.

At lower frequencies where $\lambda > r$, particle velocities peak outside the horn geometry, where the acoustic impedances are much lower. This means low radiation of acoustic power.

The maximum radiation due to Plane-Wave Radiation thus occurs over a frequency range of about 150-450 Hz. This lies <u>below</u> the cut-off frequency for higher-order tread-wave <u>modes</u>, and <u>above</u> the condition $\lambda < r$.
(λ=bending <u>wavelenght</u>; r=tire radius)

COMPUTER CALCULATIONS

The theory evolved into a computer program for calculating the influence of varying the surface roughness, varying tread width etc. The program enables calculation of acoustic impedances and resulting particle velocities. Another necessary program feature was an exact description of the loaded geometry near the contact patch. Fig. 5 shows (together with experimental results) the calculated influence of increasing the mean surface roughness from 0 to 16 mm .

PARAMETRIC DEPENDENCE

Plane-Wave Radiation decreases when:

* **road roughness increases;**
  Particle velocities and acoustic impedances would decrease at the contact edges

* **percentage of tire tread profile volume increases;**
  Particle velocities and acoustic impedances would decrease at the contact edges

* **stiffness across the tread band decreases;**
  Cut-off frequency would fall.

* **inflation pressure is reduced;**
  Tread-band stiffness would decrease, lowering the cut-off frequency

EXPERIMENTAL STUDIES - STATIONARY TESTS

Measurement set-up.

The measurement set-up is shown in Fig. 4. A non-profiled tire (dimension 155 SR 13) was loaded by a 10 mm steel plate attached to the rim by a special mechanism. The tire with its loading mechanism was placed on a U-shaped concrete foundation. Underneath the tire was mounted an electrodynamic shaker. It was linked to the tire contact edge via a hole in the concrete foundation. The link mechanism drove a T-shaped bar which enabled excitation of coherent plane waves on the tire tread.

Measurements were done in an anechoic room. The microphone was mounted 500 mm from the leading contact edge. This was to ensure far-field conditions with respect to radiation from the tread.

The "detachable-plate" experiment

One problem was how to achieve convincing experimental verifications of the theory, without changing tire geometry, excitation, road surface or other parameters that could influence results.

We solved the problem by a detachable plate, 16 mm thick, fitted to the leading edge of the contact patch according to Fig. 4. Removing this part of the "road surface" ensured that nothing else was changed except the mean surface roughness. In this way it was possible to vary this one parameter only, under fully controlled conditions. Excitation, exact tire geometry, etc. remained unaffected.

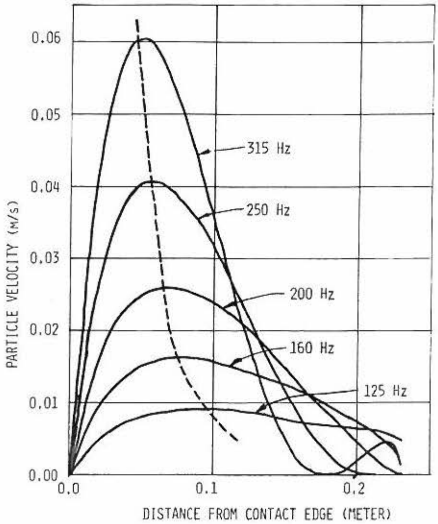

Fig. 2. Particle velocities between tire tread and road due to plane bending waves in the tire tread band. Pe: amplitude: 2 10^{-5} meter. As the frequency rises, the peak in particle velocity approaches the contact edge. Bald tire and smooth surface.

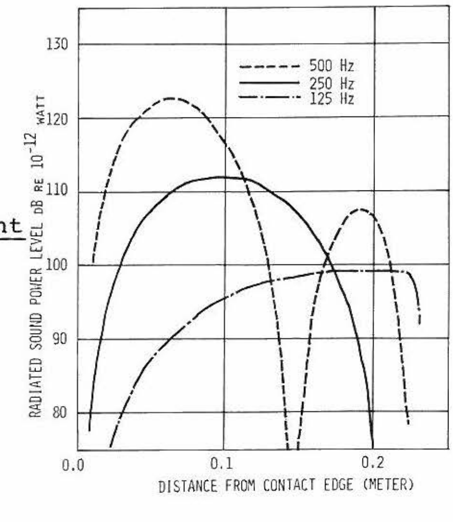

Fig. 3. Radiated acoustic power due to plane bending waves in the tire tread. Peak amplitude: 2 10^{-5} meter. Bald tire and smooth surface.

The results are shown in Fig. 5, together with the theoretical prediction. The effect of "Plane-Wave-Radiation" starts at about 100 Hz and the influence from higher-order modes starts at about 400 Hz. Agreement with theory is excellent.

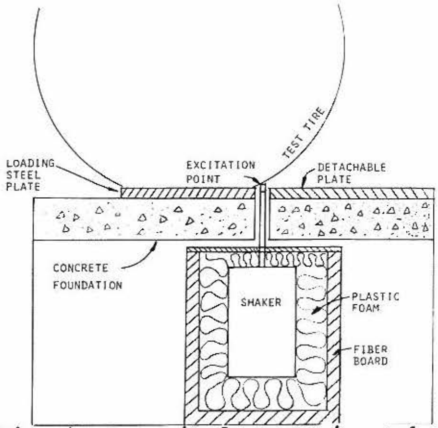

Fig. 4. Test rig for experimental verification of the Plane-Wave Radiation. The tire was loaded by a mechanism attached to the rim.

Influence from reducing inflation pressure.

Reducing the inflation pressure lowers the stiffness of the carcase and tread band. This causes reduced bending-wave velocity in the tread band, lowering the cut-off frequency for higher-order modes. Fig. 5 shows how the cut-off frequency falls from 400 to 315 Hz when the inflation pressure is reduced from 2.2 to 1.1 bars.

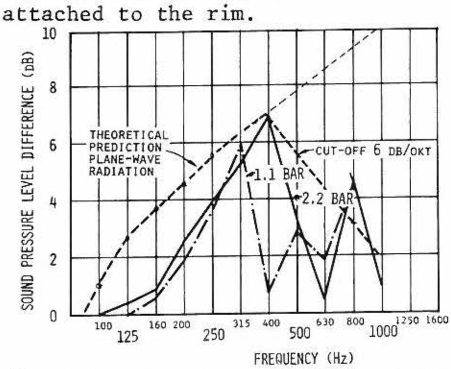

Fig. 5. Results from the "detachable-plate experiment", together with a theoretical prediction. Note the drop in cut-off frequency when inflation pressure is reduced.

ACKNOWLEDGMENTS

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I wish to express my gratitude to Professor Manfred Heckl for encouraging and helpful discussions during my visits to Berlin.

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PARAMETERS INFLUENCING NOISE EMISSION LEVELS OF PASSENGER CARS IN URBAN TRAFFIC

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INTRODUCTION

One of the most cost/efficient-methods to reduce the annoyance from road traffic noise is to reduce the noise emission levels from the vehicles themselves. This can be done by lowering the permissible noise limits for the vehicles or tightening up the test procedure.

To ensure a real effect of lower legal limits, it is necessary to have a test method which is highly correlated to driving conditions in urban traffic for all types of passenger cars. This is not the case with ISO 362 or the method developed for urban driving conditions: ISO 7188. Most European countries have adopted a test method based on the conditions outlined in ISO 362. This method has several weaknesses, especially for passenger cars. As an example, it is today possible to change design components on the car in order to achieve lower noise levels during the conformity test, which has little or no effect on emission levels in urban traffic. In the coming decade it is obvious that we will experience a lot of changes in motor vehicle technology (a highly electronically controlled vehicle may be expected), especially on motor and transmission design. To meet these changes it is necessary to develope a new type approval test method, which closely considers the most important vehicle parameters defining the noise emission levels in urban traffic.
This paper outlines the different parameters influencing the noise emission levels from a car.

PARAMETERS INFLUENCING THE NOISE LEVELS

The noise emission level from a single vehicle passing a reference point is not defined by a linear function of a single input/single output parameter, but is the result of the interaction between a whole range of parameters. However, there are four <u>primary</u> parameters which alone define the noise level from a car in a chosen situation:

- engine speed:
  Engine noise is proportional to engine speed to a factor of $N^3 - N^5$ (gasoline driven). Thus a reduction of engine speed reduces the noise level in the range 9-15 dB. Another way to describe this relationship is that <u>one</u> car at an engine speed of 4000 rpm can be equally noise as <u>32</u> cars at 2000 rpm. Kemper [1] presents the following relationship between noise level and engine speed:

  $$L_a - L_b = C \log \frac{N_a}{N_b}$$

  where C is dependent on enngine type, 30 < C < 50, a and b are index for engine speed.

- engine load:
  For gasoline driven cars, the engine noise is strongly dependent on load. The difference in noise level at, say 2500 rpm; can be as much as 10 dB. For diesel engines the difference is much less (2-3 dB).

- vehicle speed:
  This is a primary parameter at higher vehicle speed, since the tire noise then becomes the dominant noise source. To what degree, depends on the actual speed, engine rpm, load and the combination tire/road surface. Even if tire noise can be a significant noise sources below 60 km/h, we think it is nencessary to exclude the influence of vehicle speed/ tire noise in the development of an improved test procedure.

These 3 parameters are all operational parameters. The other primary parameter are <u>source</u> related parameters such as acoustical condition of the vehicle (the degree of noise control measures or maintainance). Examples here are encapsulated engines (BMW, Mercedes-Benz) or cars with large engine capacity at low max. power speed (BMW's eta-engine).

All these parameters are again influenced by a whole range of secondary parameters, which can be divided into three groups:

- vehicle related
- traffic related
- driver related

The coupling of these parameters to the primaries or to the noise level are complex and cannot easily be quanti-

fied. Of special importance for a new test procedure is the vehicle related parameters.
From [2] we have learned that the most important of these, are:

- power/weight ratio
- power/cylinder volume
- gear system.

All these must be thoroughly considered in the design of a noise immission related procedure. Figure 1 describes a model of the different parameters influencing the noise level. A more detailed discussion of these parameters are presented in [3].

A suitable noise unit must also be considered. To measure L_{Amax} on each side of the car is simple, but has several weaknesses. It does not consider any directivity of the source, as well as it is not very well correlated with noise immission levels. The use of equivalent noise level or sound power level would be a more suitable approach.

MEASUREMENT ON PASSENGER CARS

A total of 17 passenger cars, all 84/85-models were measured at a range of different driving conditions.
At constant speed (20, 35, 50, 65 and 80 km/h) in different gears and full acceleration from different vehicle speeds and using different gear. The influence of tire noise were also measured.
The results from the measurements can be summarised as follows:

- At constant speed and at a chosen engine speed (3000 rmp), no correlation between the noise level and the most important vehicle parameters (power/weight, power/engine displ. etc.) where found.

- The noise level from accelerating vehicles is strongly dependent on the approach engine speed (before the acceleration starts).

- Tire noise can strongly influence the total noise level from vehicle speeds of 35 km/h in 3rd gear (see figure 2).

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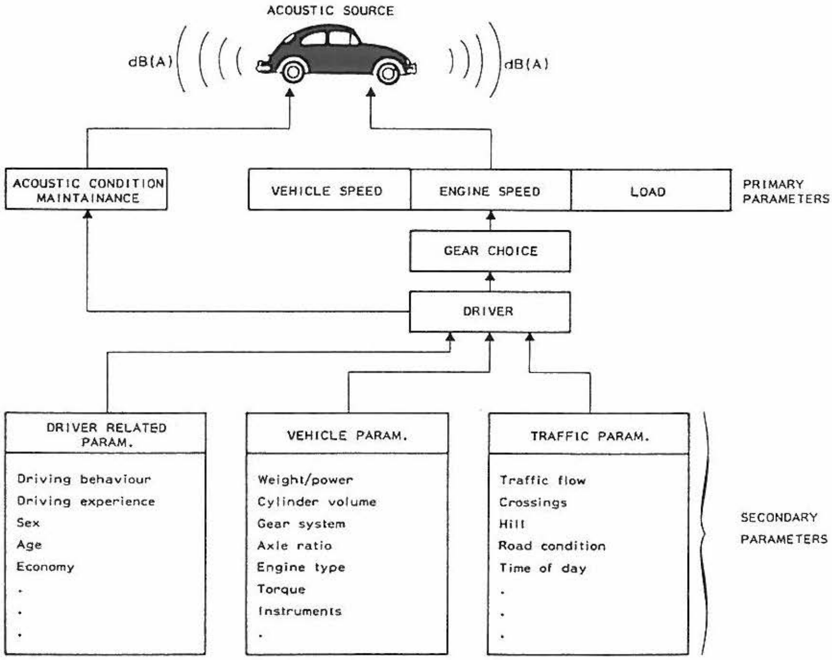

Figure 1. The influence of primary and secondary parameters on the vehicle noise emission level.

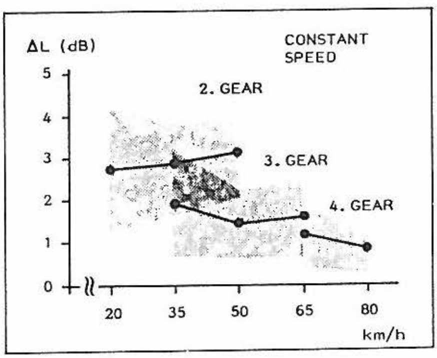

Figure 2

Difference between total noise level and tire noise. Mean values and standard deviation for 17 cars.

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FACTORS GOVERNING INDUSTRIAL NOISE CONTROL

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Introduction

Acousticians dealing with noise control of machinery, process equipment and also whole plants are inclined to design the best possible solution in terms of low noise level, short reverberation time and efficient vibration isolation. Of course, this is a correct approach in many cases, particularly when the basic problem is restricted to a few dominating causes.

Practical design

1. In practice, however, the design or modification of constructions for noise reduction in industries normally requires consideration to factors like:

- o machine tendering
- o machine service and maintenance
- o physical qualities of the type: production speed, product quality and machine failure frequency
- o cooling of electric motors and other components
- o hygiene in the production of food-stuffs and drugs etc
- o increased risk of fire hazard (use of foam plastics etc)
- o economy

When taking these aspects into account, the result is very often a lower efficiency with regard to noise reduction etc.

2. Another thing to observe when it comes to noise reduction in an industry hall is that there normally are many sources contributing to a high noise level.

This means that - in order to reduce the noise dose - not only the strongest source but also several others have to be silenced.

Here is a presentation of some cases illustrating how it is possible to work with these two groups of factors.

Case 1. Cost efficiency

A large mechanical workshop hall contains a zone of the size 400 sq.meters where 7 to 9 persons (due to the work-shift in question) are working with cutting and milling operations on cast iron details. Since the roof is saw-tooth shaped, ca 800 sq.meters of absorbers had to be installed to cover the ceiling; cost about 100.000 SEK (= ca 14.000 US$) giving a noise dose reduction of around 1 dBA.

The subjective impression of improvement is much larger than the measured, which may justify the relatively large costs. The hearing damage risk for the exposed machine operators is however almost unchanged, so it was necessary to continue with noise reduction measures on machines and equipment.

The next step was to introduce a standard muffler in the dust extraction installation. The equivalent sound level was then reduced by 4 dBA all over the zone and at a cost of ca 5.000 SEK (= ca 750 US$).

Out of these two measures, the latter is the far most cost-effective.

Case 2. Reducing a machine operator´s noise dose

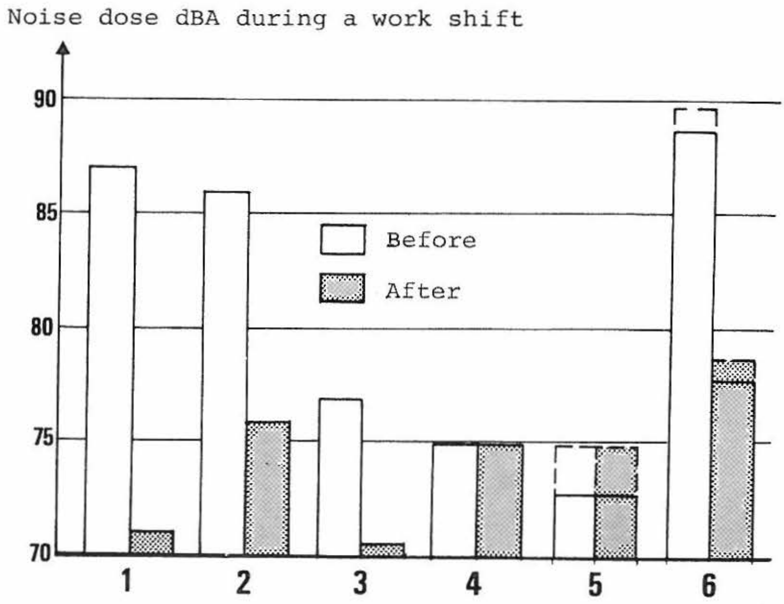
The operator´s noise dose is seldom determined by one single source (or asingle machine). Therefore it is important to find the sources and the strength of each source, using either the technique of running the sources separately or sound intensity measurements.

The equivalent sound level measured at the operator´s position in front of a table milling machine was 90 dBA during a normal work-day. Detailed measurements showed the influencing sources, which are drawn in figure 1.

It was desireable to reduce the levels to 80 dBA.

The dominating noise levels were found to be

- o fan noise from the dust extraction hoods 1
- o the machine drive unit (electric motor, toothed drive



o belt and gears) (2).

The next sources in order were:

Figure 1

- o hydraulic pump unit (3)
- o air nozzle for cleaning purposes (4)
- o milling process (5).

The total noise composed by these sources is shown by (6).

Obviously, it is most important to reduce noise from sources (1) and (2), but there is an optimum value:

- o fan noise (1) can be reduced (using a muffler in the duct) to any degree; turbulences in the air flow generate noise which will limit the possible noise reduction as can be seen from the shaded column
- o the drive unit noise (2) cannot in a reasonable way be reduced using construction modifications. A screen construction attached to the machine body with openings for cooling air was a solution giving a decrease in level from 86 dBA to 76 dBA
- o in order to reach 80 dBA the hydraulic unit had to be silenced. A reduction from 76 dBA to 71 dBA was obtained with a removable enclosure standing on legs over the pump with an opening at the floor, permitting the circulation of cooling air
- o no attempt was made to reduce air-blowing noise and milling noise.

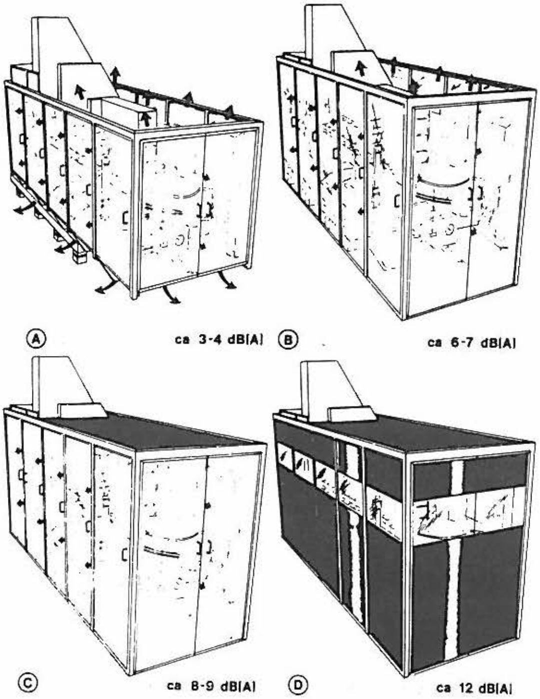
The resulting noise reduction at the operator's position was 10 - 12 dBA at a cost of 1.100 - 1.300 US$.

Case 3. Packaging machine with enclosure

Many types of production machines are equipped with protective safety screens, which also can be used for noise control if designed in a suitable manner. The packaging machine for ground coffee, condensed soup powder, cocoa powder etc in this example had originally transparent plastic windows covering the upper part of the machinery. The noise reduction was found to be 3 - 4 dBA.

The practical requirements for a screen with improved noise reduction were i.a.

- o easy access to the machinery
- o clear supervision of the function through the doors
- o easy cleaning-up of waste powder on the floor
- o accepable air circulation for cooling.



A. Short screens. Noise leakage above and below the screens

B. Long screens. Noise leakage above the screens

C. Long screens, sound absorbing ceiling

D. Long screens with sound absorbers on the inner side (partly), sound absorbing ceiling.

Figure 2

Figure 2 shows the improvement in noise reduction obtained in three steps. In C and D it was necessary to install lighting and forced ventilation in the enclosure. In all alternatives the doors are hinged and provided with magnetic locks, the door material is acrylic sheet. For sensitive products (not coffee) hygienic problems with porous absorber may occur.

As a conclusion: Practical solutions for noise control are seldom the optimal acoustic construction. With some systematic efforts and practical compromises, the results may, however, be quite acceptable.

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STØJDÆMPNING AF DIESEL-GENERATORANLÆG

Ole Clausen
AS IKAS, Danmark

I samarbejde med bl.a. B & W Holeby Diesel A/S er der projekteret og leveret støjdæmpning til 8 stk. generatoranlæg til DSBs nye færger til Århus-Kalundborg-ruten.

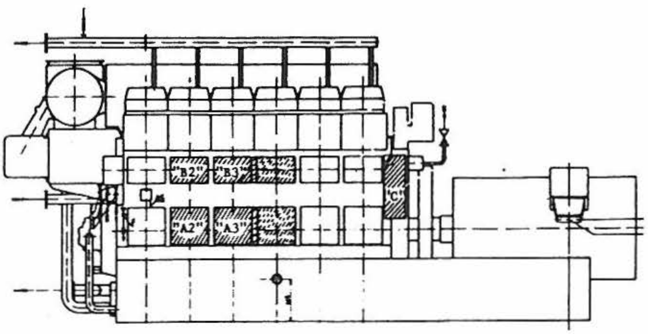

Fig. 1. Diesel-generator.

Opgaven var at opnå et middellydtrykniveau på max. 97 dB(A) målt på prøvestand i 1 meters afstand fra anlægget ved 80% belastning. Dette svarer til ca. 102 dB(A) monteret i skib.

Kildekortlægning ved lydintensitetsmålinger

Til kortlægning af støjudstrålingen fra maskinen blev der anvendt lydintensitetsmålinger. Maskinens overflade blev inddelt i 756 felter, hvor der i hvert felt blev foretaget en måling. På basis af disse resultater kunne de mest støjende zoner fastlægges. I diagrammet fig. 2 er angivet resultatet af målingerne.

Sag: B&W HOLEBY Marine GenSet Type 6S28LH-4E(LH)
LYDMÅLING NÆR MASKINENS FORSIDE, OVERSIDE OG BAGSIDE, SAMT OM TURBOLADER

Lydmålinger i frekvensområdet: 200Hz - 8kHz (mikrofonafst. 12 mm)
Måledato: 1984 MARTS 13 Datanavn: 5a

Udstrålet lyd i frekvensintervallet 200Hz til 8kHz, A-vægtet

måle-flade nr.	fladens areal m^2	intensitets-niveau dB re 1 pW/m^2	effekt-niveau dB re 1 pW	effekt i % af den totalt udstrålede lydeffekt %
761	1.64	93.2	95.4	1.8
762	.512	98.3	95.4	1.8
763	.768	99.9	98.7	3.9
764	.256	101.7	95.8	2.0
765	1.84	97.9	100.6	6.0
766	.896	103.4	102.9	10.3
767	1.89	101.3	104.0	13.4
768	.922	100.5	100.1	5.4
769	.384	105.1	100.9	6.5
770	.87	100.4	99.8	5.0
771	1.84	94.3	97.0	2.6
772	2.59	100.9	105.1	16.9
773	1.61	99.8	101.8	8.0
774	1.79	94.8	97.3	2.8
775	.512	97.8	94.9	1.6
776	.486	101.0	97.9	3.2
777	.691	99.4	97.8	3.1
778	5.88	92.4	100.1	5.3
I alt	25.4	98.7	112.8	100

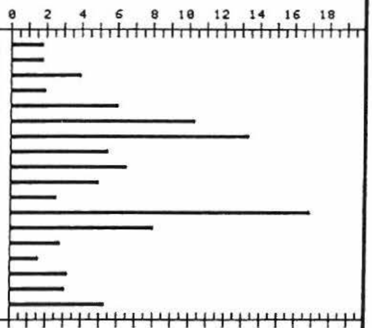

Fig. 2. Måleresultater i dB(A) (målefladenr. refererer til delarealer).

Middellydtrykniveauet blev 1 meter fra maskinen bestemt til

$$L_{p,mid} = 99,3 \text{ dB(A) re } 20 \text{ uPa}$$

Målet var altså at reducere støjniveauet med ca. 3 dB.

Støjdæmpende tiltag

På basis af lydkrav og krav om uhindret adgang til anlæg blev en "tæt på"-lyddæmpningsmodel anvendt. Det vil sige tiltag integreret på maskinen.

Følgende tiltag blev udført:

* Motortop.
  Samlet inddækning af topstykker og drivende.
  Nødvendig dæmpning: Ca. 10 dB(A).

* Oliepumpe-dæksler.
  Udskiftning af letmetal-dæksler til ståldæksler.
  Nødvendig dæmpning: Ca. 10 dB(A).

* Øvrige dæksler.
Udskiftning af tyndplade-dæksler til dæksler af tykkere materiale samt montering af strålingsformindskende beklædning på udvendig side.
Nødvendig dæmpning: Ca. 4 dB(A).

For at opnå tilstrækkelige dæmpningsværdier var det nødvendigt, at alle tiltag blev monteret svingningsdæmpet på maskinen. Endvidere blev der monteret lydabsorberende materiale i inddækningen og de strålingsformindskende beklædninger.

Denne lydabsorbent blev specielt udviklet, idet konventionelle absorbenter (mineraluld-skumplast) på grund af brandkrav/olieopsugning ikke kunne anvendes. Der blev derfor anvendt en speciel diffusionstæt olieafvisende absorbent, se fig. 3.

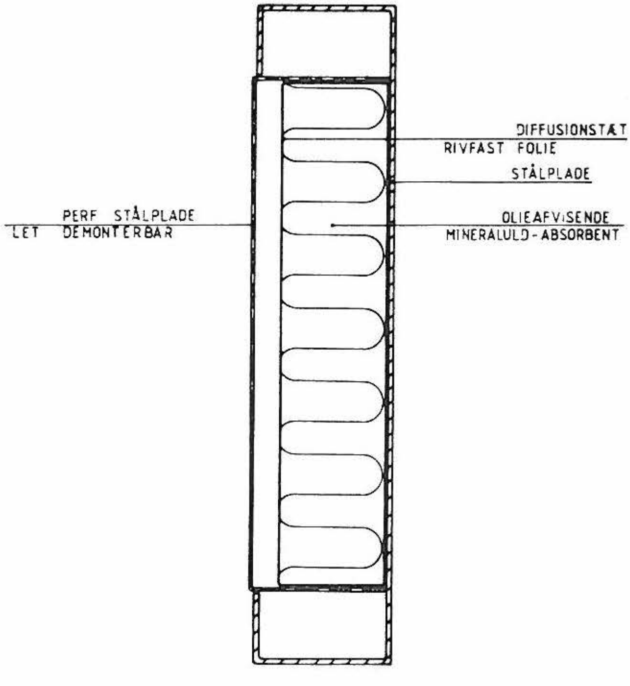

Fig. 3. Olieafvisende motorrum-absorbent

Afsluttende bemærkninger

Efter montage af de støjdæmpende tiltag blev kontrolstøjmåling på prøvestand foretaget, og fig. 4 angiver niveauerne før og efter støjdæmpning.

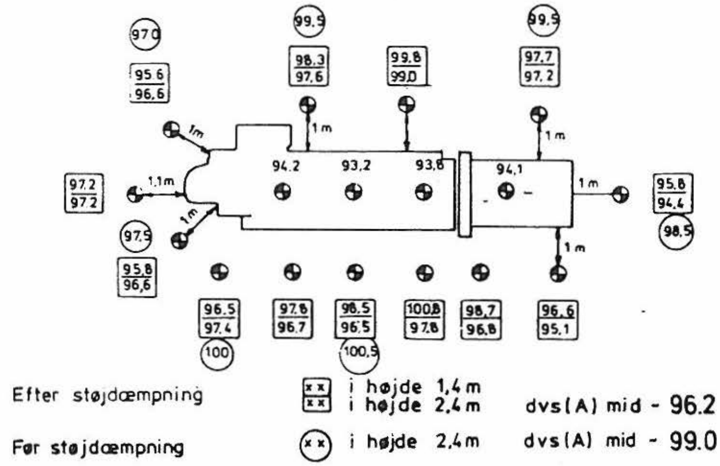

Fig. 4. Før/efter måling.

Efter montage i skib er lydtrykmålinger foretaget, og max.-niveau registreret til 101-102 dB(A) for 1 generatoranlæg i drift.

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PROJEKT INOM STENINDUSTRIN OCH MANUELLA GLASINDUSTRIN.
Efterhuggning samt sågning i sten och glas.

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Inledning
Under 1984 - 85 har tre utredningar, finansierade av Arbetarskyddsfonden i Sverige, genomförts inom stenindustrin samt inom glasindustrin. Ett projekt har avsett efterhuggningsarbetsplatser, här behandlas endast en del av projektet som avser luftljud, vibrationer och vibrationsdämpning på bildhuggarmaskiner, och två projekt har omfattat ljudalstring vid sågning med dämpade diamantsågklingor i sten samt i manuellt tillverkat kristallglas.

Bildhuggarmaskiner
Ljud- och vibrationsmätningar på bildhuggarmaskinerna Fröhlich & Klüpfel typ PM resp Atlas Copco BHV-16 standard och med extra vibrationsisolering har utförts under produktionsmässig drift vid efterhuggning i gravvårdar.
Ljudmätningarna på Fröhlich & Klüpfel och BHV-16 visar att registrerade ljudtrycksnivåer överskrider kriteriet för hörselskaderisk vid oskyddat öra. Vid nyttjande av hörselskydd under arbete med bildhuggarmaskinerna finnes ej risk för hörselskada. Prov med utloppsljuddämpare på maskinerna visar att det högfrekventa ljudbidraget reduceras, dock ej så att hörselskaderisk vid oskyddat öra elimineras.
Vibrationsmätningarna visar att dominerande vibrationer, med hänsyn till vibrationskriterier, ofta uppstår vid slagtalsfrekvensen i slagriktningen längs maskin och mejsel.
Vibrationshastighetsnivåerna inom frekvensintervallet 6,3 - 1250 Hz i slagriktningen på mejsel och maskin överskrider vid samtliga provade driftsfall det av Arbetarskyddsstyrelsen föreslagna gränsvärdet för 4 timmars daglig exponering. Enligt ASS förslag till anvisning kan en tillåten daglig exponeringstid, med avseende på vibrationer i mejsel, vara ca 10 minuter. Även i tvärriktningen på mejseln överskrides det föreslagna 4 timmars gränsvärdet väsentligt. Vibrationerna på maskinerna i längsriktningen visar att generellt kan

tillåtas en daglig blandad huggning under ca 30 minuter för båda maskinerna. I tvärriktningen på maskinerna är registrerade vibrationer väsentligt lägre än föreslaget gränsvärde. Extra vibrationsisolering av BHV-16 visar att vibrationshastighetsnivån vid slagtalsfrekvensen reduceras med 2 - 3 dB, d v s ca en fördubbling av tillåten exponering. I frekvensområdet över slagtalsfrekvensen kan erhålles en reduktion av ca 10 dB.

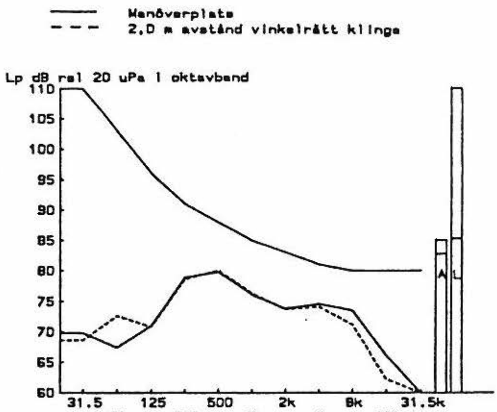
Dämpade sågklingor för kantsågning inom stenindustrin

Arbetsprocessen sågning i granit och lössten med diamantsågklinga inom stenindustrin alstrar ett mycket högt buller som kan vara 95 - 115 dB(A) vid manöverplats. De höga ljudtrycksnivåerna genereras i första hand av påtvingade böjsvängningar i diamantsågklingans liv, som i sin tur effektivt avstrålar starka ljudvågor med högfrekvent karaktär till produktionslokalerna.

I en utredning från 1975 konstaterades att sågning i granit med dämpad diamantsågklinga och optimala parameterdata medförde icke hörselskadliga ljudtrycksnivåer vid manöverplats samt dubbelt så stor avverkning per tidsenhet.

Syftet med denna utredning var att förse branschen med ett jämförbart underlag för bedömning av på marknaden förekommande dämpade diamantsågklingor för sågning i granit och lössten.

Beräknade ekvivalenta kontinuerliga ljudtrycksnivåer/medelarbetsdag visar att hörselskaderiskkriteriet ej behöver överskridas vid sågning i granit med dämpad diamantsågklinga, se vidstående diagram.



Avverkningen har varit 50% större under körning av livslängdsproven med de dämpade diamantsågklingorna än vid normalt nyttjade sågparameterdata.

Livslängden för en odämpad diamantsågklinga varierar normalt mellan 60 - 100 m^2.

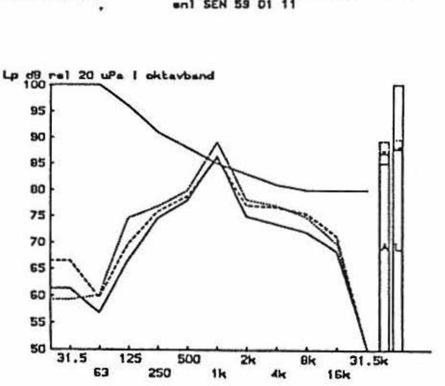
Utförda livslängdsprov med dämpade diamantsågklingor för granit visar att livslängden normalt är inom intervallet 70 - 80 m^2.

Med dämpade diamantsågklingor erhålles ej kortare livslängd än med odämpade standardsågklingor. Högre inköpspris för de dämpade diamantsågklingorna medför dock något högre avverkningskostnad/m^2.

Det bör påpekas att det kan förekomma ca 3 dB:s nivådifferenser , inom ur hörselskadesynpunkt dominerande oktavband, mellan likvärdiga dämpade diamantsågklingor från samma leverantör.

Ljudmätningar på dämpade klingor under körning av livslängdsproven indikerar att ljudgenereringen kan öka något under den första förslitningsperioden (20 - 25% nedslitning) för att därefter minska vid ca 50% förslitning. Då sågkling-

an är nästan utsliten synes ljudtrycksnivåerna öka drastiskt. Beträffande totalbedömning av i utredningen ingående dämpade diamantsågklingor, för kantsågning i granit, kan konstateras att skillnaden i pris är väsentligt större än skillnaderna i avverkning och utstrålade ljudtrycksnivåer. Klingkostnaden/m2 avverkad sten blir för de bästa klingorna 75:00 - 85:00 kronor. Totalt sett synes BTC, Diamant Boart och Diamant-D klingorna vara något bättre än övriga klingor.
Sågning i lössten (Ekebergsmarmor, Ölandskalksten G1 och röd jämtlandskalksten) visar att de ljudtrycksnivåer som genereras vid genomsågning med dämpad diamantsågklinga av lösstensskivor, för en medelarbetsdag, ej medför överskridande av kriterie för hörselskaderisk, se vidstående diagram. Det bör påpekas att vid sågning i lössten med angivna sågparameterdata har tre av de dämpade klingorna, Diamant Boart, Diamant-D och Dimas medfört en oacceptabel snittyta vid genomsågning av röd jämtlandskalksten med bordsmatningshastigheten 2,5 m/minut. En lägre bordsmatningshastighet måste användas för dessa klingor. Skillnaderna i ljudutstrålning vid sågning i lössten med de olika fabrikaten av dämpade diamantsågklingor är relativt små.



Den väsentligaste skillnaden är, som även konstaterades för granitklingorna, priset för de olika fabrikaten. Lägsta pris är 1.500:-/styck och högsta pris är 2.850:-/styck.
Planerat genomförande av livslängdsprov med de dämpade diamantsågklingorna har ej varit möjligt att genomföra på grund av att vi ej kunnat finna något stenföretag som kunde ställa upp med produktionskörning för livslängdsproven.

Dämpade sågklingor för manuell sågning inom glasbruken
Prov med odämpade och dämpade, osegmenterade (otandade, kontinuerligt segment runt klingans periferi) och segmenterade (tandade) diamantsågklingor har utförts under 1984 - 85.
Huvudsyftet har varit att för de manuella glasbruken söka fastställa vilka reduktioner av ljudtrycksnivåer som kan uppnås med olika ljuddämpade diamantsågklingor i jämförelse med standardsågklingor. Redovisa praktiska erfarenheter från sågning med dämpade klingor samt söka fastställa lämpliga parameterdata för lägsta ljudgenerering resp bästa avverkning. Dessutom har prov utförts för att bestämma glasproduktens inverkan på ljudutstrålningen.
Vid sågning av glasprodukter med diamantsågklingor genereras mycket höga ljudtrycksnivåer i frekvensområdet över 500 Hz. Ur hörselskadesynpunkt är mest besvärande frekvensområde 4000 - 16000 Hz.
Mätning av ljudtrycksnivåer vid operatörens högra och vänstra öra, avverkning samt matningskraft har utförts vid såg-

ning med olika spindelvarvtal inom intervallet 1200 - 3400 varv/minut.
Mätresultaten visar att ljudtrycksnivåerna inom högfrekvensområdet minskar väsentligt då sågning utföres med en dämpad diamantsågklinga.
Vid användande av dämpade diamantsågklingor för sågning i olika typer av glasprodukter erhålles en reduktion av ljudtrycksnivåerna med ca 10 dB inom det ur hörselskadesynpunkt mest besvärande frekvensområdet 4000 - 16000 Hz. Med antagandet att verklig effektiv tid med sågning/arbetsdag ej är längre än 1 - 2 timmar kan man påstå att det vid användande av dämpade sågklingor ej alstras hörselskadligt buller.
Av ljudmätresultaten samt av maskinfabrikanternas rekommendationer framgår att sågning i glasprodukter med diamantsågklingor, diameter 300 - 350 mm bör utföras vid högst ett spindelvarvtal av 2000 varv/minut. Sågning med ovannämnda varvtal orsakar väsentligt lägre bulleralstring än vad som genereras vid spindelvarvtal på 3400 varv/ minut.
Skillnaden i registrerad ljudnivå kan vara ca 5 - 10 dB(A).
Det bör också påpekas att automatsågar för klingdiametrar 300 - 400 mm körs med ett fast spindelvarvtal av 2000 r/m.
Produktionsmässigt medför, för vissa nyttjade klingor, ett lägre spindelvarvtal en något minskad produktivitet. Hänsyn bör även tas till använd matningskraft under sågning, varvid produktivitet anges som avverkad glasyta per tidsenhet och per matningskraft.
Sågning med lågt spindelvarvtal < 2000 varv/minut medför en något finare snittyta och mindre kantutslag än sågning med höga spindelvarvtal 3400 varv/minut. Sannolikt blir livslängden på klingorna något kortare vid sågning med höga spindelvarvtal.
Klingor som har diamantsegment med större diamantkornstorlek 50/80 mesh orsakar högre ljudgenerering än klingor med segment med mindre kornstorlek 80/100 mesh, vid samma diamantkoncentration. Bindemedlets inverkan på ljudgenerering och produktivitet har ej kunnat fastställas. Vid hög diamantkoncentration 1,32 carat/cm^3 synes segmenten ha en tendens till att sätta igen sig och förlora produktivitet samt medföra en ökad ljudgenerering.
Ljudutstrålningen från den sågade glasprodukten kan vid sågning med dämpad diamantsågklinga i tunnväggig produkt (< 10mm tjocklek) medföra en höjning av de högfrekventa ljudtrycksnivåerna med ca 2 dB. Vid sågning i homogent och tjockare glas synes ej ljudutstrålningen från produkten ha någon betydelse för registrerade ljudtrycksnivåer.
Någon markant skillnad i uppnådd avverkning mellan odämpade och dämpade diamantsågklingor synes ej. Då sågning sker i homogent tjockt glas blir dock avverkningen, med i projektet nyttjade osegmenterade klingor mindre än då sågning sker med segmenterade klingor.
Bästa alternativ av dämpad, segmenterad diamantsågklinga med avseende på ljudgenerering och produktivitet synes enligt utförda provkörningar vara följande:

- Diamant Boart 350 mm Karlebo
- Tyrolit 350 mm
- Winter 350 mm Dilox AB

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NOISE REDUCTION OF SLOW AXIAL FANS BY MEANS OF BLADE REPLACEMENT

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1. INTRODUCTION

Axial fans in cooling towers are often one of the predominant noise sources in industrial plants. Their ability to provide a sufficient cooling power is essential since a reduced cooling will reduce the efficiency of the plant.

If a noise reduction programme has to be done then the following three possibilities exist.

a) Installation of traditional absorption silencers both on the suction and on the pressure side. Addition of extra fan units to compensate for the pressure drop caused by the silencers.

b) Use of another cooling concept or total replacement of the cooling tower.

c) Reduction of the blade speed and compensation for the reduced air flow by means of more efficient blades.

This paper describes a case, where the last solution has been applied.

2. THEORY

Measured in the atmosphere at $0^{o}C$ it can be shown that the total sound power level can be expressed as (1):

$$W = 5.5 \cdot 10^{-12} \cdot B \cdot D_R \cdot D \cdot U^6 \qquad (1)$$

$$D = D_t + 2(0.37 \cdot C/4)\ R^{-0.2} \qquad (2)$$

where

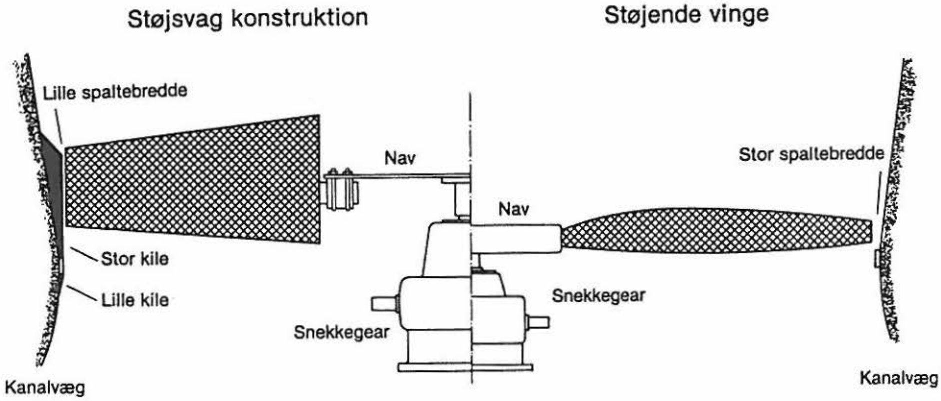
- W : Radiated sound power (Watt)
- B : Number of blades
- D_R: Diameter of the impeller (m)
- U : Tip speed of the fan (m/s)
- D_t: Blade trailing edge thickness at the rms radius of the blade (m)
- C : Chord length of the blade (m)
- R : Reynolds number based upon the chord length and air velocity parallel to the wing surface at the rms radius of the blade.

RMS radius = $D_R/2$ is valid for both blade types listed below.

Formula (1) is based upon the theory for the thickness of a turbulent boundary layer on both the pressure side and the suction side of the wing. The noise is generated in the wake behind the trailing edge.

3. PRACTICAL APPLICATION

Formula (1) has been applied, when considering replacement of a set of blades on an axial cooling tower fan. A sketch of the blade of the old and the new impeller is shown below.



Vertical sketch showing the old and new impeller.

The main data and measurement results are shown in the table below:

	Old impeller	New impeller
Blade type	Aerex ASD 156	Stork Hengelo VPFT
L_W, measured	118 dB(lin)	114 dB(lin)
L_W, predicted	115 dB(lin)	112 dB(lin)
L_{WA}, measured	108 dB(A)	99 dB(A)
Volume flow, Q	163 m^3/s	163 m^3/s
Static pressure, P_T	176 Pa	184 Pa
Rotational frequen.	302 rpm	197 rpm
Diameter	3.96 m	3.96 m
Chord length of tip	0.19 m	0.42 m
Number of blades	6	8
Trailing edge blade thickness	40 mm	14 mm
Tip speed	62 m/s	41 m/s
Pitch angle	28^o	29^o
Blade area	0.45 m^2	0.99 m^2

Main data and measurement results.

As can be seen from the main data and measurement results the predicted value of L_W has a tendency to be lower than the measured figure. No obvious explanation can be given for this discrepancy but the deviation may be caused by the uncertainty in the thickness of the make. In the actual case the calculated thickness was totally dominated by the trailing edge thickness, whereas the contribution from the turbulent boundary layer was only a few millimetres. In reality the turbulent boundary layer may have had a larger thickness giving a smaller deviation between the predicted and measured value of L_W.

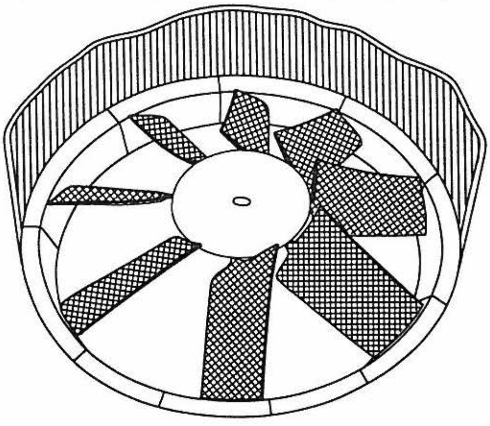
4. DISCUSSION

Another reason why the measured value of L_W was higher than the one predicted could be the effect of camber which was 22^o. In this case the noise level increases more than can be analytically expressed because of the large increase of the wake caused by separation along the suction side of the wing surface.

For the new impeller the camber angle was 10^{o} and since the lower limit for suction side separation lies around 20^{o} no unwanted separation occurs in this case.

An attempt to predict the A-weighted sound power level L_{WA} has been made but the result of this prediction did not coincide with the measured value. An explanation of this deviation may be that the rotational frequency of the fan was so small compared with normal axial fans that the traditional methods for predicting the A-weighted level did not apply.

The qualitative reason for the obtained A-weighted noise reduction of 9 dB is therefore the well known fact that a slower speed primarily reduces the high frequency components of the spectrum and hence also the A-weighted sound power level.



Sketch of the new impeller, see from above.

5. ACKNOWLEDGEMENT

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IV : The Effects of tip clearance on the noise of low pressure axial and mixed flow fans (J.S&V, 105, 291-308).

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ANNOYANCE FROM LOW FREQUENCY AND INFRASONIC NOISE

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INTRODUCTION

The A-weighted sound level is by far the most widely used objective measure of noise. For the majority of noise sources, there is a fair correlation between A-weighted sound level and annoyance. Only minor corrections of the A-weighted levels are needed, if, for instance, the noise contains pure tones or is impulsive.

If the noise contains considerable energy in the low audio frequency region 20-125 Hz, the correlation breaks down. Countless case stories have told about nuisances from power plants, compressors and ventilating systems at places where the A-weighted levels were low and all restrictions were observed.

Problems also exist at infrasonic frequencies. Several investigations have shown that infrasound may affect people. The first articles that discussed this appeared in the mid sixties and they suggested physiological effects and effects on task performance from even very low levels of infrasound. Later on, these effects have proven to be exaggerated and it is now believed that the nuisances from infrasound derive from the fact that humans can hear infrasound when the sound pressure level is sufficiently high.

The following describes results from loudness and annoyance experiments at low and infrasonic frequencies. The possibility of deriving new weighting curves for these frequencies is discussed. Emphasis is made on the special infrasonic weighting curves G1 and G2 [1].

LOUDNESS EXPERIMENTS

Curves of equal loudness were determined for pure tones in the frequency range 2-63 Hz using the method of maximum likelihood [2]. The results are given in Figure 1 together with a threshold curve based on recent data given in the literature.

From the threshold curve it is seen that the lower the frequency, the greater the sound pressure level must be in order to make the tone audible. It is worth noting that the curves go down to at least 2 Hz. The widely accepted limit of audibility around 20 Hz does not exist, although the tonal character of the sound disappears below this frequency.

It is also seen that the curves are much closer in the infrasonic region than at audio frequencies. For example, the distance between the 20 and the 80 phon curves has decreased from 60 dB at 1 kHz to approximately 16 dB at 8 Hz. Consequently, infrasound only a few dB above the hearing threshold will seem loud and possibly annoying. It is also possible to explain the fact that a small change in the infrasound content of a complex sound is able to change the loudness considerably.

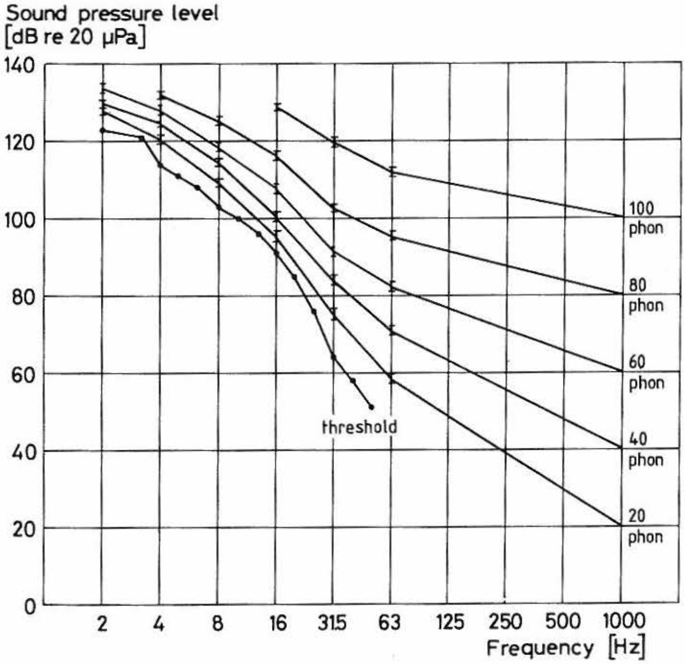

Figure 1. Curves of equal loudness. Vertical bars indicate ±1 standard error of mean. The threshold curve is based on four recent studies.

ANNOYANCE EXPERIMENTS

The loudness curves were determined through direct comparisons between two short tones. A similar procedure is not possible when determining curves of equal annoyance, as it is believed that the duration of the stimuli must be much longer in order to obtain a proper assessment of the annoyance. Therefore, an indirect method was used. After being exposed to a stimulus for 15 minutes, subjects rated the annoyance on a 150 mm scale, of which the ends were labelled "not at all annoying" and "very annoying". From these ratings, curves of equal annoyance were determined [3] (not shown).

The equal annoyance curves are very similar to the equal loudness curves, and the results seem to indicate that the annoyance from infrasound is closely related to the loudness of the sound. As the loudness curves are determined with the best accuracy, they will be referred to in the following.

WEIGHTING CURVES FOR LOW AUDIO FREQUENCIES

Our annoyance experiments included exposures to 1 kHz noise bands and to pure tones and noise bands at 31.5 Hz. These frequencies are in the range supposed to be covered by the A curve. Figure 2 shows annoyance rating versus A-weighted sound level for these frequencies. It is clearly seen that the annoyance from 31.5 Hz (unfilled circles) does not follow the same line as the annoyance from 1 kHz (filled circles). The annoyance from 31.5 Hz rises much steeper than that from 1 kHz. The two regression lines intersect at approximately 45 dB.

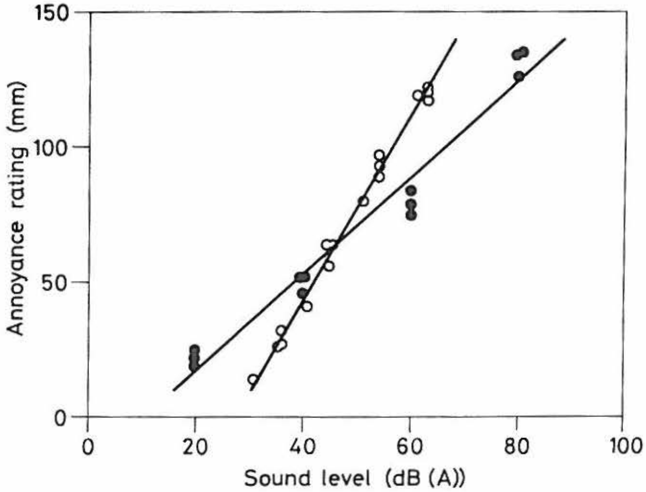

Figure 2. Annoyance rating versus A-weighted sound level. Filled circles represent mean values for 1 kHz exposures. Unfilled circles represent mean values at 31.5 Hz. The lines are regression lines (r^2=0.97 for the filled circles, 0.99 for the unfilled).

The origin of the A curve explains this. The A curve is approximately the reciprocal of the 40 phon curve. Assuming a close relationship between loudness and annoyance, then A-weighted levels will reflect the annoyance of noise with levels around 40 phon. For low frequencies at levels well below 40 phon the annoyance is expected to be lower than that predicted by the A-weighted level. At levels much above 40 phon, the annoyance is expected to be higher than that predicted by the A-weighted level. This is exactly what is seen in Figure 2.

Originally, the intention was that the A curve should be used only at levels around 40 phon, while the B and C curves should be used at higher levels. This procedure is almost never used in real life and that is most probably the reason why it has been so difficult to obtain a good correlation between objective measures and subjective ratings for noises containing considerable low frequency energy.

WEIGHTING CURVES FOR INFRASONIC FREQUENCIES

The A curve is not intended to cover the infrasonic frequency range and frequencies below 20 Hz are only transferred through the A filter due to the finite slope of the filter low frequency cut-off. When the sound level of infrasound is measured with a commercial A-weighting sound level meter, the obtained values will be low - and they will depend on the particular sound level meter since the tolerances of the A curve are large at these frequencies. Consequently, it has no meaning to refer to the A-weighted level of noise having a significant content of infrasound.

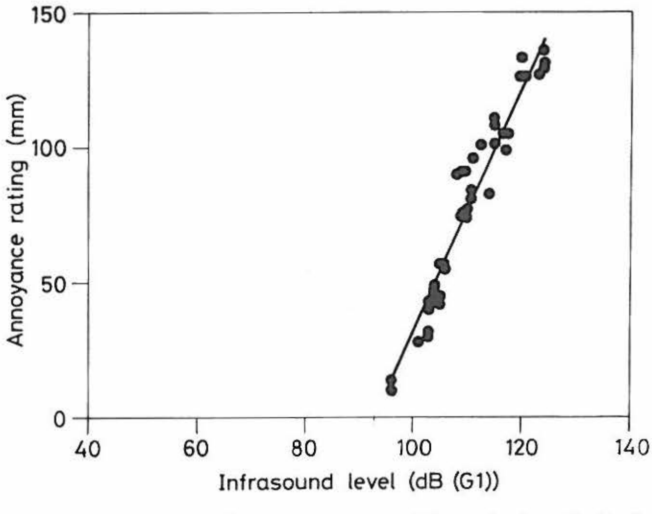

Figure 3. Annoyance rating versus G1-weighted infrasound level. Filled circles represent mean values for all infrasonic exposures in our experiments. The line is a regression line (r^2=0.93).

It is seen from Figure 1 that the loudness curves are almost parallel in the infrasonic region. Therefore, it may be possible to develope a weighting curve suitable for measuring loudness and annoyance of infrasound. The mean slope of the curves are approximately 12 dB per octave. A weighting curve with this slope and restricted to the frequency range 1-20 Hz is proposed by ISO and named the G1 curve. A weighting curve for the same frequency range but with a slope of 6 dB per octave is named the G2 curve [1]. Having the slope of the equal loudness curves in mind, one would expect the G1 curve to give a fair indication of the loudness and annoyance associated with infrasound.

In Figure 3 mean annoyance rating is shown versus G1-weighted infrasound level for all infrasonic exposures from our experiments. The figure shows a very close linear relationship (coefficient of correlation r^2=0.93).

Figure 4 shows the same results versus the G2-weighted infrasound level. Here r^2=0.77 and it is clearly illustrated that the G2 curve provides a measure of the annoyance that is much inferior to that of the G1 curve.

Figure 3 showed a good correlation between G1-weighted infrasound levels and annoyance rating. Thus, if a "one-figure" measurement is wanted for infrasound, the G1 curve might be a good choice. However, this curve provides a frequency weighting only and G1-weighted levels do not reflect the fact that the annoyance increases steeply above the threshold. Thus, the conversion shown in Figure 5 may be useful. For a given G1-weighted infrasound level it can be read which A-weighted level that causes the same rating of annoyance.

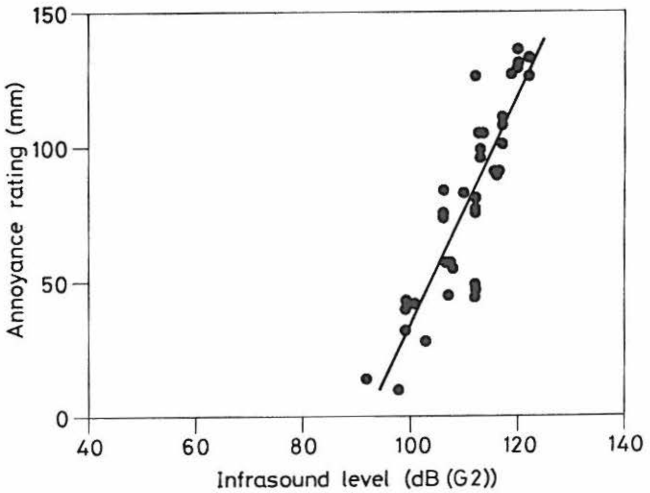

Figure 4. Annoyance rating versus G2-weighted infrasound level. Filled circles represent mean values for all infrasonic exposures in our experiments. The line is a regression line (r^2=0.77).

CONCLUSION

The proposed ISO G1 weighting curve provides an objective measure that correlates very well with subjective annoyance ratings for infrasonic frequencies. Values obtained with the proposed G2 weighting curve do not correlate as well.

Because of the low dynamic range of the ear at infrasonic frequencies, care should be taking when evaluating G1-weighted levels. The numerical values should not be directly compared to A-weighted levels.

Low audio frequencies are not covered by the proposed G-weighting curves and they are insufficiently covered by the A curve. A possible solution may be the originally intended level dependent use of the A, B and C curves. Further research is needed in this area.

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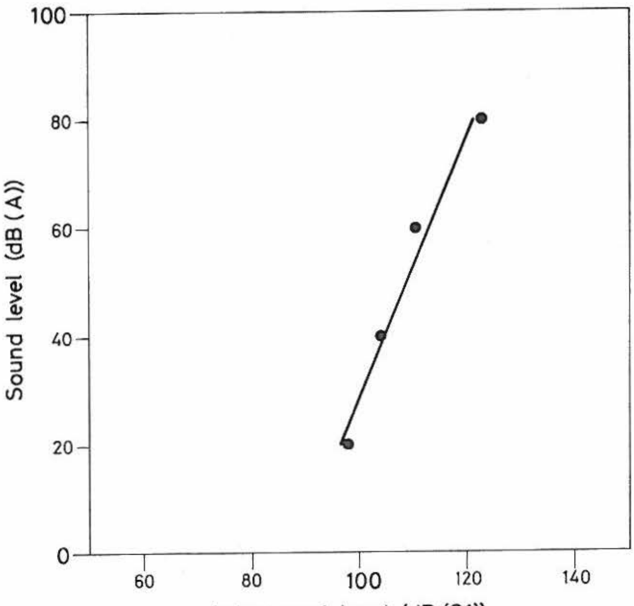

Figure 5. Conversion of G1-weighted infrasound level to the A-weighted level of an audio frequency noise that causes the same rating of annoyance.

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A MODEL DESCRIBING DIFFERENCES IN TIMBRE BETWEEN LOUDSPEAKERS

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INTRODUCTION

The purpose of developing a model describing differences in timbre between loudspeakers, is twofold:

1. To convey to the loudspeaker manufacturer an 'instrument' useful in development and quality tests.
2. To gain insight in the basis of the hearing mechanism with respect to evaluation of sound sources in rooms.

The model presented in this paper is intended as a first order approximation. Research is continued in order to increase the 'goodness of fit' and utility in other areas of electroacoustics. The results used in verification of the model originate from a research project dealing with different aspects of listening tests on loudspeakers.

THEORY

Timbre is a multidimensional attribute of sound according to the definition given by the American Standards Association [1]. However, research has shown that 3 principal dimensions are adequate in describing timbre. Different aspects of the stimulus spectrum have been found to be highly correlated with the most important dimensions in the reduced timbre space:

1 Plomp [2] describes an experiment where the subjects judged differences in timbre between single tones from musical instruments. He found the subjective results to be correlated ($r = 0.85$) with the objective differences found by:

$$D_{ij} = \left(\sum_{n=1}^{m} |L_{i,n} - L_{j,n}|^2 \right)^{\frac{1}{2}} \qquad (1)$$

where

D_{ij} = difference in 1/3 octave spectrum between tones i and j

$L_{i,n}$ = sound pressure level of tone i in 1/3 octave n

m = number of 1/3 octaves.

2 Staffeldt [4] found that two loudspeaker systems are judged to have equal timbre, if their 1/3 octave spectra, measured at the entrance of the listener's ear canal with pink noise, are equal. This result was found in listening tests using broad band noise, speech and music.

3 Von Bismark [3] found the verbal attribute 'sharpness' to be highly correlated with timbre. He has shown that the sharpness of narrow band (1/3 oct.) noise, compared to a noise band with fixed centre frequency 1 kHz, is increasing with the centre frequency of the noise band.

The results of Plomp and Staffeldt indicate that differences in the 1/3 octave spectrum calculated according to (1) are adequate in describing differences in timbre between loudspeakers.

Von Bismark's results indicate that some kind of weighting function, emphasizing differences in the upper part of the spectrum, should be included in the model.

Another effect to be considered is masking. The effects of masking mean that the spectral parts of the steady-state spectrum, having a negative slope, should be emphasized. If the long time average spectrum of a piece of music is investigated it is revealed that the upper (high frequency) part of the spectrum has negative slope.

It is noted that the results of Bismark are not due to masking effects.

Thus two results exist indicating that differences in SPL in certain parts of the spectrum have a major influence on the perceived difference in timbre.

Thus the following revised version of (1) is suggested as an objective measure of differences in timbre between loudspeakers.

$$\Delta T_{ij} = \left(\sum_{n=1}^{m} |L_{i,n} - L_{j,n}|^{p}\right)^{1/p} \tag{2}$$

where

ΔT_{ij} = difference in timbre between loudspeakers i and j

$L_{i,n}$ = sound pressure level in 1/3 octave n, of the long-time average stimulus spectrum, measured at the listening position (see definition of m)

m = number of relevant 1/3 octaves (stimulus dependent)

p = variable.

The 1/3 octave SPL's are measured at the listening position (no subject present) in the listening room with an omnidirectional microphone.

EXPERIMENTAL DESIGN

The listening tests were conducted in a listening room built in accordance with the IEC recommendation [5]. The experimental set-up comprised four mono loudspeakers with individual amplifiers, a tape recorder and a subject-controlled switching system. The stimuli included speech and three pieces of music. Five subjects participated judging the loudspeakers with respect to timbre only. The procedure paired comparisons with stating of a preference was used to record the subject answers. All subjects made one replication of the experiment.

RESULTS

In order to reveal subjectively important dimensions the results were analysed by multidimensional scaling technique.

Two principal dimensions were found. Dimension 1 was shown to be correlated with differences in SPL in 1/3 octaves from spectral parts with negative slope. Only these 1/3 octaves were used in calculation of ΔT_{ij}. The subjective results were found as the differences in the normalized scores for each pair of loudspeakers. The subject-averaged results were used in calculation of the correlation coefficients. The correlation as a function of p was calculated for each stimulus separately. The results are shown in Fig. 1.

DISCUSSION

The importance of spectral parts having a negative slope, indicates that masking effects should be included in the model. However, the major portion of relevant 1/3 octaves were found in the upper part of the spectrum, confirming von Bismark's results.

The stationarity in SPL as a function of time differed strongly for the stimuli used. However, the results obtained indicate that the long-time average spectrum (time constant equal to duration of the stimulus) is adequate for describing timbre even for strongly time varying stimuli. Thus the results found by Staffeldt are confirmed.

The high correlation coefficients shown in Fig. 1 clearly show that the model (2) gives a firm foundation for further research. Subjects to be covered include

a) the use of 1/3 octaves as approximation of the critical bandwidth. Results found by Green et al. [6] have shown that some kind of cross-talk between adjacent critical bands influences the perception of level changes.

b) The integration time involved in timbre perception. The results found by Staffeldt are only preliminary and further research is needed.

c) Dependence of the model on experimental procedure used in the listening tests and stimuli. Results of other experiments indicate that the model is capabale of revelaing small differences between the loudspeakers. Thus the procedure and stimuli must be able to reveal these differences. As the last part of an experiment, the subjects ranked the stimuli with respect to ability to reveal differences between the loudspeakers. An identical ranking is found if the stimuli are ranked according to the maximal correlation coefficient obtained.

d) The importance of von Bismark's results and masking effects. The obtained results indicate that both are important, however, no firm conclusion could be made with respect to the relative importance.

e) The importance of measuring the spectrum at the entrance of the ear canal. The results obtained give evidence to the simple measuring procedure used in this experiment. However, the resolution reached in the final model determines the accuracy needed in the measurements.

CONCLUSION

A model which describes differences in timbre between loudspeakers has been developed. MDS analyses show that SPL differences in 1/3 octaves, found in spectral parts with negative slope, are principal for the perceived differences in timbre. The long-time average 1/3 octave stimulus spectrum is found to be adequate in describing timbre for time-varying stimuli such as speech and music. The 1/3 octave spectrum is measured at the listening position in the listening room, with time constant equal to duration of stimulus. The goodness of fit for the model is found to be dependent of resolution in experimental procedure and stimuli used in the experiment. For the optimal stimuli and experimental procedure a correlation coefficient of 0.9 was reached.

ACKNOWLEDGEMENTS

The author would like to thank associate professor O. Juhl Pedersen for careful reading of the manuscript and for numerous discussions.

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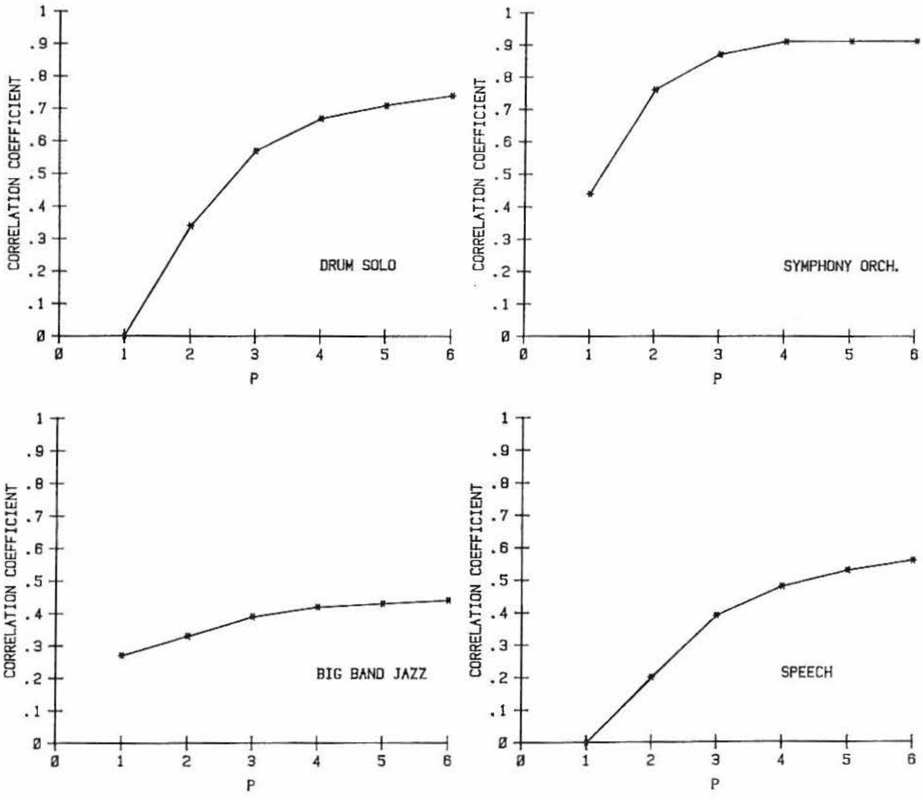

Fig. 1 *Correlation between subjective and objective results. The probability of getting, by chance, a correlation above 0.70 is 10%.*

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Investigations of Helmholtz resonators

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At Institute of Physics, University of Oslo, there is an old collection of 19 Helmholtz resonators. The resonance "frequencies" are given in musical terms (SOL) and not as frequency in Hz. It seems that some of the resonators have been tuned to certain musical notes.

Resonators like those were the frequency analysers to that time. It may be of interest to investigate this equipment with the facilities of our time.

The cavities of the resonators are approximately spheres, the volumes of which were determined by filling with water. Each sphere has an opening (sound opening) for exposing the volume to the sound, and a small opening with a short "tap" intended for listening (listening opening).

The resonance frequencies were observed both by listening, connecting a stethoscope to the listening opening, and with a microphone (1/4") connected to the listening opening by a short rubber tube. All measurements were made under free field conditions. The diameters of the sound openings is d. There is some uncertainty in measuring the geometrical length of the sound opening, as the connection to the sphere is rather curved. In any case it is less than 10 mm.

The resonance frequency f_o of a Helmholtz resonator is usually given by

$$\omega_o = 2\pi f_o = c\sqrt{\frac{S}{l'V}} \ .$$

Here S is the area of the sound opening, V the voume of the cavity and l' the corrected length of the sound opening. The velocity of sound is c.

Here all quantities except l' can be measured. The values of l' are calculated from the formula above.

Some examples are given in the table.

	V/cm^3	d/cm	l/cm	List f_o/H7	Mic. f_o/H7	Mic. l'/cm	± 3dB Q
1	2065	3,9	0,8	194	196	~ 4,49	~33
2	250	1,75	0.1	448	445	~1,45	~25
3	24	1,2	0,9	1260	1245	~0,91	~20

Velocity of sound c = 343 m/s

Conclusion: 1) The use of Helmholtz resonators as frequency standards is very much dependent on the corrected value l' of the sound opening.
2) To obtain the results about frequency distribution in sound "the old men" must have been very clever in observation.

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STÖRNINGSUPPLEVELSE AV LÅGFREKVENT BULLER

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Bakgrund

Under de senaste 20 åren har en mångfald undersökningar genomförts och resultat från ett flertal utredningar har publicerats avseende människors och djurs påverkan av infraljud. De flesta ledande forskarna inom infraljudsområdet och det samlade forskningsunderlaget visar att enbart infraljud inte utgör något uppenbart, direkt problem för människan. Det förekommer dock enstaka individer, som verkar vara överkänsliga, i relation till normalindividen, för lågfrekventa störningar.
Vid konferensen "Low frequency noise and hearing" i Aalborg Danmark under 1980 framfördes från flertalet forskare att det hittills förekommit en alltför liten forskning beträffande störningar och störningsupplevelser orsakat av lågfrekvent, hörbart buller inom frekvensområdet 20 - 100 Hz, som i alltför liten grad kan indikeras genom mätning av den A-vägda ljudnivån (dB(A)). Man var relativt övertygade om att människan, vid klagomål på högt infraljudsbuller ofta i verkligheten besväras av lågfrekventa bullerstörningar. Detta verifieras också tydligt av att många föredrag behandlade problemet lågfrekvent ljud (< 200 Hz) och ej enbart infraljud vid konferensen "Low frequency noise & vibration" i London 1985.
Lågfrekventa bullerstörningar har blivit allt vanligare både i arbetsliv, i hemmiljön och på fritiden och har under lång tid inte uppmärksammats i tillräckligt hög grad. Mättekniskt har det ej heller funnits lämpliga och användbara bedömningsgrunder för lågfrekvent buller inom frekvensområdet 20 - 100 Hz.

Inledning

Sedan 1980 har en litteratursökning avseende infraljud bedrivits av IFM Akustikbyrån AB (Olle Backteman) på uppdrag

av Försvarets Materielverk, FMV:Radio (kontaktperson Bdir Gunnar Blomgren). Detta uppdrag har resulterat i en infraljudsbibliografi, upptryckt och distribuerad under hösten 1985. Förutom bibliografin har även framställts en rapport med titeln "Sammanställning av intressanta litteraturreferenser" som finnes både i en svensk och en engelsk version.
I det basmaterial, som bearbetats beträffande infraljud finns en stor del mätdata för det lågfrekventa och hörbara frekvensområdet inom olika arbetsmiljöer både internt och externt buller samt för andra verksamhetsområden, maskiner samt för olika typer av transportenheter.
Under de senaste åren, har en ökad uppmärksamhet givits lågfrekvensproblematiken för att utreda fenomenet med besvärsupplevelser av lågfrekvent buller. Detta problem har visat sig vara mycket mer utbrett än vad som tidigare antagits. **Mätning av A-vägda ljudtrycksnivåer anses ej ge ett relevant mätetal som korrelerar med besvärsresponsen hos människan.**
Ett av huvudproblemen är och har varit val av lämpligt kriterium och viktning av ljudtrycksspektrum för bedömning av lågfrekventa bullerstörningar. Besvärsupplevelse har även förekommit då den uppmätta ljudnivån dB(A) har varit relativt låg.
En utredning har indikerat att besvärsupplevelsen är korrelerad till spektrats lutning och brytpunkt mot fallande nivå snarare än till den absoluta nivån. Det finns också indikationer på att toninnehåll inom frekvensområdet 30 - 50 Hz delvis kan vara huvudansvarigt för besvärsresponsen.
Som synes har man i de flesta utredningarna använt sig av renodlade stimuli. **Inom detta projekt skall som stimuli användas spektra och delspektra som har registrerats i arbetsmiljön, i den externa miljön och boendemiljön.**

Projektidé och målsättning

Projektet kan kortfattat sammanfattas enligt följande (se figur, sid 4):

- Bearbeta, sammanställa och bedöma mätmaterial med olika typer av vägningsförfaranden omfattande frekvensområdet 2 - 20000 Hz för olika typer av bullersituationer.

- Ur bearbetat material söka utforma lämpliga typspektrum för ex vis processbuller, arbetslokalbuller, fordonsbuller, verktygsbuller, samhällsbuller o s v. Dessa typspektrum skall bestå av tre deltypspektrum, infra-, lågfrekvens samt övrigt hörbart buller.

- Med framtagna typspektrum skall experiment genomföras på normalhörande individer, varvid olika kombinationer av deltypspektrum och nivåer skall nyttjas för subjektiv bedömning av **(kalibrerad)** besvärsupplevelse, objektiva stressmätningar och audiologiska mätningar.

Målsättningen är att finna och verifiera ett vägningsförfarande, som för samtliga typer av störningsspektrum väl korrelerar med besvärsresponsen för lågfrekvent ljud och som även förhoppningsvis objektivt kan styrkas med audiologiska resultat.

Projektbeskrivning

Med hänsyn till att projektet kommer att bli både volymsmässigt och tidsmässigt mycket omfattande har det indelats i tre alternativt fyra etapper.

Etapp 1 avser att omfatta kompletterande mätningar och sammanställningar av mätmaterial, vägningar, bedömningar samt utarbetande av typspektrum.

Etapp 2 omfattar experimentella, subjektiva försöksserier med besvärsrespons och medicinska mätningar på normalhörande individer. Detta kommer att innebära kortvarig bullerstimuli.

Etapp 3 omfattar experimentella, audiologiska försöksserier med huvudsakligen de typspektra och nivåer samt kombinationer med deltypspektra som tidigare givits stark besvärsrespons.

Etapp 4 kan tänkas omfatta en experimentell försöksserie på djur för bedömning av direkta skaderisker. Behöver troligen i sådana fall genomföras i smaråd med t ex amerikanska forskare.

Etapp 1, deletapp 1.

I projektet avses att sammanställa en jämförelse av infraljudsnivåer, lågfrekventa ljudtrycksnivåer (20 - 100 alternativt 200 Hz) samt nivåer inom det övriga hörbara frekvensområdet med olika typer av vägningar samt anmärkning om tonalstring förekommer.

Underlaget utgöres av det mätmaterial som framtagits i uppdraget för FMV samt med uppföljande kompletterande ljudmätningar inom de miljöområden som ej finnes representerat i tidigare nämnt mätmaterial. Mätningar bör först och främst genomföras inom de miljöer som bedömes har stort lågfrekvensinnehåll.

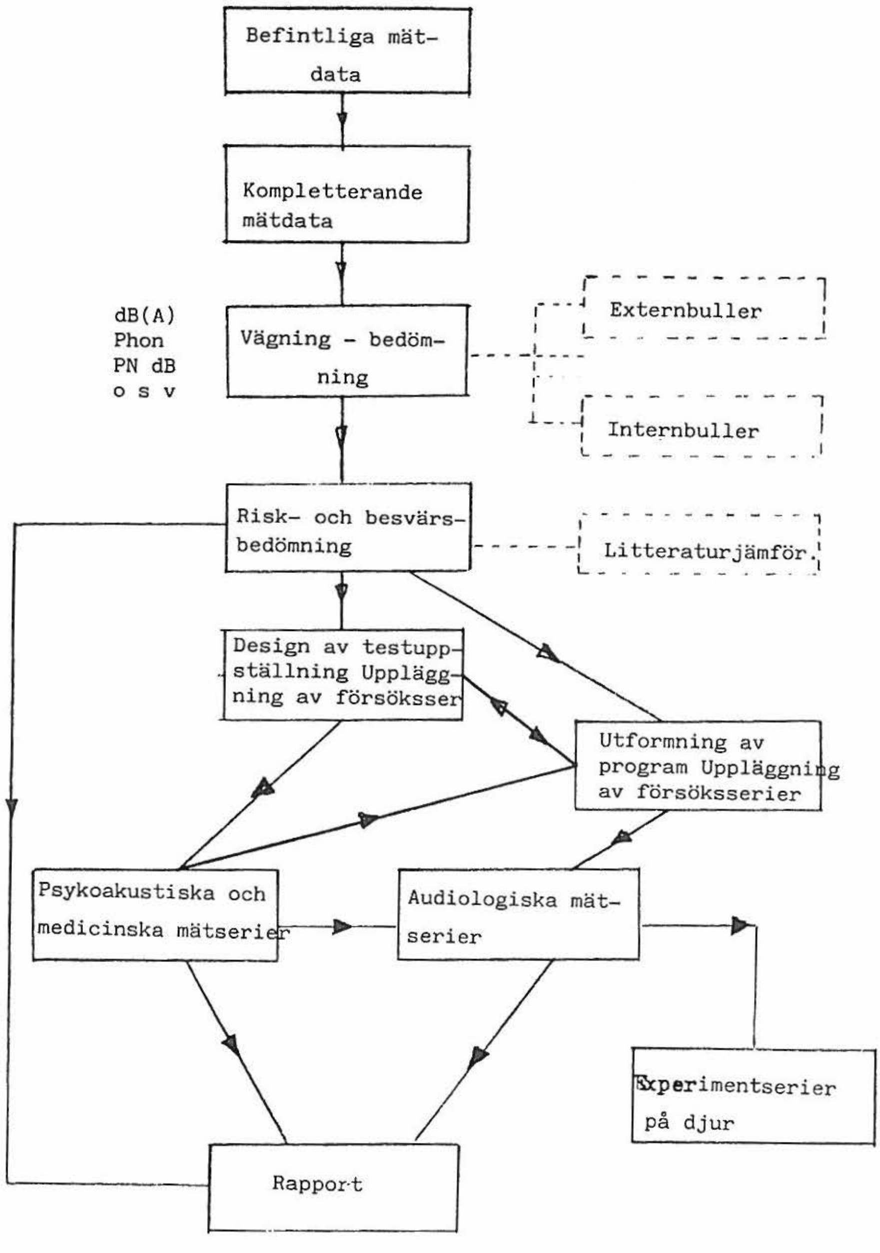
Viktningar av det totala mätmaterialet kommer att utföras inom resp delområde d v s infraljud (2 - 20 Hz), lågfrekvens (20 - 100 alt 200 Hz) samt övriga hörbara (100 alt 200 - 20000 Hz).

Följande vägningsförfaranden kommer att utnyttjas:

- Linjärt
- PN dB
- Phon
- Noy
- Sone
- dB(A)
- dB(B)
- dB(D)

samt viktning av infraljudet enligt kommande anvisningar från Arbetarskyddsstyrelsen.

PROJEKTUPPLÄGGNING



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COMMUNITY RESPONSE TO NOISE EXPOSURE
Some portuguese experience

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Introduction

Noise can be defined as unwanted sound or sound in the wrong place at the wrong time. This implies that it has an adverse effect on human beings and their environment. Noise is undesirable because it interferes with speech communication, is intense enough to damage hearing or is otherwise annoying. In this way, it affects human activities and relationships.

Community reactions to noise may take a variety of forms. The question of community noise control has become an important consideration in urban planning, in construction practices and in public administration. All stated effects of noise upon people vary greatly with the individual, depending upon the physical characteristics of the sound being classified as noise like the loudness of the sound, the duration of exposure and the time of occurence during the day. In present day conditions, especially in towns and cities, the constant and universal presence of noise has powerful effects on behaviour, way of life and also on the physiological level. The reactions of the residential portuguese population to several kinds of noises will be taken as example.

Some portuguese case studies

Several studies have been directed towards the reaction of residential population to noise environment in general and to certain noise sources in particular. Results have usually been assessed in terms of numbers of people who report that they are highly annoyed or who are actively complaining.

Road traffic noise In 1972, the first portuguese study on the assessment of community annoyability due to road traffic noise, both in residences and schools was undertaken by Martins da Silva from the Civil Engineering National Laboratory (LNEC) (1). The noise emitted by road traffic was measured at the same places where questionnaires were administrated. The criteria established by the model considered a daytime exposure of 24 hours in residential areas and 16 hours in school areas. It was found out that for more than 71 dB(A) and a traffic density of more than 1500 vehicles per hour there was an incontestable annoyance at residential places, these limits being 67 dB(A) and 1000 vehicles per hour in case of schools. It can be estimated from final results that about 46% of Lisbon population considered road traffic noise as being highly or extremely annoying. Figure 1 shows the relations established between annoyability and audible sound intensity level, both for residences and schools.

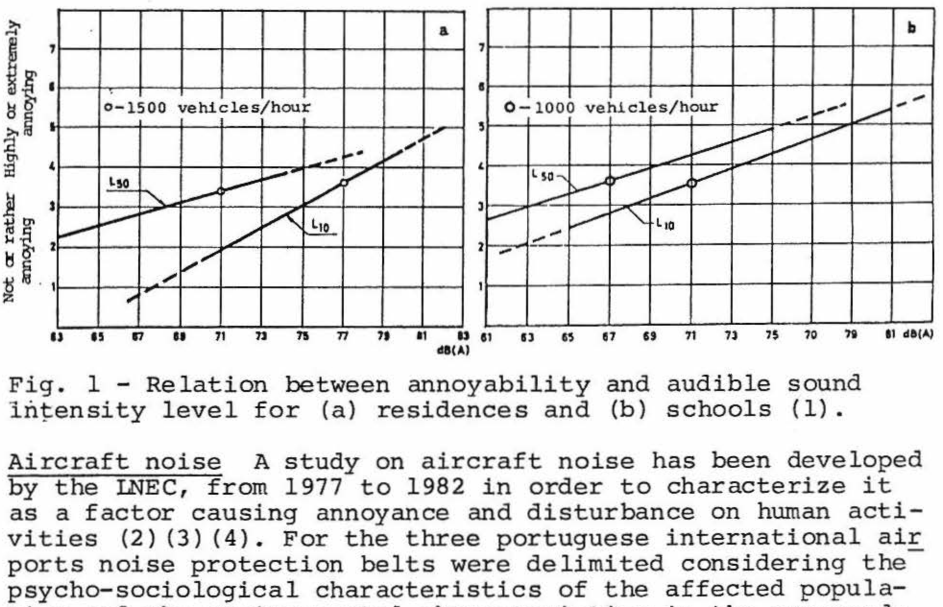

Fig. 1 - Relation between annoyability and audible sound intensity level for (a) residences and (b) schools (1).

Aircraft noise A study on aircraft noise has been developed by the LNEC, from 1977 to 1982 in order to characterize it as a factor causing annoyance and disturbance on human activities (2)(3)(4). For the three portuguese international airports noise protection belts were delimited considering the psycho-sociological characteristics of the affected population and the environmental characteristics in the surrounding area. Annoyability was considered incontestable within the noise protection belt. The search for the best index to assess portuguese population annoyability was also the purpose of this study. Questionnaires were administrated to the population living in the airports region. From the final results it can be extrapolated that almost 59% of the total Lisbon population considered in the study were annoyed by aircraft noise, being less than 15% the annoyed population on each of the other places (Faro and Porto) (5). Considering the high number of flights and the levels of urbanization at Lisbon, these results can be easily understood. Figure 2 shows the relation between the noise and number index (NNI) and the annoyability scale established for the three airports.

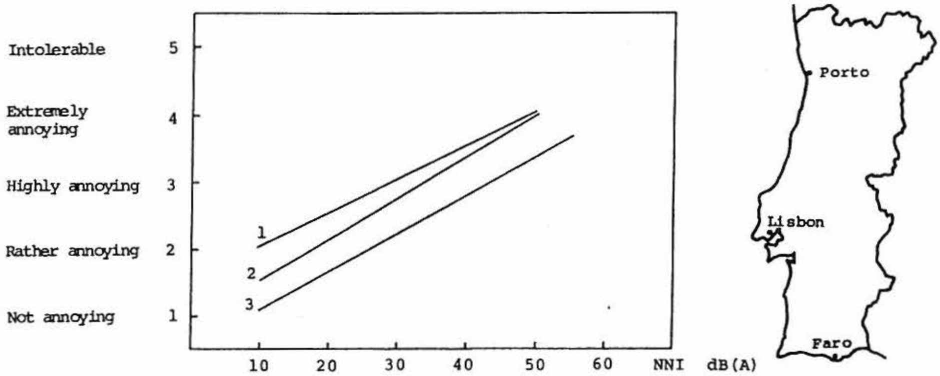

Fig. 2 - Relation between noise and number index (NNI) and annoyability for the three airports: 1 - Faro; 2 - Lisbon; 3 - Porto (2)(3)(4).

Industrial noise For each of the industrial administrative regions, another study has been undertaken in 1977 by the LNEC (6). The purpose was to analyse the public complaints to the administrative regions caused by industrial noise, from July to December 1977. Noise was quoted as the most important annoyance nuisance.

Community reaction to different types of noise During the airports study, a questionnaire was administrated to Faro's population in order to assess the types of noise that mostly affected the community (7). It is interesting to see that motocycles are the main cause of community annoyance, as it is shown in Figure 3.

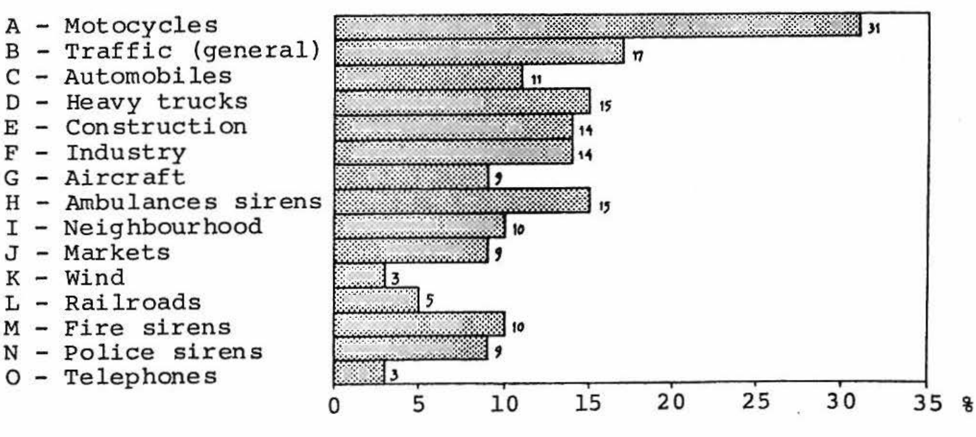

Fig. 3 - Types of noise mostly affecting Faro's population in 1979 (7).

The environment state department is receiving public complaints due to different kinds of pollution causes. Noise represents almost 50% of the total number of complaints received from March 1984 to April 1986. Table I lists the different types of noise and the number of times mentioned. It is interesting to see that traffic noise has been referred only three times, while noise from factories are the most mentioned.

TABLE I - COMPLAINTS RECEIVED AT THE ENVIRONMENT STATE DEPARTMENT (March 1984 - April 1986).

TYPES OF NOISE	NUMBER OF TIMES MENTIONED	%
Industry	47	31
Air-conditioners, exhaust fans	44	29
Discotheques, pubs	33	22
Other recreational activities	8	6
Church bells	5	3
Traffic (general)	3	2
Neighbourhood	3	2
Other	7	5
TOTAL	150	100

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PSYCHOPHYSIOLOGICAL EFFECTS OF NOISE

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The purpose of this project was to investigate the physiological non-auditory and psychological reactions to different types of noise (i.e. unwanted sound). We here report the results of two experiments.

Experiment I

In experiment I the physiological and psychological reactions to industrial noise (IN) and sounds specially selected for their annoying qualities (AS) were studied in two groups of 24 students. Three sound conditions (quiet(Q), homogeneous(HS), and inhomogeneous(IHS)) and two task conditions (card sorting and rest) were used. The 3 x 2 conditions were counterbalanced in a repeated measures design. The duration of each experimental condition was 2oo seconds. The mean sound level varied between llo and 93 dB(A).

Physiological reactions to noise were evaluated by measuring various cardiovascular, electrodermal, electromyographic, and EEG variables (the specific variables used in the two experiments may be seen in table 1 and table 2). A PDP-11 processor computed mean and SD of all physiological variables. Annoyance was evaluated with a questionnaire about the experience of sound exposure. Besides the subjects took two personality questionnaires (the Taylor Manifest Anxiety Scale and the Eysenck Personality Questionnaire).

The mean level of the physiological variables may be seen in table 1. Analysis of variance showed significant effects of noise on heart rate (HR), EMG frontalis (EMG(F)), and skin resistance level (SRL). Analysis of SD showed significant effects of noise on EMG(F) and EEG-beta. The noise effects were small and they did not depend on type of noise; the effects were comparable to the physiological effects of performing the simple card sorting test. No significant relations between questionnaire data and physiological reactions to noise were found.

Table 1. Means for groups and sound conditions

physiol. variables	group IN	AS	sound condition Q	HS	IHS
HR(bpm)	82.6	77.1	79.9	8o.3	79.4
EMG(F)(μV)	9.5	13.7	11.1	11.9	11.7
EEG-beta(μV)	6.4	5.9	5.9	6.5	6.8
EEG-alpha(μV)	6.8	5.8	6.3	6.3	6.3
SRL(KΩ)	48.4	84.o	67.5	65.5	65.5

First, we were quite surprised to find relatively small noise effects on the mean level of the physiological variables - both because of the intensity level of the sounds and also because the subjects reported that they found the sounds very annoying. However, discussing the experiment with our subjects we realized that the experimental sounds were not necessarily noise in the sense that they were unwanted by the subjects (in fact, the subjects were paid for listening to these sounds).

Experiment II

Experiment II was conducted with the specific aim of investigating the effect of knowing or not knowing that one is participating in noise research. Twenty-four subjects were told the true purpose of the experiment (the socalled "knowing" subjects) and their results were compared with 24 subjects, who believed they were participating in research on reading (the socalled "unknowing" subjects). The unknowing subjects were told that their understanding and retention of a difficult psychological text would be tested and that the primary purpose of the experiment was to compare the reading abilities of different groups of students (all subjects were non-psychology students).

The text used in the experiment was Freud's "Three essays on the theory of sexuality". This text was chosen because we expected that the students would at the same time find it both interesting and rather difficult to understand (thus distraction during reading should be particularly annoying). The text was presented to the subjects as 32 slides. Each slide was presented for 2 1/2 minutes, and during reading 13 of the 32 slides the subjects were disturbed by noise from the adjoining room. For each subject the noise was either present during the first or second half of the experiment (the order was counterbalanced across all subjects). There was no noise while the subjects were reading the first 6 slides because this might make the subjects suspicious about the purpose of the experiment. Thus, each half of the experiment proper consisted of 13 slides with a total duration of 32 1/2 minutes.

The noise in the adjoining room was produced by a tape recorder and simulated repair work. The intensity level variated, but the average level was about 55 dB(A) in the subject room. During reading the same physiological variables were registrated as in experiment I (a new variable was finger pulse volumen (FPV)).

Before the experiment the subjects answered the Eysenck Personality Questionnaire, and after the experiment they answered a questionnaire about their experience of the experiment and Weinstein's Noise Sensitivity Questionnaire. Before answering the latter questionnaires the unknowing subjects were interviewed by the experimenters to check whether the subject had realized the true purpose of the experiment (namely investigating reactions to unwanted sounds during reading). Subjects who expressed the slightest suspicion about the purpose of the experiment were replaced.

The differences between the means for noise and quiet may be seen in table 2. Repeated measures analysis of variance showed only significant effects on mean and SD for EMG-temporalis (EMG(T)). There was no significant difference between the physiological noise reactions of the knowing and the unknowing subjects.

Types of reactions

Both experiments show that noise induced mean differences are small in comparison with the size of individual differences in the level of the physiological variables existing when the subjects are not exposed to noise.

Therefore, an attempt has been made to analyze the individual differences systematically by looking at patterns of psychophysiological reactions. The variables used in this analysis have been the differences between the noise and the quiet condition of experiment II (the differences have been corrected for any sequence effects by regression analysis). These differences have been computed for all physiological variables and have been subjected to a type Q factor analysis. This analysis has identified two patterns of physiological reactions to noise. One pattern consists of a rise in finger pulse volumen and alpha level, and a fall in skin conductance level (SCL). The other pattern consists of a fall in alpha and theta levels, and a rise in skin conductance and heart rate.

Based on physiological reaction patterns it is possible to divide the 48 subjects into two reaction types. Differences in the means between noise and quiet for the two types may be seen in table 2. It turns out that the two reaction types score significantly different on the Neuroticism scale of the Eysenck Personality Questionnaire and on the Weinstein Noise sensitivity Scale. Furthermore, there was a significant difference between the types in electrodermal activity in a resting condition without noise.

Due to its post hoc nature the analysis of the physiological reaction patterns must be interpreted with great care.

Table 2. Differences in means between noise and quiet. EEG values are for the left(L) and right(R) hemisphere respectively.

physiol. var.	Type 1	Type 2	Total
HR(bpm)	1.o5	÷o.6o	o.42
FPV(μV)	o.o5	o.25	o.13
EMG(F)(μV)	o.37	o.16	o.29
EMG(T)(μV)	1.57	1.o4	1.36
EEG-beta(L)(μV)	÷o.o5	o.o2	÷o.o2
EEG-beta(R)(μV)	o.o9	÷o.17	÷o.o1
EEG-alpha(L)(μV)	÷o.25	o.29	÷o.o5
EEG-alpha(R)(μV)	÷o.28	o.31	÷o.o5
EEG-theta(L)(μV)	÷o.o6	o.o7	÷o.o1
EEG-theta(R)(μV)	÷o.o7	o.o5	÷o.o2
SCL(μmho)	o.56	÷o.99	÷o.o3

Conclusion

The size of the physiological noise reactions is comparable to the reactions in experiment I and as the intensity levels are quite different in the two experiments this may seem surprising. However, the physiological reactions in the two experiments may be caused by different mechanisms: In experiment I the primary factor may be intensity level and in experiment II it may be distraction. Experiment II does not convincingly demonstrate the importance of creating truly unwanted sounds, but it does demonstrate that distraction physiologically may be as disturbing as intensity level.

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ACTIVE NOISE CANCELLATION OF TRANSFORMER NOISE

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Introduction

Conventional passive measures against transformer noise, like enclosures or screening are often not feasible, and can be very expensive due to the size of a normal transformer. The use of active noise control is therefore an alternative approach, especially as this technique is very applicable to low frequency noise and pure tones.
The transformer noise is caused by vibrations of the core induced by magnetostrictive (alternating) forces. The laminated core is magnetised at twice the electrical frequency. The fundamental frequency of transformer noise is therefore 100 Hz (in Europe) and the total noise consists of harmonics of this frequency.
Measurements have shown that at a distance from the transformer, the dominating frequencies are 100 and 200 Hz. Higher frequencies are attenuated due to air and ground absorption. A typical frequency spectrum close to a 20 MVA transformer, compared to the the spectrum at a distance of 60 m, is shown in figure 1.

Several investigations have been carried out using active control of tranformer noise [1,2]. However, these projects have used idealized conditions, like having a small transformer in an anechoic room surrounded by anti-noise loudspeakers.
In a real situation, the noise propagating over a distance from a large, distributed source will be modulated by changes in wind and weather conditions. An active system based on the principle of cancellation near a residental area must consider these variations and be designed as an adaptive system. Our main objective with this project was

to investigate the characteristics of the weather on the noise and the performance of the adaptive system designed to cancel the noise.

The design of the active system The system consist of an anti-noise generating part and a measuring part. In the first part, the magnitude and phase of a 100 and 200 Hz signal are generated and fed to the anti-noise loud-speakers, which are placed close to the transformer.
A microphone is positioned at the point where cancellation is wanted. The microphone signal is fed back to the control system and band-pass filtered around 100 and 200 Hz, before registration in a microcomputer. A program has been written for the computer to minimize the noise signal from the microphone by variations of the magnitude and phase of the generated anti-noise signals. The microphone signal was also recorded on a tape recorder for further analysis. Figure 2 shows the principle set-up of the system. The system is described in more detail in [3].
To be able to investigate the accuracy in the detection of the resulting microphone signal (noise and anti-noise), three different bandwidths (2.5, 10 and 40 Hz) around 100 Hz and 200 Hz were chosen and the following number of averaging were used; 2,8 and 32. For all combinations of bandwidths and no. of averages, the cancelling effects due to variations in the generated tones were measured over a period of about 30 minutes (low no. of averages means high no. of datapoints in that period).

Results

The system was tested at two different transformer stations in Trondheim. The distance between transformer and microphone was 50 m at the first site and 80 m at the second.
A statistical analysis of the recorded residual microphone was performed and the following noise levels were calculated: L_{99} (typical background levels) L_{eq}-level (for the measuring period) and L_1 (peak levels).

A graphic plot of some of the results from this analysis is shown in figure 3 (200 Hz). The results are presented in an afternoon period (15.30-22.30) and a night period (23.00-06.00).
The horisontal axis is bandwidth and the vertical is number of averages. The graphic plot then illustrates the following:

lower left hand corner:	wide bandwidth, short measuring time and high number of data-points within a measuring period
higher left hand corner:	wide bandwidth, long measuring time and few datapoints.
lower right hand corner:	narrow bandwidth, short measuring time and many datapoints.

higher right hand corner: narrow bandwidth, long measuring time and few datapoints.

The dark shaded areas represent maximum cancellation achieved. Positive dB-values mean cancellation and negative enhancement of noise levels.

This figure indicates the complexity of outdoor active cancellation. None of the results are synonymous, i.e. none of the combinations of bandwidth and accuracy (no.of averages) are more consistent than the others. However, it seems that a small bandwidth (sharp filters) and high no. of averages are preferable to reduce low background levels. The results also show that the background level (L_{99}) filtered around 100 and 200 Hz are significant reduced (in the range 15-20 dB) and that peak levels (L_1) can be increased due to the anti-noise. The peak levels are not only caused by uncorrelated sources (like traffic) and are therefore difficult to cancel using this method. One of the problems concerning outdoor transformer noise is that the source is not uni-directional, not even at frequencies of 100 and 200 Hz. The directivity lobes can be very sharp and presumably easily modulated by the weather conditions. This probably also explains the inconsistency of our results. The directivity of the transformer noise is also illustrated in figure 4, where the sector of cancellation is plotted. This illustrates the problems of using a relatively small loudspeaker (one-channel system) trying to simulate a large, distributed source. As expected, a very limited zone of cancellation is achieved and the noise is increased to both sides of the microphone.

Conclusion

Even if the transformer noise consists of pure tones, the soundfield around a typical transformer is directional with sharp lobes that are easily modulated by weather conditions. This increases the complexity of a anti-noise system designed to cancel the noise at some distance from the tranformer.
L_{99}-levels are more easily reduced than L_1 (peak levels). A multi-channel system is needed if a wide cancellation zone shall be realised.

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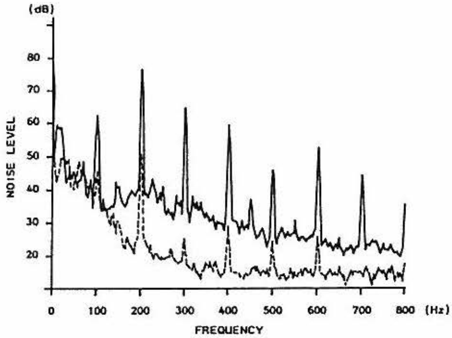


Figure 1. Transformer noise frequency spectra.
——: close to transformer.
-----: at a distance of 60 m.

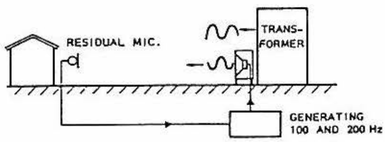


Figure 2. Principle set-up for active system.

Figure 3 Active cancellation of transformer noise, L_1 and L_{99}-levels.
Positive values = cancellation
Negative values = enhancement

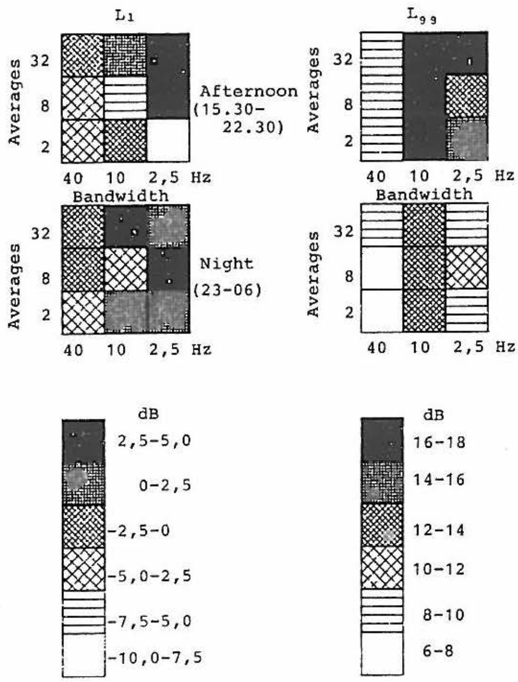
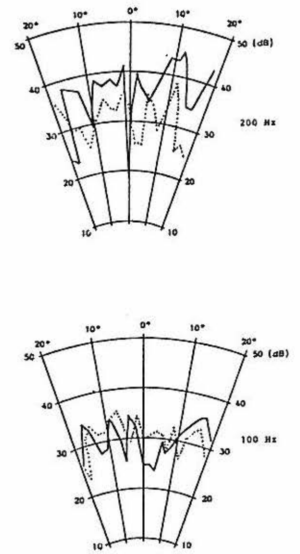


Figure 4 The cancelling zone at a distance of 60 m from the transformer.
——: active system on.
----: active system off.



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ACOUSTIC PERFORMANCE OF LOW- AND HIGH-IMPEDANCE TELEPHONES.

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INTRODUCTION:

The frequency response of an earphone is often measured by means of an "artificial ear". CCITT has adopted the artificial ear described in IEC Rec. 318, see [1] & [2]. The measurements are made with the earphones acoustically sealed to the IEC 318-ear to provide handy methods of measurement and reproducible readings. Under normal use of a telephone handset there is an acoustic leak which depends on the static force (against the ear) and the geometrical design of the telephone handset.

The difference between artificial and real ears is described by a correction factor L_e (Earphone coupling loss factor). Determination of L_e requires sound pressure measurements in the auditory canal of many people with controlled static force of the earphone against the ear. This is difficult and time-consuming. Therefore CCITT suggests a "typical" L_e correction curve which has been generally adopted by telephone manufacturers.

For a conventional handset with a high-impedance earphone the difference between this "typical" L_e correction curve and the real curve is in the order of 10-15 dB at low frequencies (static force about 300 grammes). This is very unsatisfactory for the user.

In the following it is shown theoretically and verified by measurements that these problems can be overcome by designing a low-impedance telephone handset.

A: Model for an earphone coupled to the ear.

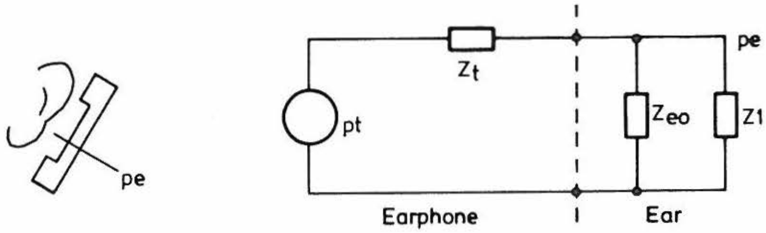

fig. 1

p_t	Thévenin pressure generator (blocked pressure)
p_e	Pressure in front of the earphone
Z_t	Thévenin generator acoustic impedance
Z_{eo}	Acoustic impedance of the ear (sealed condition)
Z_1	Acoustic leak-impedance (dependent on static force)

Generally it applies that the coupling between a telephone handset and the human ear can be analyzed using Thévenin's theorem and an electric (impedance) analog circuit. See fig. 1.

$$H_e = p_e/p_t = Z_{eo}\ Z_1/(Z_t+Z_{eo}\ Z_1) \qquad \text{e.1}$$

1: High-impedance earphone.

H_e depends on $Z_{eo}\ Z_1$ which varies significantly (through Z_1) with the static force against the ear.

2: Low-impedance earphone.

If $Z_t << Z_{eo}\ Z_1$ then:

$H_e \cong 1$

In this case the pressure response is independent of earimpedance and leak-impedance. This is a very desirable situation.

B: L_e - Theoretical consideration.

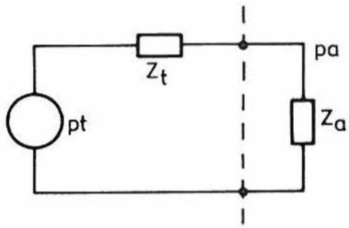

fig. 2

Z_a Acoustic impedance of artificial ear, see [2].

p_a Pressure measured with the microphone (B&K 4134) mounted in the artificial ear (B&K 4153).

$$H_a = p_a/p_t = Z_a/(Z_a+Z_t) \qquad \text{e.2}$$

By definition, and using e.1 and e.2, we get:

$$L_e = p_a/p_e = (Z_t\ Z_a)/(Z_t\ Z_{eo}\ Z_1) \qquad \text{e.3}$$

If $Z_a \cong Z_{eo}$ as it should be for a quality artificial ear, we get:

$$L_e \cong 1 + (Z_t\ Z_a)/Z_1 \qquad \text{e.4}$$

1: High-impedance earphone

Condition: $Z_t >> Z_a$

$$L_e \cong 1 + Z_a/Z_1 \qquad \text{e.5}$$

2: Low-impedance earphone.

a: Ideal condition: $Z_t << Z_a\ Z_1$

$$L_e \cong 1 \qquad \text{e.6}$$

In this case the real-ear response is equal to the artificial-ear response and independent of Z_1.

b: Non ideal condition.

L_e is given by e.4.

The difference between high- and low-impedance design is illustrated in fig. 3.

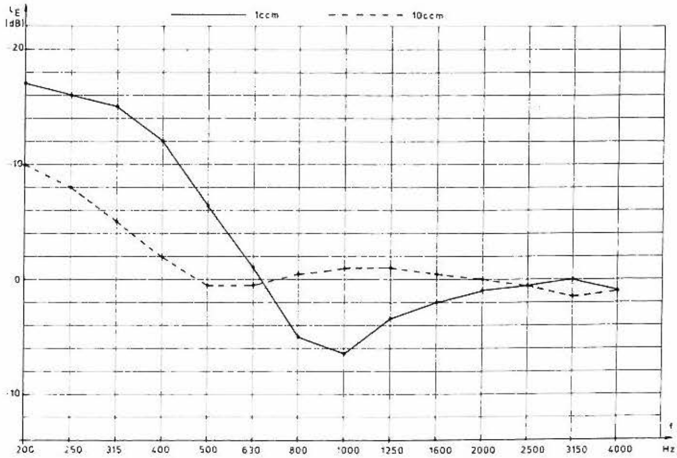

fig. 3

The calculations shown in fig. 3 are based on (scale factor 10E-5):

Z_e = 15 ohms + s 10mH ("normal" use of a handset)

Z_t = 26 ohms + s 5mH + $1/sC_t$

Z_a = impedance of IEC 318-ear, see [2].

Upper curve, C_t corresponding to 1 ccm.

Lower curve, C_t corresponding to 10 ccm.

The figure shows clearly that the "low-impedance" handset produces considerably lower L_e values at low frequencies.

Further simulations show that the given structure of Z_t cannot absorb the full range of Z_1 variations occurring in practice.

This problem can be overcome by introducing a small slit from the earphone front-cavity to the free. A more detailed description will be published later.

C: L_e measurements.

a: General.
L_e is measured at the 14 standardized 1/3 octave frequencies in the range 200-4000 Hz. The sound pressure level should be close to the level corresponding to "normal" use of a telephone set. Static force of the handset against the ear is controlled by means of a movable arm.

b: Measurements on artificial ear (IEC 318).
These measurements are performed with the earphone sealed to the 318-ear. The sound pressure is measured with the built-in microphone (B&K 4134).

c: Measurements on real ears.
The subjects must have normal hearing and be free of infections (eustachian tube in normal condition). In accordance with CCITT's recommendations the microphone position selected is close to ERP (ear reference point). A new B&K probe-microphone based on a 1/4 inch condenser-microphone (4135) is used for the auditory canal measurements.

d: Instrumentation.
The measurements have been automated using a personal computer to control the B&K Measuring Amplifier (2636) and the HP Audio Analyzer (8903). A standard Feeding Complex (2 µF, 400 ohms and 2 Henry) and a 48 V Power Supply are used to "power" the telephone sets. Generator voltage and impedance are 251 mV and 600 ohms.

e: Measurement procedure.

The subject is placed on a chair and the movable arm is adjusted until the handset is placed close to the "normal" position. It is important that the subjects are placed comfortably to be able to hold the head still.

f: Results.

Fig. 4 shows L_e for a "low-impedance" handset and fig. 5 shows L_e for a traditional "high impedance" handset. The measurements represent the average response for 5 subjects.

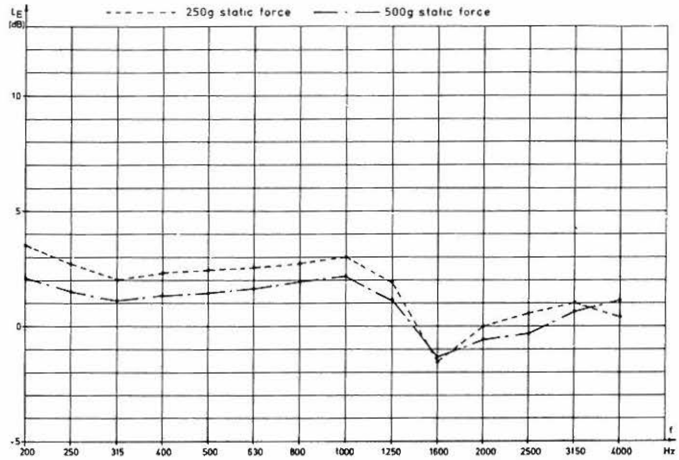

fig. 4 L_e for a "low-impedance" handset

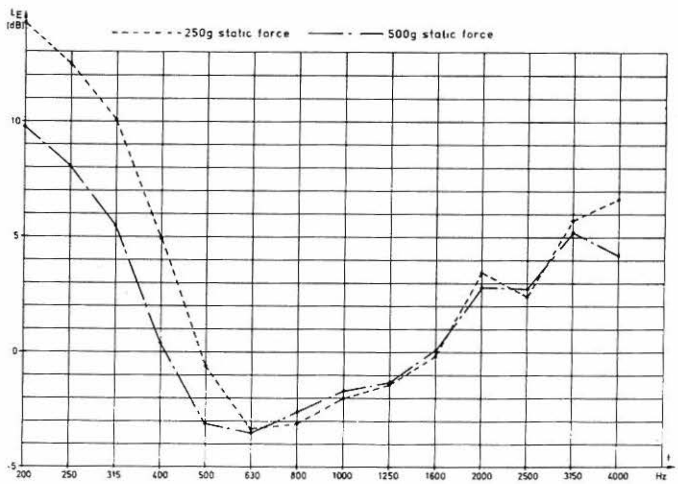

fig. 5 L_e for a traditional handset

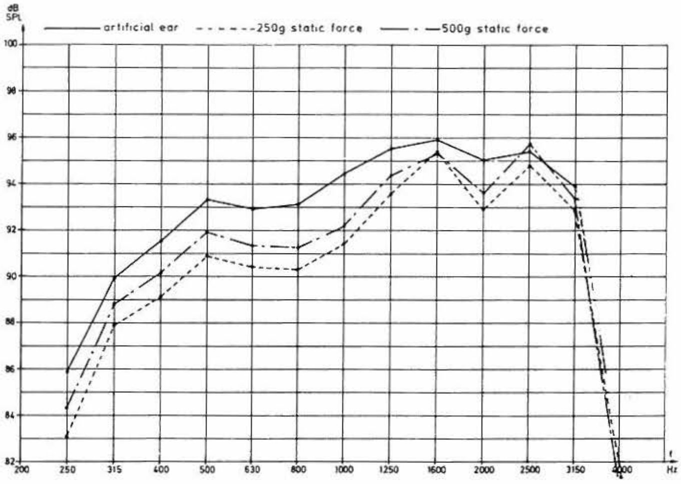

fig. 6 Frequency response of a "low-impedance" handset

For the traditional handset, we observe (by 250 grammes) a variation from +14.6 to -3.3 dB (total 17.3 dB) in L_e. For the new "low-impedance" handset, the variation (by 250 grammes) in L_e is seen to be from +3.5 to -1.6 dB (total 5.1 dB).

Thus it is evident from the measurements that the new "low-impedance" design offers a much better suppression of the ear-coupling-induced "gain-error".
The above shows that a flat frequency response measured in the standardized procedure, sealed coupling to IEC 318 artificial ear, results in a flat response on real ears only when the acoustic impedance of the earphone is sufficiently low.

D: Conclusion.

A simple theoretical model can explain the main difference between pressure response measured by means of IEC artificial ear and real ears. Measurements of L_e for a traditional "high-impedance" handset are compared with those of a newly designed "low-impedance" handset. The results show clearly the much better reduction of the gain-variations introduced through the acoustic leak, which is strongly dependent on the application force of the handset against the ear.

Ref. 1 IEC R 318
2 B&K Technical Review No. 4 1961 & No. 1 1962.

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ON THE MEASUREMENT OF THE INSERTION GAIN OF AUDIO COMMUNICATION SYSTEMS USING HEAD AND TORSO SIMULATORS

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INTRODUCTION

In his book "Telecommunication by Speech" (1973) [1], D.L. Richards of British Telecom points out that an appreciable quality interval exists between even the best ordinary telephone connection and a speech link set up to simulate direct air-to-air conditions, by using the best quality microphones and earphones. We recently examined this issue by using a high quality Head and Torso Simulator (HATS) featuring speech and hearing.

1. ACOUSTIC REFERENCE

The geometry and the associated acoustic variables of the reference situation are shown in Fig. 1. A Talker Reference (TR) point and a Listener Reference (LR) point are spaced horizontally 1 m apart. A talker is placed (centre of head) at TR and a listener at LR. The talker is energized (V_M) and produces a sound pressure, P_{MRT}, at the Mouth Reference point of the Talker. With the listener absent a pressure, P_{LR}, results at the LR. With the listener present a sound pressure, P_{DRL}, results at the ear Drum Reference point of the Listener, [2, 3].

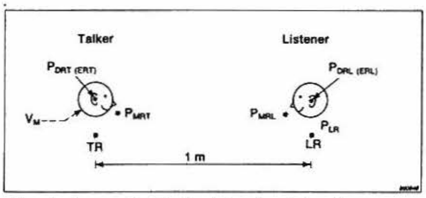

Fig. 1. Acoustic (Orthotelephonic) reference

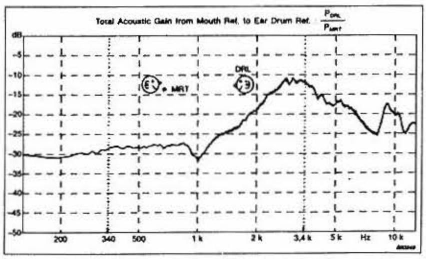

Fig. 2. Total acoustic gain for talker and listener – Orthotelephonic reference

Figure 2 shows the total acoustic gain from P_{MRT} to P_{DRL}. The result was obtained using a B & K Dual Channel FFT Analyzer, Type 2032, in time selective mode, thus simulating anechoic conditions. Substantial gain above the $^1/_r$ related – 30 dB is seen for the speech frequencies. This is important for intelligibility. The maximal relative gain, about 18 dB, is seen around 3,4 kHz, coinciding with the upper frequency limit of the transmitted telecom voice band (dotted line). The design objective for a speech link is reproduction of the major features of this acoustic gain curve, the orthotelephonic reference, within the transmitted voice band [1].

The orthotelephonic reference, P_{DRL}/P_{MRT}, consists of two separable components (Fig. 3). The first:

(a) represents the acoustic gain of the talker with the listener absent: P_{LR}/P_{MRT}.

The second:

(b) represents the acoustic gain of the listener: head diffraction, ear canal transfer etc: P_{DRL}/P_{LR} when the listener is placed in the talkers voice field.

The essential features of these curves have been validated against human acoustic reference data (4 and elsewhere) and are known to properly represent the median human acoustic characteristics.

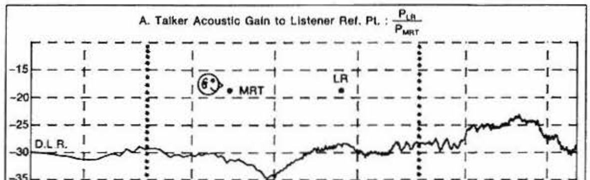

Fig. 3. Talker and listener components of total acoustic gain

2. APPLIED AUDIOCOMMUNICATION (TELEPHONE) LINK

Next the talker and the listener are placed in acoustically separate locations and a telephone link is established between them (Fig. 4). The talker is energized to a known P_{MRT} (with the microphone absent), and as a result a Transmitted Voltage, V_T, is sent to the line by the talker's telephone set. Further the Voltage Received from the line, V_R, is used by the listeners telephone

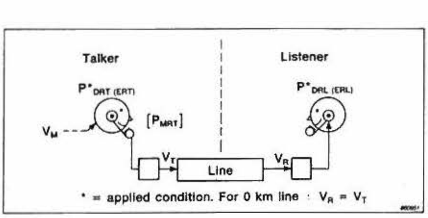

Fig. 4. Applied audiocommunication system (telephone link)

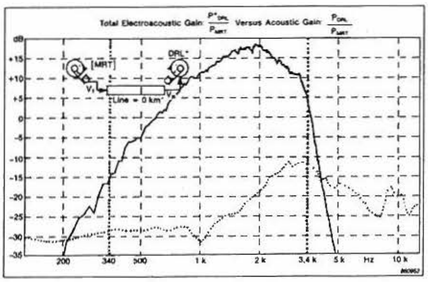

Fig. 5. Total electroacoustic gain: —— with applied (*) telephone link versus total acoustic gain: ·······. Line length: 0 km i.e. $V_R = V_T$ (in 600 Ω and with standard feed bridge)

set to energize his earphone, producing a sound pressure, P^*_{DRL}, at the ear Drum Reference point.

The total electrocacoustic gain, P^*_{DRL} / P_{MRT}, for one of the better modern telephone types is shown in Fig. 5, full curve. The measurements were done with standardized circuitry and excitation levels while using a B & K Type 2032 FFT Analyzer and pseudorandom excitation signals. Steps were taken to secure a representative low frequency leak on the HATS. The validity of the pseudorandom approach was verified by cross checking with swept sine measurements using a B & K Type 9573 system. Zero km line length and no junction attenuation was used. For a typical telephone connection 25 dB attenuation may be added.

Except for a level difference, the electroacoustic gain curve should track the design objective, the acoustic gain curve (shown dotted) within the transmitted voice band. The ratio of the electroacoustic (in situ) gain to the acoustic (reference) gain is by definition the INSERTION GAIN alias the orthotelephonically referred gain of the telephone link, i.e.:

$$\text{Insertion Gain: } \frac{P^*_{DRL}}{P_{DRL}} = \frac{P^*_{DRL}}{P_{MRT}} \Big/ \frac{P_{DRL}}{P_{MRT}}$$

and appears on Fig. 6.

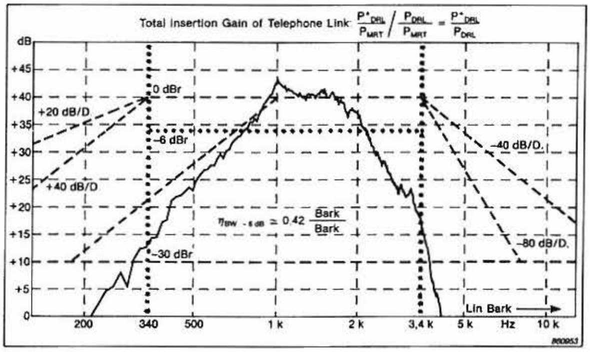

Fig. 6. Total INSERTION GAIN of telephone link, i.e. total electroacoustic gain divided by total acoustic gain. Efficiency of utilization of the available 340 to 3400 Hz bandwidth is about 0,4 when measured in Barks to the –6 dB points of the INSERTION GAIN

In frequency bands where the electroacoustic gain tracks the acoustic reference gain curve, the Insertion Gain curve exhibits no frequency distortion. This is approximately so from 1 to 2 kHz.

Translation of the design objective for a telecom speech link into Insertion Gain terms yields:

i) no significant frequency distortion in the transmitted voice band, i.e. between 340 to 3400 Hz (0 dBr, dotted horizontal line)
ii) an in band gain which is close to 0 dB using representative junction and line losses (analog nets)
iii) a controlled roll-off outside the transmitted voice band, possibly steepest towards the higher frequencies (examples are shown by dotted lines).

The actual efficiency of utilization of the available 340 to 3400 Hz band can now be computed for our telephone link (–6 dB points and in Bark/Bark):

$\eta_{BW,\ 6\,dB} \simeq 0{,}4$

i.e. **less than half** of the available bandwidth is actually used!

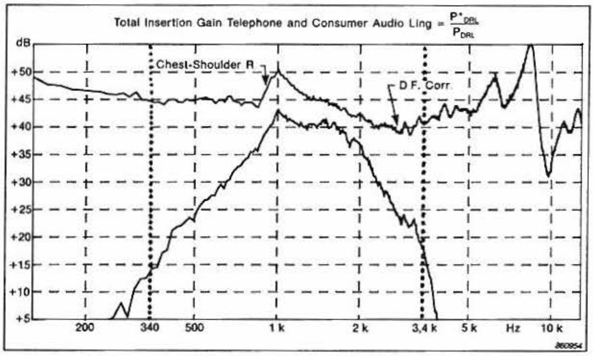

Fig. 7. Total INSERTION GAIN of telephone link compared to a link using consumer audio transducers. Earphone was DIFFUSE-FIELD, not Free-Field corrected. No compensation for chest-shoulder reflections was incorporated (around 1 kHz, see Fig. 5)

In Fig. 7, the insertion gain of our telephone link is compared to a link using readily available consumer audio components: microphone, amplifier and headphone (upper curve). This link has an extremely wide frequency response. The moderate frequency distortion in the telecom voice band is due to:

i) uncompensated chest-shoulder reflections (around 1 kHz, see Fig. 5),

ii) **diffuse field**, not free-field correction of contemporary headphones.

Used properly present consumer audio technology is capable of 100% telecom voice band utilization.

CONCLUSIONS:

- An appreciable "performance gap" is (still) present between telephone links and consumer audio links – it may even have widened since 1973.
- Major achievements in consumer audio transducer technology have not yet reached the telecom field.
- In the light of coming ISDN audio channels, the limitations of the present telecom audio transducers will become even more apparent.
- There are means to significantly narrow the present performance gap even within the transmitted voice band and thus allow telecom audio terminals to enter into the "high tech" era.

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Udviklingstendenser inden for elektronisk rumakustik

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INTRODUKTION

I den sidste halve snes år er der sket en betydelig udvikling inden for den gren af elektroakustikken, der beskæftiger sig med forbedring af lytteforholdene i store og akustisk vanskelige rum. Til akustisk vanskelige rum regnes i denne forbindelse også flerformåls-sale, hvor de forskellige anvendelser stiller hver deres akustiske krav til salen.

GRUNDLÆGGENDE AKUSTISKE KRAV

Lad os slå fast endnu engang, hvilke krav der må stilles til et rum, for at det kan kaldes et akustisk godt rum:

1. Passende efterklangstid
2. Jævn lydfordeling
3. Passende lydstyrke
4. Passende lavt baggrundsstøjniveau
5. Ingen ekko/flutterekko
6. Passende fordeling (både rumligt og tidsligt) af tidlige refleksioner

Disse krav gælder, hvad enten der er tale om et rent akustisk rum eller et rum, hvor der anvendes elektro-akustisk forstærkning.

Ved multipurpose-halls gælder desuden, at krav nr. 1 og 6 vil afhænge af rummets aktuelle anvendelse.

Krav 6 er det sidst tilkommende krav, som dækker over, at den subjektive oplevelse af akustikken i et rum er stærkt sammenknyttet med, hvor kraftige de tidlige refleksioner er, og hvorfra de kommer. Forenklet kan det siges, at tidlige, kraftige refleksioner fra loftet og fra en evt. reflektor over scenen giver en distinkt, veldefineret, "tør" lytteoplevelse, medens dominerende refleksioner fra sidevæggene giver en "rumlig" lytteoplevelse, tilhørerne føler sig omgivet af lyd.

Et rum med kraftige, tidlige loftrefleksioner vil således fremme taleforståeligheden (forudsat korrekt efterklangstid), medens et rum med kraftige, tidlige siderefleksioner vil være gunstigt for den musikalske oplevelse.

Erkendelsen af disse forhold stammer fra uheldige koncertsale, teatersale og flerformåls-sale, bygget inden for de sidste 30 år. Nye arkitektoniske udformninger, fx vifteformen, gav en svækkelse af siderefleksionerne, og til trods for en korrekt efterklangstid, fik man en utilfredsstillende musikalsk oplevelse. En analyse af den klassiske "shoebox" koncertsal viste omvendt, at denne netop udmærkede sig ved kraftige siderefleksioner.

Der er således store akustiske vanskeligheder ved at realisere en virkelig multipurpose-hall, men det er forfatterens opfattelse, at det elektro-akustiske udstyr, der er til rådighed idag, giver nye muligheder inden for dette område.

De traditionelle bygningsakustiske løsninger giver muligheder for at variere efterklangstiden (krav 1) inden for ret beskedne rammer (typisk $\pm$ 15%), og de er ret kostbare og besværlige at anvende i praksis. Teoretisk set er det også muligt at variere krav 6 ad denne vej, men det komplicerer konstruktionerne så meget, at det så vidt vides ikke er forsøgt.

ELEKTRO-AKUSTISKE LØSNINGSMULIGHEDER

Udviklingen af hornhøjttalerne

De klassiske hornhøjttalerkonstruktioner går, hvad angår den teoretiske behandling, tilbage til 1920'erne. En karakteristisk svaghed ved disse konstruktioner er de frekvensafhængige retningsegenskaber.

Sådanne horn, der dækker mellem- og højtoneområdet, karakteriseres som oftest ved den rumvinkel, inden for hvilken lydtrykniveauet ikke varierer mere end 6 dB, fx 90° x 40°. Det skal bemærkes, at hornet kun opfylder dette inden for et begrænset frekvensområde, ved lavere frekvenser er rumvinklen større og ved højere frekvenser er rumvinklen mindre. Som følge heraf vil diskantområdet således udstråles i et snævert bundt omkring aksen, medens de lavere frekvenser spredes ukontrolleret ud i rummet.

Forskellige hornudformninger, fx multicellehornet, har med større eller mindre held forsøgt at modvirke denne skavank.

I begyndelsen af 1970'erne dukkede nye hornudformninger op, som nu under ét betegnes "constant-directivity"-horn. Karakteristisk for dem er de stærkt forbedrede retningsegenskaber, som opnås gennem en ændret, ofte "knækket" hornkurve.

Samtidig er der fremkommet forbedrede metoder til beregning af den lyddækning i en sal, der kan opnås med et givet højttalersystem. Metoderne er baseret på rumlige retningskarakteristikker for hornhøjttalerne og på en rumvinkel-baseret afbildningsform af tilskuerområdet, set fra højttalerpositionen. Vore erfaringer med disse systemer viser, at der kan opnås en lyddækning på tilskuerområdet, som er ensartet inden for typisk 5 dB.

Elektronisk forlængelse af efterklangstiden

Det første vigtige anlæg til elektronisk forlængelse af efterklangstiden i en sal var Parkin og Morgans anlæg i Royal Festival Hall i London, som blev installeret i årene 1964-69. Anlægget fik navnet "assisted resonance" og bestod af et stort antal kanaler, hver bestående af en mikrofon, en forstærker og en højttaler. Mikrofonen er anbragt i en lille Helmholtz-resonator, således at kun et smalt frekvensområde forstærkes. Den efterklangsforlængende virkning består, i at hver kanal i systemet arbejder på grænsen af ustabilitet, således at anlægget er på nippet til at gå i sving. Systemet er kostbart og besværligt at indregulere, men der er alligevel installeret sådanne anlæg i en række andre sale.

I slutningen af 1960'erne fremkom Philips med et andet system også bestående af et antal mikrofon-forstærker-højttaler-kanaler. I dette system er kanalerne dog bredbåndede, og mikrofonerne opsamler lyden i rummets efterklangsfelt. Også her kræves et stort antal kanaler (50-100) og en nøje indregulering af systemet. Der kan opnås en forøgelse af efterklangstiden på op til ca. 50%. Også dette system er blevet anvendt i en række sale i bl.a. Holland, Norge og Vesttyskland.

Fremkomsten af digitale efterklangsmaskiner i slutningen af 1970'erne har åbnet nye muligheder for elektronisk forlængelse af efterklangstiden. Schröeder har i 1960'erne opstillet algoritmer for, hvorledes en "naturlig" efterklang kan genskabes af et digitalt signalbehandlingssystem, og udviklingen af microprocessorerne har gjort, at sådanne systemer idag findes tilgængelige som færdige apparater.

I princippet fungerer de som en række digitale tidsforsinkelsesmoduler, hvorigennem mikrofonsignalet passerer. En del af de forsinkede signaler ledes tilbage i systemet, tidsforsinkes igen osv. før det endelige signal aftappes på udgangen. I de mest avancerede af disse maskiner kan vi kontrollere de 40 første refleksioner med hensyn til niveau og tid ved hjælp af en personal computer. Derved kan man i høj grad efterligne efterklangsforholdene i et virkeligt rum. Systemet kræver - sammenlignet med de førnævnte - kun relativt få mikrofoner og kanaler og er betydeligt enklere at indregulere. Desuden kan efterklangstiden varieres over et væsentligt større interval.

Elektronisk rumakustik og flerformåls-sale

Det er herefter naturligt at tænke på disse systemer ved løsning af de akustiske problemer i flerformålssale. Af økonomiske grunde skal de fleste større sale, der bygges idag, anvendes til mange forskellige formål: Teater, shows, symfonikoncerter osv. Hver for sig kræver de specielle akustiske forhold: Forskellige efterklangstider, forskellige efterklangskurver og forskellige fordelinger af laterale og mediale refleksioner.

Rummets egen, naturlige akustik indrettes efter den funktion, der kræver den korteste efterklangstid. Akustikken til de øvrige funktioner skabes med elektroniske midler, dvs. digitale klangmaskiner, hvor den aktuelle efterklangskurve forefindes i form af et computerprogram.

Ved hjælp af programmerbare, digitale tidsforsinkelsesmaskiner og et antal højttalerkanaler kan fordelingen af laterale contra mediale refleksioner styres uafhængigt af efterklangen. Herved kan lytteindtrykket ændres, således at rummet kan virke enten distinkt til fremme af taleforståeligheden eller rumligt til fordel for den musikalske oplevelse.

De mange parametre, man med systemet får kontrol over, vil styringsmæssigt kunne samles i computerprogrammerne, og opdatering baseret på indhøstede erfaringer fra brugen af systemet kan herefter løbende ske gennem programændringer.

Med disse systemer er der skabt helt nye muligheder for fleksible og økonomiske akustiske løsninger, specielt for flerformåls-salene.

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GAIN IN HEARING AIDS

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The gain in a hearing aid and the frequency response, which is the gain as a function of frequency, are ambiguous concepts as several definitions exist due to the difference in measuring conditions. The reason for the existence of several methods of measurement of the gain is the different purposes that the measurements serve.

In general, the acoustic gain of a hearing aid is defined as the ratio of the sound pressure, developed in a specified acoustic load i.e. an artificial ear or a coupler, to the input sound pressure under specified acoustic conditions. In practical use there are four different definitions of the acoustic gain viz. the free field gain, the pressure gain, the insertion gain, and the simulated insertion gain.

Free Field Gain.

The definition is given by IEC [6]. The purpose of this publication is to prescribe reproducible methods for the determination of certain electro-acoustic performance characteristics of hearing aids e.g. in connection with type approvals.

The definition of the free field gain is illustrated in fig. 1. The hearing aid is placed in a calibrated, plain progressive sound field in an anechoic room without any baffle or other device, simulating the head or body of a wearer. The hearing aid receiver is loaded with an ear simulator, IEC [5]. The gain is defined as the ratio of the sound pressure in the ear simulator to the sound pressure in the undisturbed free field.

Pressure Gain.

This is defined by IEC [7]. This publication describes simplified methods of measurements for delivery control purposes. The receiver is loaded with a 2 cm^3 coupler, IEC [4]. The input sound pressure is measured with a microphone close to the microphone of the hearing aid as shown in fig. 2. The pressure gain is thus defined as the ratio of the sound pressure in the coupler to the sound pressure at the hearing aid microphone.

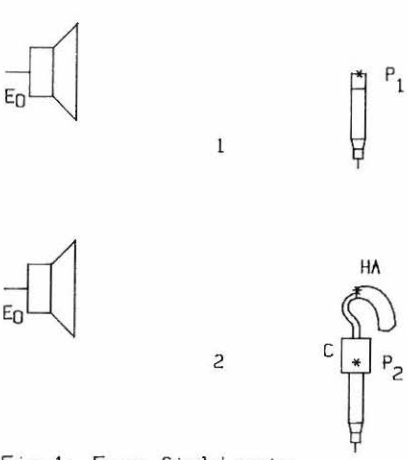

Fig. 1: Free field gain

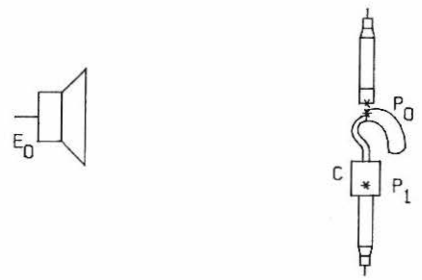

Fig 2. Pressure gain

Insertion Gain.

Due to the influence of the acoustic properties of the wearer the free field gain or the pressure gain do not portray the performance characteristics under actual conditions. This has led to the introduction of the concept of insertion gain. Dalsgaard & Jensen [3]. This definition is illustrated in fig. 3.

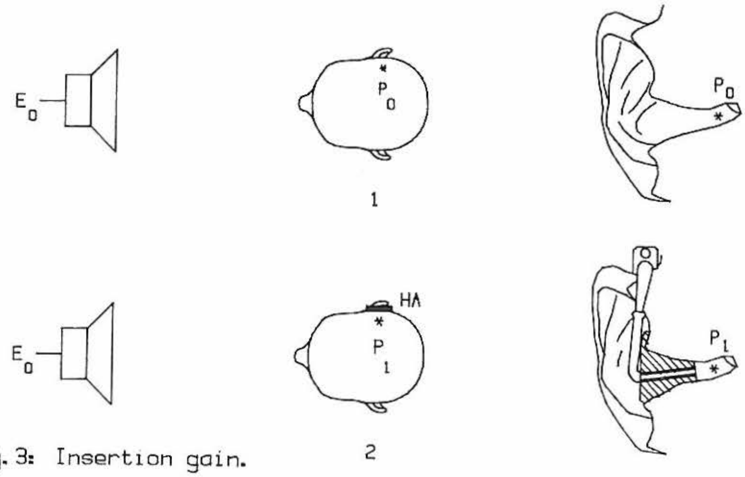

Fig. 3: Insertion gain.

The insertion gain is defined as the ratio of the sound pressure at a specific point in the ear canal of a person using a hearing aid to the sound pressure at the same point without the hearing aid and earmold. The insertion gain will be unique for the person in question.

Simulated Insertion Gain.

In some cases e.g. for research and development purposes it is desirable to eliminate the individual variations in the insertion gain. This has led to the development of head and torso simulators (HATS) such as the KEMAR [2] , which is an anthropometric average of a large population. The KEMAR is equipped with the above-mentioned occluded ear simulator. The innsertion gain of a hearing aid measured on a HATS is the simulated insertion gain [8].

Functional Gain.

The insertion gain does not necessarily correspond to the gain experienced by the patient (Ayers [1]). This has led to the introduction of the concept functional gain by Pascoe [10].

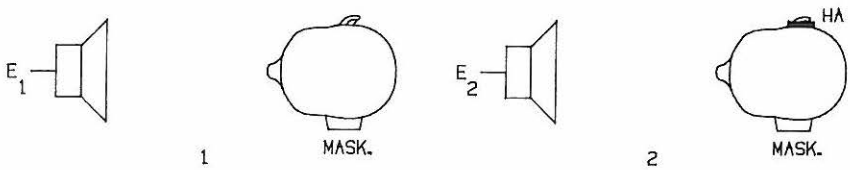

Fig. 4: Functional gain.

The functional gain is based upon psycho-acoustical measurements either by determining the change in the threshold of hearing with and without a hearing aid, or by a loudness balance test using a reference sound signal in the contralateral ear as sketched in fig. 4.

The receiver on the contralateral ear delivers either a masking signal (for threshold measurements) or the reference signal (in loudness balance tests).

Conclusion.

The different definitions of the gain will of course lead to different values of the gain, which should be born in mind.

Between the free field gain and the pressure gain will be a difference due to the absence of diffraction effects by the latter. The insertion gain will again be different from these two gains due to diffractional phenomena around the head and the body.

Measurements by Lyregaard [9] show furthermore that the functional gain and the insertion gain are not identical. More interesting is the fact that there seems to be a difference in the functional gain depending on whether is is determined by threshold or by loudness measurements.

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HEARING AIDS AND THE DAMPING EFFECTS OF IMPULSE NOISE.

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When you are on audiological touring in Greenland it is striking to ascertain that 25 per cent of all the hearing impairments are due to noise trauma, mainly caused by impulse noise from gun shooting.

It is quite an impossible task to persuade the hunters to wear ear protectors. On the other hand many of the rehabilitated hearing impaired people tell you that they wear their hearing aids when they go hunting, and more astonishing is the fact that they are very happy about doing so.

If you consider the last statement carefully it is obvious that the hunters of Greenland are right. Fitted with a closed earmold and a switched on hearing aid the hearing impaired hunter will at the same time be able to hear the different sounds from the nature surrounding him and be protected from the injurious impulse noise produced from his gun because of the damping effect caused by the power limitation of his hearing aid.

Therefore we will try to illustrate some of the acoustic relations which occur in the ear canal when a shot is fired and the ear is open and unprotected, and also what happens when the ear is protected by a closed earmold. Further on is shown which protection of the ear you can achieve by using a hearing aid.

All the measurements are carried out with a toy pistol (dog pistol) in an anechoic chamber to imitate the free field. The time amplitude which is developed from the noise of the shot is on the whole identical with the noise from the hunter's gun when you ignore the fact that the amplitude is 30 to 40 dB less dependent on the type of gun [1].

The time amplitude pattern of a shot from the pistol is shown on fig. 1. It can be seen that the course is nearly single-phased. The curve results from a measurement in free field condition and is measured with a quarter inch condenser microphone (Brüel & Kjær type 4135) which has been set up 50 cm lateral to the pistol.

From an energy point of view [2] the most possible estimate of the effective impulse time is 30 µs and the amplitude is found to 123 dB re 20µPa in Imp Hold mode (B & K 2607). The matching frequency analysis from the signal analyzer (B & K type 2033) redrawn to octave values is shown on fig. 2 which shows the most powerful energy radiation in the range 1 to 4 kHz.

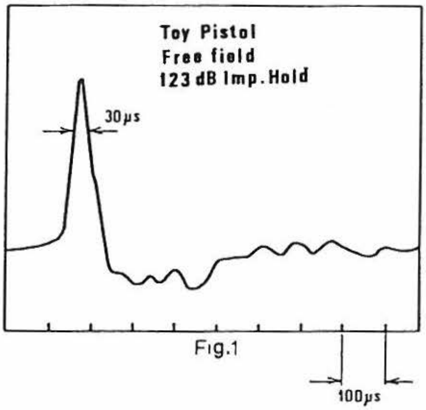

Fig.1

What happens in the ear canal when the ear is exposed to a shot?

To prevent a person from an injurious hearing impairment during the experiment measurements have been made on KEMAR with the artificial ear B & K type 4157. The muzzle of the pistol was placed on the axis of the ears 50 cm lateral to the right ear of KEMAR and with the barrel directed opposite the nose and perpendicular to the axis.

Results

Open ear canal. Compared to the free field spectrogram in fig. 2 a strong amplification of the sound pressure is found in the open ear canal which shows up to 15 dB within the frequency range of 2 to 4 kHz. The same phenomenon was earlier found by Wiener and Ross in 1946 [3] and is due to a powerful resonator in the ear canal.

The time function in fig. 3 has changed to a markedly two-phased course, and the sound pressure level Imp Hold measurement has changed to a 14 dB higher level which is in full agreement with the change of the spectrogram.

Unfortunately the frequency range of the most predominant sound pressure levels is the same as for the resonance effect with the result of an easier impairment of the unprotected ear.

Closed ear canal. Blocking the ear canal is the best way to avoid damage caused by noise. With a closed earmold we still see a very small rest of the initial two phasic courses in fig. 3, but the powerful peak has disappeared due to the fact that a considerable suppression takes place which reaches an attenuation of 30 dB in relation to the open canal. This is partly due to suppression in the material of the earmold and partly due to changing of the physical conditions of the resonator in the ear canal.

This is the reason why most of the resonance frequencies now are out of the range of usable hearing and can be seen from the lowest spectrogram in fig. 2 for the treble. The bass frequencies obviously have a less attenuation caused by a leak in the earmold fitting.

The above-mentioned situation could also be carried out by using a closed mold fitted with a hearing aid switched off. - But what happens with a hearing aid switched on?

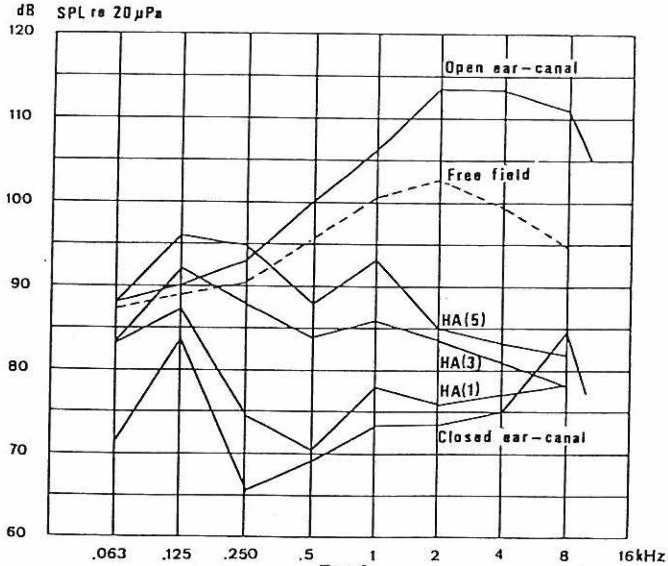

Fig.2

Hearing aid, switched on. We have used a general hearing aid without a compression system (Danavox 115V AGCO), fitted with a closed earmold. Gain setting 6 (33 dB) at all three different levels of the maximum output (MPO). In fig. 2 corresponds MPO HA(1) to 96 dB, HA(3) to 110 dB and MPO HA(5) equals 114 dB. All of them measured at 1000 Hz.

On fig. 2 are seen the responses of the pistol shot noise from the hearing aid in use on KEMAR. Protection against impulse noise is obvious at all three levels of the maximum output. The attenuation is greatest in the trebled area i.e. 25 to 35 dB depending on the maximum output setting. Greatest effect is obtained at the lowest output setting and is nearly of the same magnitude as for the closed ear canal which can also be seen from the time pattern in fig. 3.

Measurements on other types of hearing aids have also showed a protective effect on impulse noise.

From the experiments it can be concluded that the decrease of the gun shot peak or suppression of the impulse noise spectrum are attributed to the power limitation and the filter effect of the hearing aid.

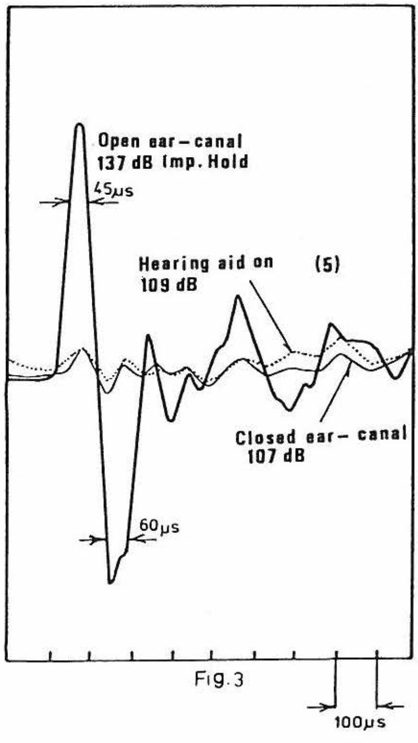

Fig.3

How the hearing aid acts as a hearing improving device can be seen on fig. 4. The lower curves with single circlets show the ordinary audiogram of a Greenlandic sealer.

On the basis of this and the gain curve for maximum output setting HA(1) calculations show how the audiogram would have looked like if the hearing aid had been in use. It is shown in the upper curve with concentric circlets.

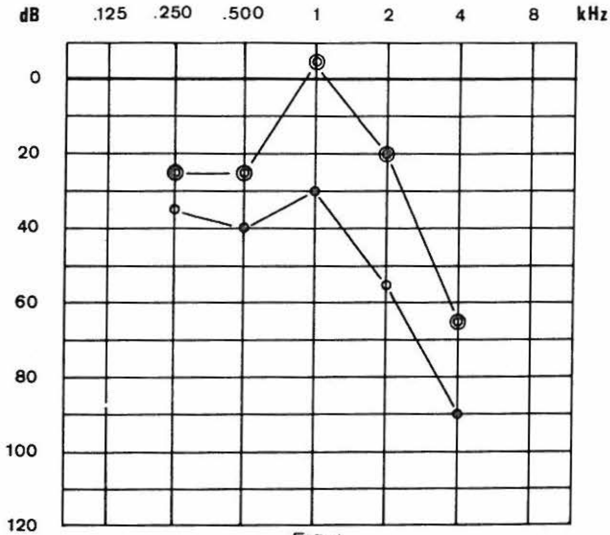

Fig. 4

Conclusion

A hearing aid in situ on KEMAR fitted with a closed earmold both improves the hearing and simultaniously gives a considerable suppression of impulse noise, which means that the hearing aid not only improves hearing but also protects hearing.

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Sound Field Audiometry in a Small Hearing Test Booth.

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Gothenburg, Sweden.

Introduction.

Many decisions regarding hearing-aid selection and use were earlier based mainly upon clinical impressions. New techniques have developed in the later years using insertion gain measurements based on acoustic evaluation of either miniture microphone or of probe measurements (1). These techniques have resulted in application of new rules and improved high frequency gain and better speech discrimination. However, there are indications that the insertion gain measurements does not significantly differ from the psycho-acoustical real ear measured gain (2). The latter method however has two requirements: presentation of warbletones and a freefield technique. Generally the sound field audiometry is applied in rooms of such constructions that a uniform field can be achieved (3). Such rooms are not always available. Arlinger (2) described the application in a moderate size test booth. The aim of this project was to find if sound field audiometry could be applied using the direct sound field from a speaker in a small hearing test booth.

Material and methods.

Using an ordinary audiometer, Inter-acoustics AC-5 in a small hearing test booth, (Tegner, sized 1.80 x 1.2 m). We investigated the sound field produced by different kinds of speaker positions.

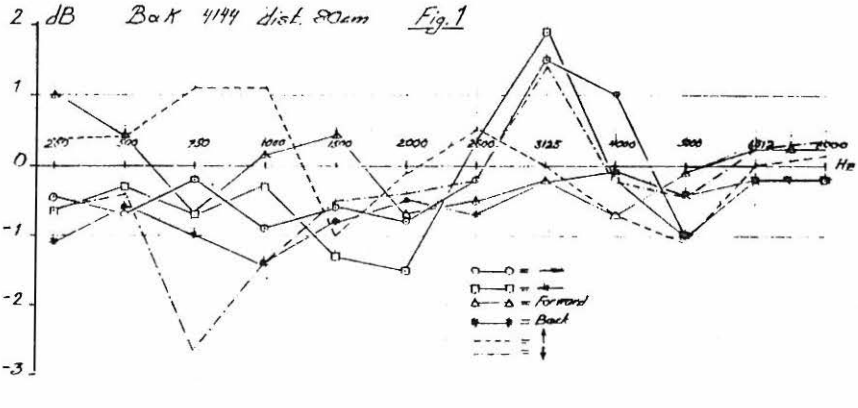
The hearing booth was provided with further damping material (Ilsonics Waffelstruktur). The inside of the door and the cable connecting plate was completely covered and 80 % of the window of the booth was also covered with removable damping material. The audiometer was re-built in such a way that tones with different degrees of modulation could be produced .

A test of the optimal speaker position was then done. The sound pressure was measured with the microphone in an average patient position and the ordinary hearing test frequencies were tested. In order to check the uniformity of the sound field the microphone was moved 10 cm in all directions and deviations were noted.

The described procedure was used for testing different speakers positioned at all corners of the booth, with different kinds of modulation. Beside testing the corner positions, the location of the speaker at the entrance door, and at the opposite wall was also tested and finally different distances between the speaker and the microphone position were evaluated.

Results

The best results with least variation in sound pressure was achieved in the near field about 0,8 m from the speaker positioned at the level of the head of the supposed patient. Figure 1 gives the result when moving the microphone 5 cm in each direction in this near field. Except for the frequency 750 Hz all measurement values was inside ± 2 dB from reference point and most values were well within ± 1 dB.



Five experienced subjects also participated in a test - retest experiment. Each of the five subjects underwent two series of hearingtests with the contralateral ear plugged with an EAR-plug. Each series consisted of five audiometry measurements containing all ordinary frequencies from 250 - 8000 kHz. In the first series testing was performed five times without the subject being permitted to move from the testchair. In the second series the subject was permitted to leave and re-enter the booth between each of the testruns. The results are given in table I showing the mean values of the variance (standard deviation). The mean standard deviation does not exceed three dB at any frequency. Significance testing (t-test) demonstrated no significant difference between the subjects, between the frequencies and or between the two testseries.

Table I. Test - retest results. Meanvalues of variance in SD for five repeated tests in 5 subjects. A: tests repeated without leaving the test booth. B: Tests repeated after leaving an re-entering the booth.

Test-frequency	250	500	1	2	2.5	3.125	4	5	6.3	8
A:	1.90	2.00	1.67	1.26	2.19	2.00	2.28	2.37	2.28	1.67
B:	2.53	2.28	1.79	2.28	1.90	2.45	2.76	2.83	2.28	2.76

Discussion.

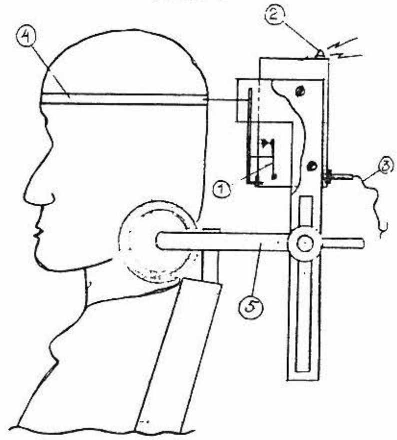
Measurement of the gain of hearing-aids has become an important procedure for evaluating the efficiency of hearing aids. Insertion gain measurements is a valuable method for measuring the acoustical part of the improved amplification (4). The method, however, contains some problems. The uncertainty of the probe position may contribute to variations of the acoustical response. Also the feelings and experience of the patient are not investigated and the equipment for insertion gain measurements is fairly expensive.

Sound field audiometry offers a less expensive solution where also the patients experience is included in evaluation. Suitable anechoic or acoustically damped rooms where diffuse sound fields can be produced is not easily available. Re-building of a hearingtest booth could therefore offer a more practical approach.

Figure 1 indicated that in order to keep the acoustical variation inside 1 dB for most frequencies it is important that the position of the ear does not move more than ± 5 cm in any direction. To be certain of this requirement being fulfilled a fixture was constructed (Fig. 2). The test - retest performance indicates a test - retest reliability which is of the same level as an ordinary earphone audiometry procedure.

Our investigation indicate that the results of measurement are dependent of fixation of the head of the patient in a stable position and that deviations from this position must be possible to register and control not to exceed ± 2 cm. It has been argued (3) that it is impractical to keep the position of the subject's ear constant mainly based on the need to be certain that the subject's ear is in position during the calibration and testing procedure. The easiest way to solve this is to provide the patient with a frontlet which is mechanically connected to a microswitch (figure 2).

FIG. 2



The microswitch construction which directly gives an indication when the subject is not able to cooperate. Because of differencies in size of the patients it is necessary that the chair is adjustable in height and is provided with a good headsupport. It is then possible to adjust the axis of the speaker with patient's ear.

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Title:

Interference & Frequency Analysis In The Cochlea

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Introduction

How the basilar membrane (BM) acts as a highly selective frequency analyser in spite of its large damping is one of the burning questions of cochlear mechanics. Though a multitude of hypotheses have been proposed, no one theory can resolve this problem, [1].

I have been examining cochlear transduction with respect to fluid motion. Earlier a hypothesis was presented ascribing importance to fluid motion within the organ of Corti, [2]. Two major suppositions of this theory were a stiff tectorial membrane (TM) and a fluid connection from the spiral sulcus to the scala media via the reticular lamina.

Experimental observations in rat cochlea showed the hypothesis to be incorrect, [3]. Further experiments showed the major suppositions were likewise incorrect. This has led to a reformulation of my ideas and a new hypothesis concerning cochlear transduction. The hypothesis is as follows. Frequency analysis in the cochlea is due to an interference phenomenon.

In this paper I will present a synopsis of the interference interpretation of frequency analysis. Calculations are presented from a simulation where interference yields sharpening. In addition, experimental testing of this new hypothesis is indicated.

Interference Model Formulation

A one-dimensional idealization of the organ of Corti is presented in fig. 1. The reticular lamina is closed and the TM is replaced by a moveable piston with displacement (x_3). The stimulus exciting the inner hair cells (IHC) is assumed to be the point of closest approach of the TM to the IHC cilia.

Analysis of the fluid equations follows from 2 with the following exceptions. The diameter of the sulcus is assumed to vary along the cochlear partition and a lower bound is presented to TM motion that being the IHC cilia height.

The resulting equation describing the pressure, p, in the sulcus is of the following form

$$-\alpha \frac{\partial^2 p}{\partial Z^2} - \frac{\partial \alpha}{\partial Z}\frac{\partial p}{\partial Z} + \beta \frac{\partial p}{\partial t} = \frac{\partial x_1}{\partial t} \qquad (1)$$

where Z is the direction along the BM, x_1 is the BM displacement and α and β are quantities dependent on the width and stiffness of the TM, diameter of the sulcus, the height of the arch and the thickness of the reticular lamina. Equation (1) combines terms involving a travelling wave with those of diffusion. Substituting the values of the aforementioned quantities as measured by [4], [5], [6] yields the following surprising result. The terms involving diffusion and the travelling wave are of equal importance. Furthermore, the magnitude of the diffusion terms are highly dependent on the thickness of the reticular lamina.

A computer simulation of egn (1) has been made, [7] The resulting "tuning curve" is shown in fig. 2. Curve b shows the assumed BM envelope while curve a presents the computed value of x_3 (proportional to p) at 10 mm from the stapes. Curve a is calculated with a high density of points about the peak region.

An increase in the slope of the low frequency amplitudes for curve a is noted. Depending on which points one chooses to define the sharpening of curve a, this curve can be sharper than curve b. In addition two side peaks or shoulders at about 6 and 11 kHz are calculated.

Essentially, the force stimulating the IHC is due to the interference between the change of sulcus volume due to the travelling wave and the diffusion of pressure within the sulcus.

Proposed Verification

It is a supposition of the interference hypothesis that diffusion has the same speed as the travelling wave. It may be possible to measure the diffusion speed by mechanically stimulating the BM and recording the acoustic and electrical outputs [8] and [9].

Additionally the interference hypothesis predicts a greater sharpening in the TM over the sulcus than for the BM. One might thus expect a greater sharpening in the BM under the tunnel region than further toward the spiral ligament. An indication of the predicted shoulders can be seen in neural tuning curves. [1] .

Discussion

Frequency analysis with interference phenomena is an exciting new aspect warranting further study. At present this hypothesis affords the following interpretations.

1) Damping Of BM - Due to the incompressibility of the fluid within the organ of Corti, the resulting pressure in the sulcus will feed back to the BM. This will change the BM's vibration in a manner proposed by [10]. The order of magnitude of this effect can be calculated.

2) Physiological Vulnerability - Changes in the ionic content in the scala media initiate changes in the thickness and position of the TM, [11], [3]. It is probable that the thickness of the reticular lamina changes likewise. This will drastically alter the relationship between diffusion and the travelling wave thereby eliminating the interference and hence frequency analysis.

Conclusion

This paper has described an interference theory of frequency analysis. Considering the maping of frequency to position along the cochlear partition, the following concept is envisioned. The cochlea behaves as if it were a set of "diffraction gratings".

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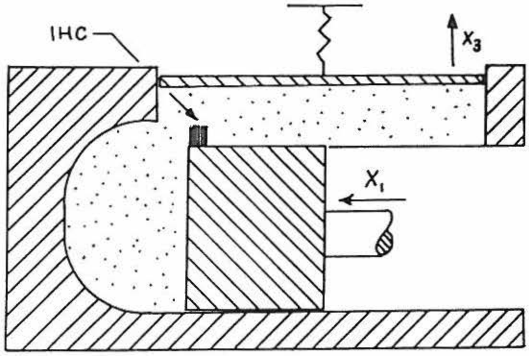

Fig. 1. A one-dimensional idealization of the organ of Corti.

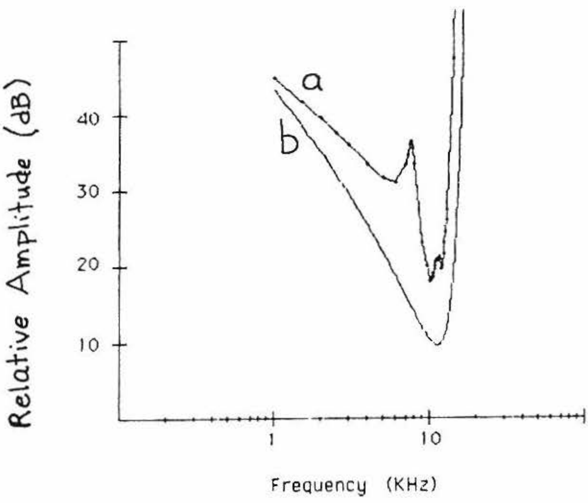

Fig. 2. Simulated tuning curves.

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Glenn Frommer/JT

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Speech Recognition at Aalborg University.

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I : Introduction.

In 1981 R&D within the area of Speech Recognition was initiated at the Institute of Electronic Systems at Aalborg University.
Since then much work have been focused on bringing existing automatic speech recognition (ASR) techniques into real world applications. Therefore, much of the hardware developed have been designed in close collaboration with partners from Danish Industry and semipublic companies.

The ultimate goal of the research in ASR is to be able to design a man-machine interface, which will enable communication with computers by spoken, natural language. Consequentely a number of strong demands on the R&D are given among which can be mentioned 1) the implementation of a rule-based acoustic-phonetic expert system, 2) the elicitation and use of extended phonetic knowledge and 3) development of hardware to extract robust phonetic features in real-time.
The following sections present the status of the R&D work. Section II gives information on the man-machine interface, which is being developed for the activities. Section III give some details on dedicated hardware and software developed for a one hundred word real-time speech recognition system using the DTW-technique. Section IV presents the status of some preliminary experiments on HMM-technique in the man-machine interface. Section V outlines the basic ideas behind the development of the rule-based phonetic/lingvistic expert system. Finally, section VI describes the computing facilities

available for establishing the rulebases for acoustic-phonetic speech recognition.

II : Microcomputer development- and application system.

A microcomputer system has been set up to function as a general man-machine interface for different applications, which will use Dynamic Time Warping (DTW)-, Hidden Markov Model (HMM)- or Acoustic-Phonetic techniques to achieve speech recognition.
The basic microcomputer is an industrial standard Intel 310 system running under the real-time, multiprocessing operating system iRMX 286. The system is equipped with a 19 Mb disk, 2 Mb RAM and a graphics board connected to a colour monitor. The system has two input channels for data acquisition and can be connected to a peripheral 24-bit signal processor and a dedicated signal acquisition and processing card, SAPCA.
The SAPCA contains an 80186 CPU, a COMBO chip, which reduces the frequency contents in the speech signal to telephone bandwidth and contains AD/DA converters, an advanced programmable DMA controller and three NEC77P20 signal processors.

To prepare for the use of ASR in application systems in which recognition of short phrases of continuous speech is essential, a dedicated rule-based expert system has been developed and implemented in the microcomputer system. This development system is able to apply phonetic rules given acoustic-phonetic features - e.g. formant transitions, energies in specific frequency bands and distinctive features such as sonorant etc. The development system is based on a number of knowledge sources representing specific expertise on phonetics, phonology and syntax, and a number of real-time signal processing facilities.

III : Software and hardware for DTW-technique.

The SAPCA is developed for use with applications, which uses DTW-and HMM-techniques for recognition of isolated words.
One processor on the SAPCA is fully engaged in real-time signal processing. The speech signal is transferred into the signal processor via the COMBO chip, which takes care of antialiasing, filtering and sampling. The speech signal is sampled at 7.8 kHz in 8 bit and A-law converted into 13 bit resolution. The signal processor performs the conversion and computes the autocorrelation coefficients and the rho-vector coefficients. The autocorrelation coefficients are computed sample by sample using a recursive window of lenght 45 msec. For each 15 msec the processor delivers a set of rho-vector coefficients. The computation of the rho-vector - which is the autocorrelation of the linear prediction coefficients - is performed intermixed with the updating of the autocorrelation

coefficients.
The training session produces the reference templates for the words in the chosen vocabulary for each specific speaker. Reference templates are modelled using the autocorrelation coefficients.
Based on the stored reference templates the remaining two signal processors perform the distance measuring. Each stored reference template is compared to the test template - equivalent to the word to be recognised and modelled by the rho-vector coefficients - using the Itakura-Saito distance metric.
Each signal processor handles four reference templates at a time in a recursive manner. This means that distance measurements begin as soon as a start/stop detection algorithm indicates that a test word is present at the input. Distance computations continue until a stop is detected. The start/stop detection is running in real-time in the CPU of the SAPCA.

The SAPCA is layed out on a 8-layer printed circuit board. The external bus interface complies with the MULTIBUS standard.
A diagram of the SAPCA is given in figure 1. Further details will be given in the poster presentation : **Signal Acquisition and Processing Card - SAPCA** by Thomas Balle.

IV : Preliminary experiments on HMM-technique.

To prepare for applications where integration of HMM-technique will be useful for recognition of isolated words, experiments in the development system has been conducted to test the feasibility of this technique.
The technique requires stored models of the words in the vocabulary and stored codebooks, which models the statistics of the speech signals from the training session.
Training of the codebooks - a shape codebook and a gain codebook - was carried out on a CDC Cyber 170 mainframe, which also has been used to establish the models of the specific words using the Baum-Welch reestimation algorithm.

The system has been trained by 20 male and 20 female speakers. The vocabulary chosen consist of the numbers '0' - '9' and the words 'Start' and 'Stop'.
Experimental tests on recognition of isolated words using the development system has shown preliminary results which indicates a recognition rate higher than 97%. Viterbi's algorithm is used in the recognition task.
Work is in progress to utilise the real-time signal processing capabilities of the SAPCA for the HMM-method.

V : Rule-based acoustic-phonetic speech recognition.

The long term goal of the R&D within speech recognition at

Institute of Electronic Systems is to be able to recognise short phrases of natural speech.
This research is done in close collaboration with Institute of Phonetics, Copenhagen University (IPUC). The research combines expertise within signal processing and hardware engineering, phonetics and computer science to establish a system, which will make it possible to communicate with microcomputers using spoken, natural language. The approach is to use a rule-based phonetic/lingvistic expert system in connexion with advanced signal processing, which in real-time extracts robust phonetic features from the speech signal.
The recognition system is an AI-system and the operation is based on a number of independent knowledge sources (KS) each of which on a blackboard display several hypotheses of what was uttered. Based on these hypotheses the driving system - the inference engine - causes synthesis of new or further refined hypotheses until an acceptable hypothesis for the utterance is achieved.
The independent knowledge sources, which e.g. contains knowledge on relations between acoustic segments and allophones, are described by rules. These are expressed using a syntax, which is very close to the description presented in Chomsky and Halle's book 'Sound Pattern of English'.
The different KS's relates to different levels of phonetic interpretation. There exist KS's, which are related to the levels themselves, and KS's which describes relationships between different levels, upwards and downwards from each level. The levels used in the present state of the research are shown in figure 2.
The KS's shown describes the expertise on the segmental, the allophonic, the morphological and the phrase levels formulated into rules.

A tool for establishing the expert system - HEAD (heuristic experiments, analysis and development) - is being developed. The goal of this is to be able to express the rules for and make experiments on speech recognition using acoustic-phonetic technique. During the development of the HEAD system it has been found necessary to be able to formulate rules using a terminology and notation, which is close to ordinary phonetic notation, such that a phonetician will be able to express his/her knowledge in a well known and precise manner. The phonetic rule language is processed by the phonetic rule compiler, which is an integral part of HEAD.
The HEAD system is interactive. It manages the rulebases, communicates with the phonetician via graphics display and does reasoning using the rules. Additions and changes of specific rules are easy, therefore these can be tested out very fast.

The structure of the HEAD system is given in figure 3.
Input/output, windows etc. are manipulated via the human interface module. The editor module manages the rulebases, the editing etc. The inference engine controls the application of rules for the different rule bases and the manipulation of the

hypotheses on the blakboard. The inference engine is controlled by the strategy database.

All rules in the databases are compiled into AI-Pascal code for the expert system. AI-Pascal is a language, which is developed for this research, having facilities for representing rules and representing and manipulating hypotheses. Another facility of the language is backtracking.

VI : Facilities for development of rule bases.

For the purpose of developing the necessary rule databases for the research towards recognition of short phrases of natural spoken language, a complete development system has been set up. The system will also be used to establish large databases for use with HMM-technique.
The development system is equipped with a Symbolics 3675, an array processor FPS 5105, a Digital Sound Corporation 200/240 data acquisition system, and the SPIRE speech analysis program developed at MIT.
This system is available also to industry, who may want to utilise techniques for speech recognition in their products and production.

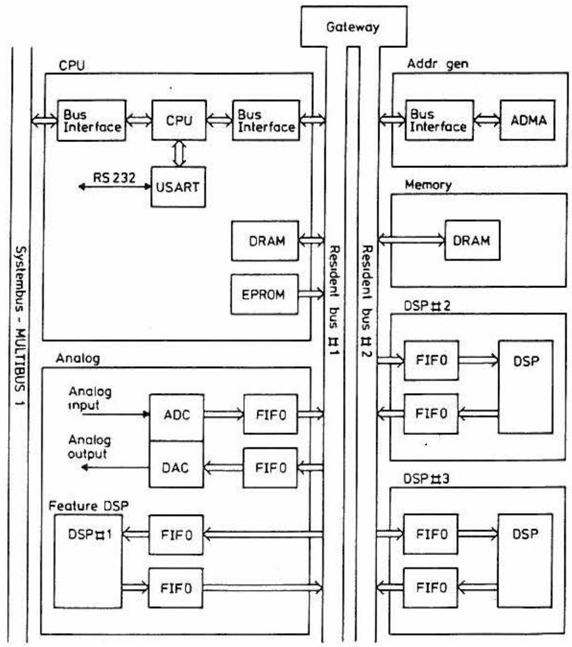

Figure 1. SAPCA.

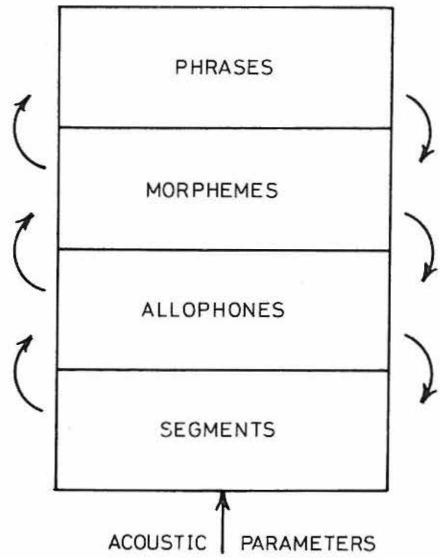

Figure 2. Levels of Phonetic expertise.

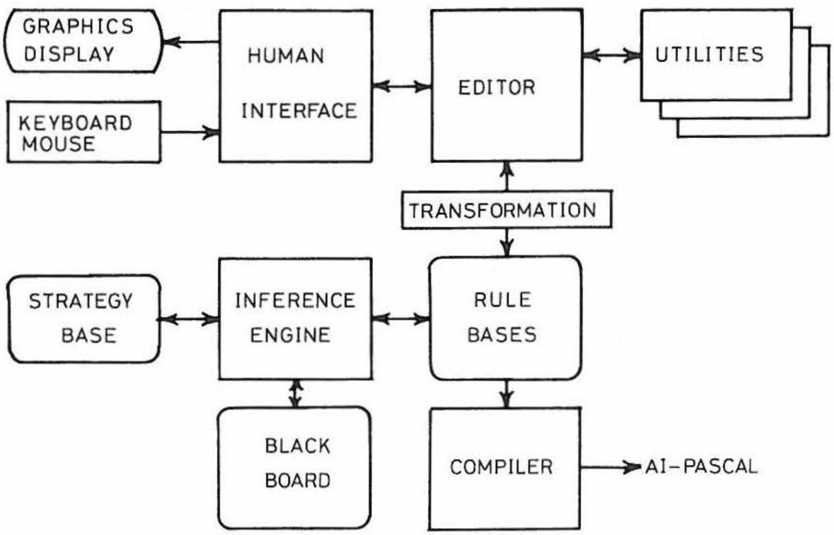

Figure 3. Structure of HEAD system.

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RULE SYNTHESIS OF VARIABLE INTONATION

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In our rule synthesis project, designed primarily for synthesizing **Hungarian** [6], we have been using the speech synthesizer OVE IIIb controlled by an HP 21 MX real-time computer [1]. The system is based on the use of four-phase diphones (keys), developed starting out from Sovijärvi's [2] beat phase theory for word stress (Fig 1). According to this theory, the primary phase of stressed vowels is almost always followed by a beat phase. This phase is followed by a weaker after phase and a final phase. Early synthesis experiments resulted in the corresponding division of all consonants into four phases. Each diphone has four matrix rows consisting of two subphases both before and after the boundary of two speech sounds [3]. In order to make the synthesized speech meet reasonable quality requirements, we have taken into consideration a total of 41 qualitatively distinct classes of Hungarian speech sounds. The letter codes for these 41 sounds have been appended to an earlier article [6]. The whole 'library' contains a total of about 1300 keys, 50 of which represent special Finnish and German sounds.

In order to avoid an impression of monotony, **microprosodic** alternation was encoded in the structural information contained in the keys. The falling contour of A0 typical of unstressed vowels has been imitated (Fig 1). (A minus sign has to be used because of the necessary glide function.) The fundamental frequency of each vowel varies within a range of one semitone (Fig 2). In the phases of consonants the values of A0 depend on the manner of articulation. The key value of F0 is constantly 82 Hz for all consonants.

PRODUCING PROSODIC VARIANTS

In actual synthesis the input sequence also contains special prosodic control symbols (Fig 3) needed for regulating the different optional variations in intonation and stress contours [4]. In

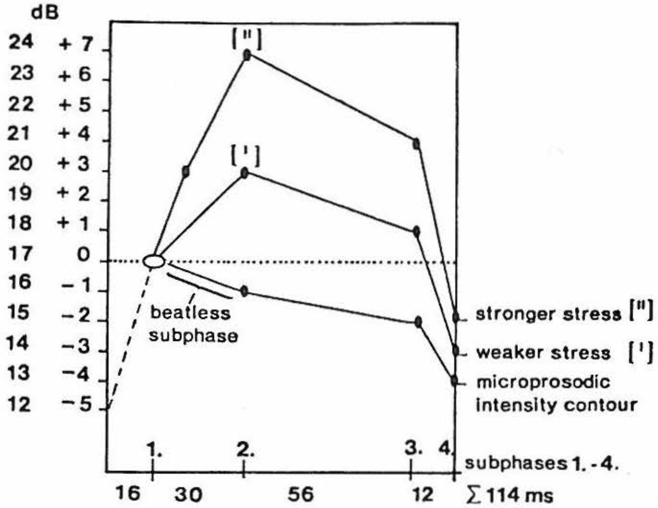

Fig. 1. The intensity contours corresponding to the stress degrees of vowels. The duration values of the four subphases are those of a short open labial [å] vowel.

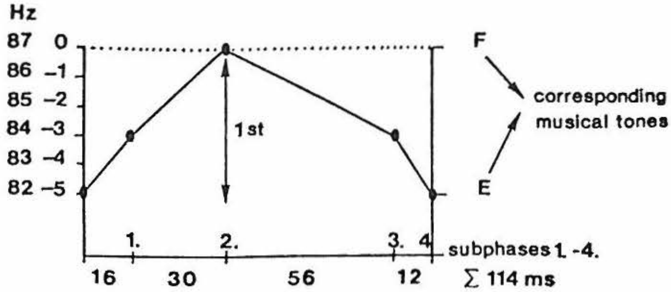

Fig. 2. The basic pattern of the microprosodic vowel intonation has a slightly rising-falling form.

vowels, the fixed, slightly rising-falling F0 contour of the microprosodic basic pattern (the range of which is one semitone) is always preserved, regardless of the application of any prosody symbols. As many as 23 different combinations of two or three of these control symbols can be used in the input sequences.

" and ' Stronger and weaker degrees of stress. The former symbol " produces a 10% increase in duration, an increase in F0 by 4 semitones calculated as percentages, and a rise in A0 by the dB values 3, 8, 6 and 2 in the successive subphases of the stressed vowel; furthermore, in the vowel of the following syllable, F0 increases by 2 st, and, in the consonant(s) appearing between the syllables, by 3 st above the level of 82 Hz. The latter symbol (ie weaker stress) produces an increase in F0 by 2 st and a slight rise in A0 by 0, 4, 3 and 1 dB in the successive subphases of the vowel after the symbol.

? Interrogative intonation. F0 rises by 7 st during one syllable.

+ Rising intonation. Before the slash or the pause sign F0 increases by 1 st for each sound until the fifth sound.

- Falling intonation. F0 falls until the slash or the first pause sign gradually during the three next sounds.

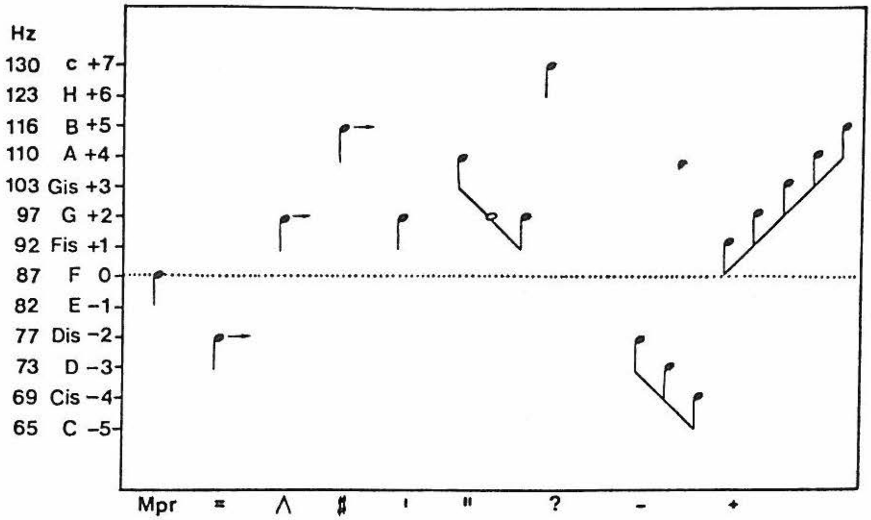

Fig. 3. The F0 ranges of the symbols used for controlling the levels of pitch and stress. The peak value (87 Hz = F) of the basic microprosodic contours of vowels has been fixed in this system as the zero point (= Mpr) of the pitch space. The length of the stem of the notes corresponds symbolically to one semitone.

\# and ^ Higher and lower raised F0 level. In a sequence delimited by brackets (<>) the overall pitch is increased by 5 or 2 st, respectively.
= Slightly lowered constant level of F0 (2 st).

In order to test the auditive quality of intonation in our synthesis system, examples of some affirmative and interrogative sentences in Finland-Swedish, German and Finnish were constructed. These examples are presented with their input symbol strings and the pitch levels of the syllable nuclei (in semitones, relative to the microprosodic zero level). (Other examples of synthesized sentences, especially in Hungarian, can be found in some earlier papers published by our team [1, 3, 5, 6, 7]).

ON THE APPLICATION OF THE METHOD

At the present stage the controlling system does not allow the synthesis of intonational phenomena of those languages which are characterized by an alternating "winding" tonal patterning and often by its relatively variable durative dimensions. The languages of this kind include for instance American and British English, French, and Swedish spoken in Sweden. In contrast, the language type in which the intonational patterns are largely based on the fixed or nearly fixed patterning of quantity and stress in the concatenation of syllables and/or words is suitable for the synthesis system presented in this paper. Besides Hungarian, at least the following languages seem to fulfil these conditions: Finnish, Swedish spoken in Finland, Estonian and German [5]. For further investigations of the special prosodic phenomena of these languages it is, however, necessary to write additional rules, and to create new keys.

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APPENDIX (SF = Swedish spoken in Finland)

1. **Några intonationsformer** (Neutral information; SF)

```
_N'O2GR-A2;/'INTONA2;Λ<C2:'U:NS-FOR>MER_
   2    -2   2  0 0        4    -1  -4
```

2. **Vilken dessert behagas det?** (Polite question; SF)

```
Λ<_V'ILK::E2;ND-E2;S'E:R_>=<BE2;H-A2/+GA2;SDE2;_>
     4    2     0    2        -2   -4   0    3
```

3. **Da es nieselte, blieb ich im Hotel...** (Progredient intonation)

```
_D'A2E2;SΛ<N'I:-ZE2;LTQ;_>BLI:BIJ2_IM+H2OTE2;L:_
   2 0       4   -1  -2     0  0   0    2 4
```

4. **Willst du heute nach Hause fliegen?** (Neutral question)

```
_VILST:UΛ<H'OE2;TQ;>NA2;X:=<A2;O2;>Λ<ZQ;+FLI:GQN_>
  0    0    4    2    0      -2         2    5  7
```

5. **Lisääntyvätkö happosateet?** (Wondering question)

```
#<_LI>Λ<SE:NT-YVETKQ/>H2'A2;P:O-SA2;TE2T_
    5    2    0-2 -2     2    0  -3  -4
```

6. **Uutislähetys päättyy tähän.** (Polite announcement)

```
_'U:TIS=<LEH2E2;TYS>Λ<PE:>#<T::Y:>=<TEH2-EN:_>
  2  0   -2   -2 -2     2         5    -2  -4
```

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Signal Acquisition and Processing CArd - SAPCA.

Thomas Balle
Jutland Telephone Ltd.
30, Sletvej
DK-8310 Tranbjerg, Denmark.

This poster presents the Signal Acquisition and Processing CArd - SAPCA described in the paper : **Speech recognition at Aalborg University** by Paul Dalsgaard.
SAPCA is developed specifically for use with applications where DTW- or HMM-techniques are utilised for recognition of isolated words. Using DTW-technique SAPCA is able to recognise 100 isolated words within the response time of 200 msec.

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MEASUREMENT OF THE MATCHING BETWEEN TWO MICROPHONE CHANNELS

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1. Introduction

Accurate methods for determining or compensating the gain and phase mismatch between different measurement channels are of great interest today in applications such as *acoustic intensity* measurement methods and *in-duct two-microphone* measurement methods.

Note, when only the gain mismatch between the channels are of interest standard calibration methods are normally available.

The focuse in this paper will be on methods to measure the matching between two microphone channels. Here, no generally accepted method seem to exist, instead several methods has been discussed in the literature. To our knowledge only one systematic investigation on the subject has been published, Seybert and Graves [1]. They studied three different microphone configurations (fig 1b), and found that in general the so called "end-mount" was the best choice.

The aim of this paper is to give some general comments on the problem in question. These comments mainly concerns points which we feel was either overlooked or misleading in reference [1]. Also some practical experiences from our own small investigations will be given.

2. Some theoretical considerations

Consider an acoustic measurement channel, it generally consists of several parts, like; transducer, preamplifier, cables, LP-filters, etc. Within the dynamic range of the channel it can be completely charcterized by its transfer function (T). Assume that the acoustic field quantity we want to measure is pressure ($\hat{p}$), then the following relationship is valid in the frequency domain

$$\hat{e} = T\,\hat{p} \tag{1}$$

where $\hat{e}$ is the output voltage from the channel.

If we now assume that we have two measurement channels (1 & 2) of the same type, then their transfer functions will normally differ somewhat. This difference will cause the need for amplitude and phase calibration.

It can also be noted that we can assume that the disturbance from the transducers on the original acoustic field is included in the channel transfer function T.

Directly from basic definitions the following relation is easily derived

$$H'_{12} = H_{12}K_{12} \tag{2}$$

where H_{12} is the true transfer function between two points in an acoustic field, "prim" denotes the measured quantity and $K_{12}=T_2/T_1$ is the transfer function between the measurement channels.

The fundamental method for determining K_{12} is the *switching technique*. With this technique we interchange two parts of the measurement channels, i.e., one part of channel 1 is interchanged with the corresponding part of channel 2. We now assume that the transfer functions of the channels before the interchange are $T_1=A_1A_2$ and $T_2=B_1B_2$, where A_1, B_1 describes the parts to be interchanged and A_2,B_2 describes the rest.

After the interchange (switch) the transfer functions for the channels are $T_{1,s}=B_1A_2$ and $T_{2,s}=A_1B_2$. The measured transfer function now becomes

$$H'_{12,s} = H_{12}K_{12,s} \tag{3}$$

where $K_{12,s}=T_{2,s}/T_{1,s}$.

From eqs (2) and (3) we directly obtain the following results

$$(H'_{12}H'_{12,s})^{1/2} = H_{12}(B_2/A_2) \tag{4}$$

$$(H'_{12}/H'_{12,s})^{1/2} = B_1/A_1 \tag{5}$$

Eq (4) should be used when we are not directly interested in K_{12}, but only want to compensate for it.

We see that the true transfer function H_{12} is obtained if $B_2/A_2=1$. This means that only the parts of the measurement channels that fulfil this can be left unswitched. When no parts of the measurement channels fulfil this relation, the channels must be completely interchanged in *all* their parts.

In reference [1] it is stated that it is not possible to interchange the analyser part of the measurement channels, this statement is however incorrect. However, it can be noted that there is one part of the channels that is physically impossible to switch. That part corresponds to the disturbance caused by the transducers on the field. Examples on studies of this problem can be found in references [2-3]. The main disadvantage in using eq (4) is that the effective measurement time is doubled. One advantage in using it is that we become almost unaffected by drift problems in the instrumentation. This last point can be a problem when we use eq (5) as described below.

When we use eq (5) we are either interested in determining the complete transfer function between the channels or the transfer function between two parts of the channels. When we know the complete channel transfer function K_{12}, we can use it in measurements to determine the true transfer function from eq (2).

To perform measurements according to eq (5) we should use a special calibration set-up, see fig 1. In reference [1] it was found that the end-mount was the best choice, mainly because it eliminates the occurence of pressure nodes at the microphones.

Most investigators using the end-mount uses the assumption that, below the first cross-mode, the sound field is equal at the microphones. This means that in eq (2) $H_{12}=1$, and we can directly obtain K_{12} from this eq. without any switching. However, Seybert and Graves [1] argues that misallignment of the microphones could make channel-swithcing necessary also for the end-mount case.

To study this question we assume plane wave propagation and that the microphones are located in two different duct cross-sections. A simple deduction then gives that the true transfer function between the microphones is

$$H_{12} = \cos(k\Delta) + i[(1-r)\sin(k\Delta)]/(1+r) \qquad (6)$$

where k is the wavenumber, r is the coefficient of reflection at the mic.1 duct cross section, i is the imaginary unit and Δ is the distance from mic.1 to mic.2.

From eq (6) we can estimate the resulting error, if we choose to not perform channel-switching.

Example: $\Delta=0.5$ mm f=1000 Hz

r	Rel.error in the gain (%)	Error in the phase (deg.)
1	- 0.004	0
0.95	- 0.004	0.01
0	0	0.5

The conclusion one can draw from this example is that, when the microphones are mounted in a cross-section where r is close to 1 then the sensitivity to error in position is very small. Therefore for the end-mount case, in a nearly rigid termination, channel switching is unnecessary.

3. Some practical considerations

At our department we have used the so called two-microphone method [3] to study sound propagation in ducts, pipes and similar systems. To compensate for the mismatch between the two microphone channels we have used a calibration procedure based on the end-mount arrangement. For our calibration set-ups we have used standard PVC-plastic tubes with an end plug also made in PVC.

The fitting between the plug and tube should of course be a little tight. The only problem we have encountered with this type of low cost set-up is that, above some critical sound level the plug will start to vibrate. This can strongly disturb the calibration, however, the disturbance is easily detected from the measured result and this problem can therefore be avoided.

To get an optimum use of the dynamic range in the analyser, we would ideally want the input signal to be frequency independent.

However, when we use the rigid end-mount configuration the system can be highly resonant unless we introduce some damping. This can be done in two ways, either by inserting some porous material near the source end of the tube or by using long tubes. An example of the latter is given in fig 2. The increase in smoothness for the phase-curve measured in the longer tube is obvious from this figure.

Ref 1 A F Seybert and D K Graves, Noise Con 85 p 423-428.
Ref 2 P S Watkinson and F J Fahy, J Sound Vib 94(2),p299-306 (1984).
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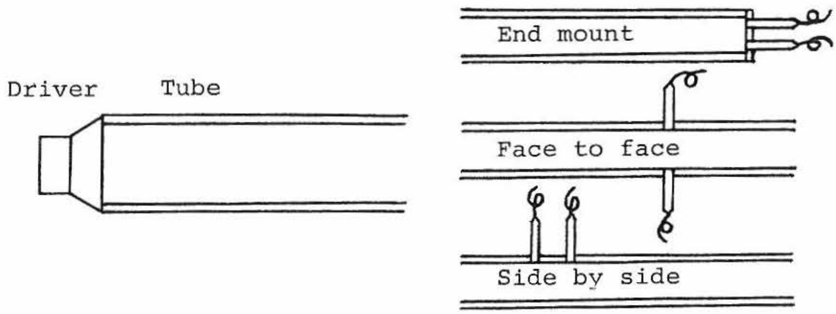

Figure 1 a) Measurement setup b) Microphone configurations

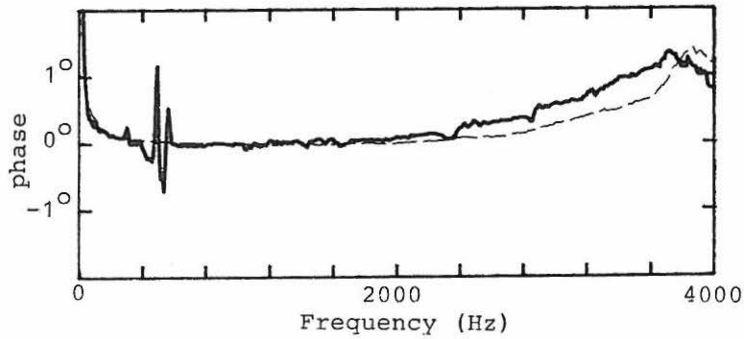

Figure 2 Phase mismatch measured with the end mount configuration, ▬▬ 0.3m long tube, – – –6.9m long tube

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NEW TYPES OF PRESSURE MICROPHONES FOR SOUND INTENSITY MEASUREMENTS

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INTRODUCTION

Application of pressure microphones for sound intensity measurements places stringent requirements on the microphone phase chracteristics. No particular characteristic is required, but the characteristics of microphones used together should be identical. This is especially necessary at low frequencies and in highly reactive fields where very small phase differences must be detected.

Microphones cannot be produced within the needed phase tolerances even under carefully controlled production conditions. Sets of matching microphones are therefore obtained by selection; in spite of this the microphones are the limiting factor in intensity measurement systems of today.

New microphones and a new phase calibration method have been developed. This is described together with extended possibilities for the measurement of sound intensity opened by these new microphones.

TRADITIONAL AND NEW TYPE MICROPHONE SETS

There are mainly two mechanisms which cause significant phase spread between microphones within a type.

At higher frequencies the diaphragm damping causes a spread which is practically proportional to the frequency; at 1000 Hz it is typically 2 deg., while the phase discrepancy is less than 0,2 deg. for selected pairs of the same type. Comparison of this discrepancy with the differences in actual sound fields shows that damping has a minor frequency independent influence on the intensity measurement accuracy. At low frequencies the pressure equalization causes a spread which is very disturbing for intensity measurements due to the small phase differences of low frequency sound fields.

The diaphragm's deflection and thus the microphone's output signal is determined by its front and rear-side pressure.

The cavity pressure, which comes from the pressure at the vent, has a magnitude which decreases proportional to frequency and a phase lag of about 90° above 20 Hz.

This phase lagged rear-side pressure leads to a resulting phase lead of the microphone's low frequency response which is typically between 2 and 6° at 20 Hz. The significant variation is mainly due to reproducibility problems with the vent resistances.

The significant influence which the rear-side pressure has on the low frequency phase response can be reduced by attenuation of the pressure's magnitude and by a change of its phase.

Some solutions have been analyzed theoretically and experimentally (Ref.[1]). These analyses have lead to the development of new types of microphones containing two extra compliance-resistance networks in series with the primary pressure equalization vent: see Fig. 1. The two networks are built into an extension of the microphone housing.

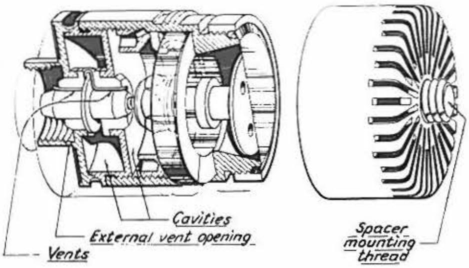

Fig. 1. New pressure microphone with two extra RC–networks

Each RC-network causes a phase lag of a further 90° and a magnitude reduction which at 20 Hz is about 20 dB. The combined effect of all three networks is a phase lag of the diaphragm's rear pressure of about 270° (a lead of about 90°). This angle is actually as critical as the 90° in the traditional microphones. But the magnitude of the non-desired pressure signal caused by the vent in the new microphones is typically 55 dB lower at 20 Hz and it decreases for increasing frequency by 18 dB/oct. instead of 6 dB/oct.

This explains why the new types have far less low frequency phase spread and very low sensitivity to sound pressure at the external opening of the pressure equalization systems than earlier types.

Mathematical microphone models including the heat conduction effect have been made for a number of types including those mentioned.

Calculated phase response characteristics are shown in Fig. 2 for a new and a traditional type. These calculations are made for cut-off frequencies of the primary venting network of 1 Hz and 2 Hz respectively. Notice the significantly smaller phase deviation of the new type; however, there is an influence from the production tolerances of the extra networks but the resulting spread is reduced by a factor of 15. Sets of the new microphones are selected to a low frequency matching better than 0,05° which is a reduction of 4 in comparison with existing microphone sets.

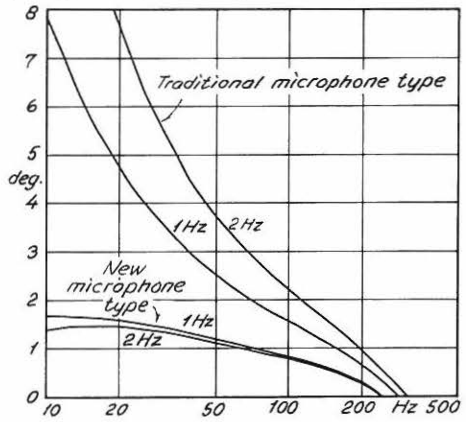

Fig. 2. Phase characteristics for cut-off frequencies of 1 Hz and 2 Hz of primary venting systems

ADVANTAGES GAINED FOR INTENSITY MEASUREMENTS

The frequency range, dynamic capability and measurement accuracy of intensity instrumentation can be improved by the new and better matched microphone sets due to reduced system phase errors below about 300 Hz; system errors are typically reduced by a factor of 2 at 20 Hz, and are dominated by electronic mismatch. Measurement ranges for probes with new and old microphone sets are calculated and shown in Fig. 3.

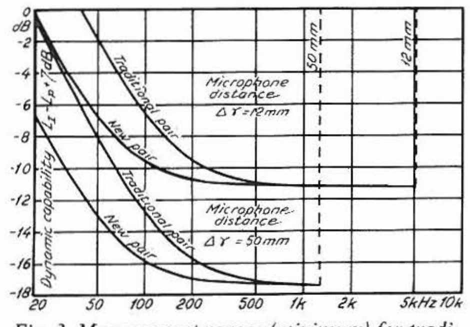

Fig. 3. Measurement ranges (minimum) for traditional and new microphone sets, for an error of 1 dB

One consequence of the very low vent sensitivity is that most practical calibrations can be made simpler, for instance by the use of small (wide band) couplers, as only the diaphragms have to be exposed to the sound pressure. Phase calibration by electrostatic actuators connected in parallel has also become a possibility for these microphones.

Point source measurement are improved close to sources. This is shown in Fig. 4 by the calculated intensity frequency responses for probes with traditional and with new microphones. Close to the source the magnitude of the pressure decreases significantly for small distance increments, therefore if the probe is placed in such a field the pressure will be different at diaphragms and vents.

As the field conditions are not equal for the two microphones their phase shifts become different, depending on source distance. For traditional microphones this leads to a significant intensity measurement error, while for the new microphones having low vent sensitivity this type of error is practically eliminated.

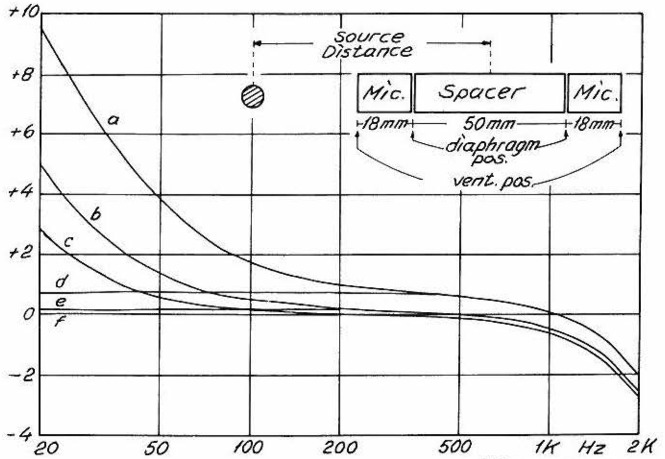

Fig. 4. Intensity probe frequency responses close to point source when equipped with traditional microphones of 2 Hz cut-off frequency and with new types of microphones.
(a), (b) and (c) Traditional microphones; source distances are 63 mm, 125 mm and 250 mm respectively
(d), (e) and (f) New types of microphones; same distances respectively. The calculated curves are valid for the indicated distances between source and center of spacer.

PHASE CALIBRATION METHOD

To measure the small phase errors of the new microphone sets, a better calibration method had to be found.

As a phase measurement system having sufficiently small absolute phase errors is not available, the interchange principle which excludes system phase errors is applied. The phase discrepancy between the microphones is found by two measurements where the second measurement is performed with interchanged microphones.

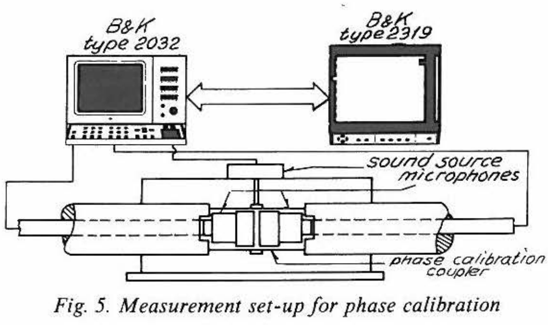

Fig. 5. Measurement set-up for phase calibration

The set-up shown in Fig. 5 was chosen due to its good resolution and stability. The sound source of the phase calibration coupler is excited by the pseudo random noise generator of Type 2032 (dual channel FFT analyzer). The cross and auto spectra of the two microphone signals are measured before and after the interchange of the microphones. A subsequent equalized frequency response mode gives the phase difference (in radians) equal to half of the imaginary part of the ratio between the two frequency response functions. The results are only valid for small phase discrepancies, but this condition is fully satisfied for the actual application. Measurements of phase discrepancies below 1 deg. have been performed with an accuracy of 0,005 deg. and a resolution of 0,001 deg. by the use of the analyzer, Type 2032.

A wideband phase calibration coupler has also been developed. In this cylindrical coupler the two microphones are placed closely together on the coupler axis with their protection grids facing each other. The microphone vents are also inside the coupler cavity even though this is of no importance above 20 Hz for the new microphones, due to their extremely low vent sensitivity.

The sound inlet to the coupler which consists of a number of ports is designed to produce a rotational symmetrical sound field inside the coupler over the frequency range from 20 Hz to about 6 kHz. During calibration the microphones are excited from the periphery as when they are slit-mounted in intensity probes.

Fig. 6 shows a phase calibration of a typical new microphone set covering 0–800 Hz and 0–6,4 kHz respectively.

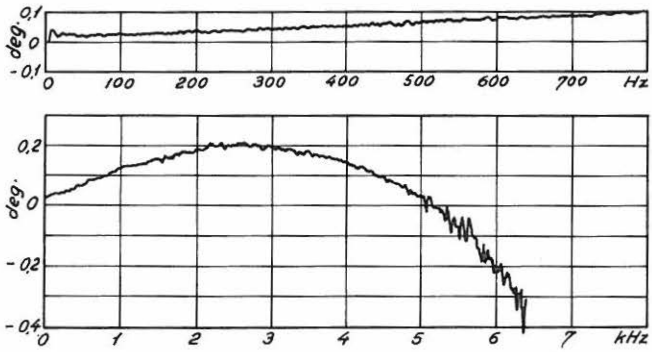

Fig. 6. Phase calibration for typical set of new microphones

CONCLUSION

New microphone types characterized by a very low vent sensitivity and a small low frequency phase spread have been developed. Improved intensity microphone sets are selected from these. Existing intensity measurement systems can utilize most of the advantageous properties which are obtained.

Before the reduced phase discrepancy can be fully used one has to wait for a reduction of phase errors in the electronic equipment which at low frequencies has taken over the role of being the limiting factor.

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SOUND INSULATION OF VENTILATION ELEMENTS USING SOUND INTENSITY TECHNIQUE

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INTRODUCTION

The sound insulation provided by a wall depends to a great extent on the windows, doors, air vents and other elements which introduce, from the acoustical view point, a weak spot in the wall. Manufacturers of such building elements are often called upon to document the sound insulation provided by their products.

Halton Oy of Finland manufacture amongst other items a large range of equipment for use in ventilation systems. Halton would like to measure the sound insulation of their products and include the results in the Product Data Sheets. The most obvious measurement method to follow would be the NORD TEST method NT ACOU 037 approved March 1982, entitled "Small building elements: sound insulation". Examples of equipment covered by NT ACOU 037 are transfer air devices (TAD), airing panels (ventilators), outdoor air intakes and cable ducts. The basic descriptor described in the document is the unit insulation denoted by D_I and evaluated from

$$D_I = L_1 - L_2 - 10\,\mathrm{Log}\left(A/S_0\right) \quad \mathrm{dB} \tag{1}$$

where L_1 = average sound pressure level in source room
L_2 = average sound pressure level in receiving room
S_0 = 1 m^2
A = equivalent absorption area in receiving room.

If the sound fields are not completely diffuse and if sound is transmitted by paths other than through the device under test then equation (1) is an approximation. Measurements made in accordance with the wording of NT ACOU 037 therefore require a full transmission suite, a facility which Halton Oy do not possess. However, Halton do have a 200 m^3 reverberation chamber fulfilling ISO 3740 series and a Sound Intensity Analysing System Type 3360 so it was decided to try to devise a method based on sound intensity which would obviate the need for building a second chamber. This paper presents measurements of the sound insulation of a transfer air device as measured according to the "Classical", sound pressure method and the sound intensity method. Measurements were performed using the 3360 intensity system and also the newly developed intensity analyzer Type 4433 and probe Type 3520.

INSTRUMENTATION

The Type 3360 is a parallel, 1/3 and 1/1 octave intensity system and is widely used. The battery operated, octave intensity analyzer Type 4433 is of interest to those who wish to measure quantities such as sound power and sound reduction in situ. The 4433 weighs less than 6 kg and can operate for more than 7 hours on its internal batteries. Its small size (138 × 251 × 300 mm) enables it to be brought right to the measurement site even when space is restricted. It is capable of measuring sound intensity, sound pressure and particle velocity in the eight octave bands with centre frequencies from 63 Hz to 8 kHz as well as the linear and A-weighted values. For a complete sound intensity analysing system, the analyzer must be equipped with a sound intensity probe. A new probe has been specially developed for use with the portable analyzer. This probe is based on the two microphone technique as is the well-known Type 3519 but it employs phase matched 1/2″ prepolarized condenser microphones of modified design [1]. The 4433 is a serial analyzer, that is, the measurements are performed in the octave bands consecutively and stored in the internal memory [2].

MEASUREMENT CONDITIONS

Measurements wer performed on a Transfer Air Device (TAD) of dimensions 900 mm × 125 mm mounted in a sandwich construction of three hard board panels and a layer of mineral wool which in turn was mounted in a wall of the reverberation chamber. The chamber itself stood on pneumatic vibration isolators in a very large and irregularly shaped hall. The particular volume of the hall in the vicinity of the transfer air device was high and narrow and divided into two by a metal catwalk.

MEASUREMENTS

The measurements can be grouped under the following headings:

1. Calibration
2. Testing postulate that energy density within reverberation chamber does not change whether or not TAD is in test opening
3. Insertion loss of TAD
4. Unit insulation of TAD using classical and intensity method.

The frequency range of interest was in the 6 octave bands with centre frequencies from 125 Hz to 4 kHz. For all intensity measurements a 12 mm spacer was used.

CALIBRATION

The Residual Intensity Index, $L_{K,0}$ of the intensity analyzer Type 3360 was measured by applying the same pink noise signal to both direct inputs of the analyzer simultaneously. The difference between the measured sound intensity level, L_I, and the measured sound pressure level, L_p, was calculated and printed out by the Graphics Recorder Type 2313 fitted with the intensity Application Package BZ 7004. The Residual Intensity Index of the complete intensity analysing system, that is, Probe Type 3519 and analyzer, was checked by first calibrating the two microphones individually using a Pistonphone Type 4220. Then the acoustic coupler WA 0344 was mounted on the pistonphone, the assembled probe was inserted into the coupler after which the sound pressure level and the intensity level within the coupler was measured in third octaves using the two frequencies available, 250 Hz and 315 Hz. The total system was found to be well within the manufacturers spsecifications. The specification states that the total phase mismatch of the system should be within 0,3°. The measured values are better than 0,05°.

The Residual Intensity Index of the portable Intensity Analyzer 4433 and Probe 3520 was checked in a similar manner using a broadband coupler before arrival at Halton.

TESTING POSTULATE CONCERNING ENERGY DENSITY

The sound intensity measurement technique enables the insertion loss of objects to be determined using only one special room. The method requires that a diffuse field be established in a reverberation chamber which is furnished with a test opening. The environment outside the reverberation chamber must be acoustically dead. The insertion loss of the object is then simply the level difference between the two intensity spectra measured before and after the insertion of the object in the test opening. The postulate made here is that the energy density within the reverberation chamber does not change significantly after the object is put in place. This postulate was tested by measuring the sound pressure level within the chamber over a circular arc of radius 0,8 m with an averaging time of 64 s with and without the TAD installed. The results in Table 1 show that the postulate was justified and that the method could be employed to determine the insertion loss of the TAD.

Hz	With TAD dB	Without TAD dB	Difference dB
125	0,4	80,4	0,0
250	82,2	82,2	0,0
500	85,3	85,4	− 0,1
1 K	83,7	83,7	0,0
2 K	86,4	86,4	0,1
4 K	82,6	82,5	0,1

Table 1. Difference in the sound pressure levels in reverberation chamber measured with and without the TAD installed in the test opening

INSERTION LOSS OF TAD

The diffuse sound field in the reverberation chamber was produced by a Sound Source Type 4224 operating in wideband noise mode. This sound source had been modified to produce somewhat greater effect at higher frequencies.

To measure the intensity level over the test opening, with and without the TAD in place, the intensity probe was swept back and forth over a square measurement surface situated at about 10 cm from the test opening. The measured intensity spectra and the insertion loss of the TAD are given in Table 2. The measured Reactivity Index before and after insertion of the TAD indicated that the measurement conditions were within the capabilities of the instrumentation.

Hz	Without TAD dB	With TAD dB	Insertion Loss dB
125	64,1	58,7	5,4
250	64,9	59,6	5,3
500	69,2	65,3	3,9
1 K	69,1	55,9	13,2
2 K	70,4	46,7	23,7
4 K	66,3	39,2	27,1

Table 2. Sound intensity radiated from measurement area before and after insertion of TAD into test opening and insertion loss of TAD

UNIT INSULATION OF TAD USING CLASSICAL AND INTENSITY METHOD

The unit insulation of the TAD using the classical method was measured according to equation (1) in both directions first using the reverberation chamber as the source room and the hall as the receiving room and then reversing the rôles of the rooms. The equivalent absorption areas of the reverberation chamber and the hall were measured by placing a reference sound source of known sound power into the room under test and measuring the spatially averaged sound pressure level produced.

The equation is:

$$10 \,\mathrm{Log} \frac{A}{A_0} = L_{wr} - L_{pr} + 6 \,\mathrm{dB} \tag{2}$$

where A is the equivalent absorption area
A_0 is $1 m^2$
L_{wr} is the sound power level of the reference sound source
L_{pr} is the spatially averaged sound pressure level.

The expression for the unit insulation using the classical method $D_{I,\text{classical}}$ thus becomes:

$$D_{I,\text{classical}} = L_1 - L_2 - L_{wr2} + L_{pr2} - 6\text{ dB} \tag{3}$$

where the suffices 1 and 2 indicate source and receiving room respectively.
Using the intensity method, the expression for the unit insulation $D_{I,\text{intensity}}$ becomes:

$$D_{I,\text{intensity}} = L_1 - L_{I2} - 10\text{ Log}\frac{S}{S_0} - 6\text{ dB} \tag{4}$$

where L_{I2} is the intensity level in the receiving room as measured over area S close to the TAD [3].

DISCUSSION OF RESULTS

The results of the unit insulation measurement are shown in Fig. 1. The low values of unit insulation obtained by the classical method when the reverberation chamber was used as the source room are due to the invalid assumption that the sound field in the hall was diffuse. Thus the method for determining the equivalent absorption, A, could only give an indication of the effective A and thus the value of $D_{I,\text{classical}}$ obtained is also fraught with inaccuracy. The value of $D_{I,\text{classical}}$ obtained when the hall was used as a source room gives more reliable results. The lack of a diffuse field at low frequencies on the source side, is the probable reason for the low values in the 125 Hz and 250 Hz octave bands. The dip in the curve at 4 kHz is probably due to the pronounced directivity of the sound source at this frequency. This is being investigated further.

The unit insulation obtained using the intensity method with the reverberation chamber as source room gives results very close to the classical method (reverberation chamber to hall) in the octave bands 500, 1 k and 2 kHz. The higher values at 125, 500 and 4 kHz are reproducible to within ± 1 dB and are considered to give a more realistic indication of the unit insulation than the classical method.

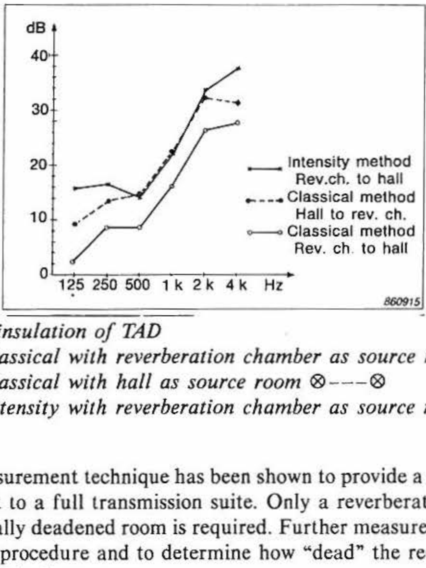

Fig. 1. Unit insulation of TAD
D_I, classical with reverberation chamber as source room ⊙—⊙
D_I, classical with hall as source room ⊗---⊗
D_I, intensity with reverberation chamber as source room X—X

CONCLUSION

The sound intensity measurement technique has been shown to provide a means of measuring unit insulation without resort to a full transmission suite. Only a reverberation chamber with a test opening into an acoustically deadened room is required. Further measurements are being made to refine the measurement procedure and to determine how "dead" the receiving room must be to produce results of the desired accuracy.

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INTENSITETSMÅLINGER I KANALSYSTEMER.

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1. Innledning

Direkte måling av intensitet gir nye muligheter for å kartlegge og forstå lydtransportfenomener i komplekse kanal- eller rørsystemer. Tomikrofon-metoden etter Chung og Blaser [1] er i utgangspunktet begrenset til en-dimensjonale bølgefelt, men den har den åpenbare fordel at det er mulig foruten netto-intensiteten å bestemme intensiteten i innfallende og reflektert bølge. I tillegg er den heller ikke basert på den vanlige gradient-tilnærmelsen for partikkelhastigheten, men et krav er at avstanden mellom mikrofonene er $< \lambda/2$. At metoden er begrenset til en-dimensjonale bølgefelt er en alvorlig begrensning ved anvendelse i praktiske ventilasjonskanaler. En kan tenke seg flere muligheter for å utvide det brukbare frekvensområdet. En måte er å dele opp den aktuelle kanal, dvs. en måleseksjon av denne, i et antall mindre parallelle kanaler som samlet har det samme areal som den opprinnelige kanal og hvor kravet om en-dimensjonal bølgeforplantning er oppfylt [2]. Et alternativ er å undersøke om metoden lar seg anvende for frekvenser over "cut-off" hvis en foretar en nødvendig midling over tverrsnittet av kanalen. På bakgrunn av det ekstremt komplekse lydfelt en vil ha for frekvenser over cut-off er det komplisert, kanskje umulig teoretisk å få en god nok oversikt over hvilke feil som introduseres [3].

Utgangspunktet for dette arbeidet er å undersøke muligheten for bruk av intensitetsmåleteknikk til å framskaffe "lyddata" for komponenter i ventilasjonssystemer, spesielt data som normalt er vanskelig målbare eller sparsomme i litteraturen. Dette gjelder bl.a. transmisjon gjennom kanalvegger, såkalt "break out" og "break-in". I tillegg til å se på mulighetene ved å utvide ovennevnte intensitetsmetode til høyere frekvenser skal vi her vise eksempel på slike data.

2. Teori.

Under forutsetning av plane bølger kan intensiteten i den innfallende bølge i en kanal skrives som

$$|\vec{I}|_{i1} = \frac{(G_{pp})_1}{4\varrho_0 c_0 \sin^2(k_0 s)} \left| e^{jk_0 s} - H_{12} \right|^2 \qquad (1)$$

og nettointensiteten

$$|\vec{I}|_1 = - \frac{(G_{pp})_1}{\varrho_0 c_0 \sin(k_0 s)} \mathrm{Im}\{H_{12}\} \qquad (2)$$

Her betegner indeksene 1 og 2 mikrofonposisjoner i en avstand $s < \lambda/2$, $(G_{pp})_1$ er effektspektral-tetthet av totaltrykket i posisjon 1 og H_{12} overføringsfunksjon mellom totaltrykkene i de to posisjoner. k_0 og $\varrho_0 c_0$ er bølgetall og karakteristisk impedans for mediet. Merk at uttrykkene forutsetter ingen strømning og intet energitap mellom mikrofonene.

3. Målinger og resultater.

I alle målingene er det benyttet kanaler med diameter 200 mm av type Spiro-rør. Disse har en langsgående skrueformet skjøt eller "søm". Diameteren tilsier en cut-off frekvens på ca. 1000 Hz. Det er utført målinger i frekvensområdet 50-6200 Hz. I frekvensområdet opptil 450 Hz er det målt i en posisjon, mens det for høyere frekvenser er målt i inntil 30 posisjoner over tverrsnittet. Frekvensområdet er vanligvis delt i tre områder med en viss overlapping. Dette er gjort delvis for å få til en tilstrekkelig høy linjetetthet idet FFT-analyseutstyret kun gir 256 komplekse data pr. måling, dels for å bruke så stor avstand som mulig mellom mikrofonene.

3.1. Intensitetskomponenter.

Figur 1 viser typiske data for innfallende intensitet (I_i) netto intensitet (I_n) og intensitet basert på kvadrert lydtrykk (I_{p^2}) i en kanal hvor lydfeltet er sterkt reaktivt. Kanalen er ca. 3 meter lang med en høyttalerkilde i den ene enden og helt åpen i den andre. Figur 1a dekker frekvensområdet 50-450 Hz mens figur 1b som representerer middelverdier over 30 posisjoner dekker området 1000-8000 Hz. I det siste tilfellet hvor refleksjonen fra den åpne enden er minimal faller kurven for I_i og I_n sammen mens I_{p^2} hele tiden gir noe høyere verdier. Figur 1b viser også noen verdier hvor kravet til mikrofonavstand ikke lenger er oppfylt. For en sammenligning av disse måleresultatene med data for mer konvensjonelle metoder har vi latt kanalen, 5 m lang munne ut i et klangrom og bestemt utstrålt effekt til dette etter ISO 3741. Netto intensitet i kanalen er målt i avstand 2 meter fra åpningen eller ca. 3 meter fra høyttalerkilden. I figur 2 er vist netto effekt i kanal i 1/3 oktavbånd sammenlignet med effekten trans mittert til klangrommet. Den gode overensstemmelsen mellom måledata over cut-off er bemerkelsesverdig og forskjellen kan kanskje i sin helhet tilskrives energitap i kanalen mellom måleposisjon og rommet. Ved å benytte en midling over mange posisjoner synes derfor denne to-mikrofon-metoden for intensitet også å kunne benyttes utover plan-bølgeområdet. I vårt tilfelle ville anslagsvis 10 mikrofonposisjoner være tilstrekkelig til å gi en nøyaktighet på 1 dB innenfor et 1/3 oktavbånd.

3.2. Effekt transmittert gjennom kanalvegg.
Som en illustrasjon på hvilke typer av målinger metoden hensiktsmessig kan anvendes på, vises noen data fra måling på såkalt "break-out". Vi har definert en transmisjonsfaktor for break-out ved

$$\tau_{k-r} = \frac{W_{tr}}{W_{ik}} \qquad (3)$$

hvor W_{tr} er den transmitterte effekt via kanalveggen til det omgivende rom og W_{ik} er den innfallende effekt i kanalen mot det betraktede kanalstykke. Figur 3 viser resultater hvor den lydstrålende lengde av kanalen utgjør henholdsvis 3, 6 og 12 meter. I de siste er det inkludert to, henholdsvis tre 90 graders bend. Data er angitt ved et reduksjonstall med utgangspunkt i (3) men normalisert til 1 meters kanallengde. Sett på bakgrunn av de forskjellige kanalkonfigurasjoner som danner utgangspunktet for disse resultatene og at det ikke er gjort noen korreksjon for energitap langs kanalen er avvikene mellom kurvene ikke avskrekkende.

4. Konklusjoner.
Det er vel kjent at tomikrofon-metoden etter Chung og Blaser har vist seg nyttig i forbindelse med kanal- eller rørmålinger, spesifikt ved bestemmelse av intensitetskomponenter, av impedans og absorpsjon. Våre resultater tyder også på at man i visse tilfelle kan benytte metoden også over "cut-off" hvis en foretar en nødvendig midling over kanaltverrsnittet. Dette har den fordel framfor den "vanlige" tomikrofon-metoden at kravet til avstand mellom mikrofonene blir langt mer lempelig.

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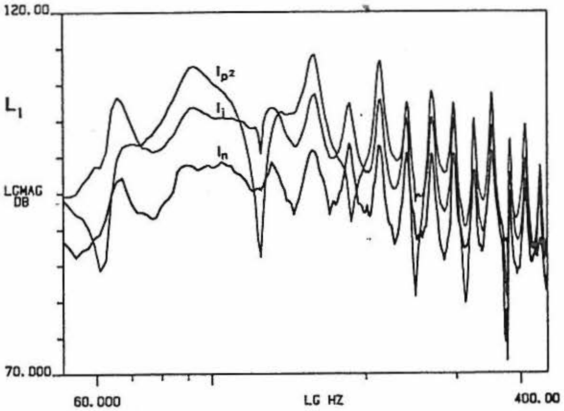
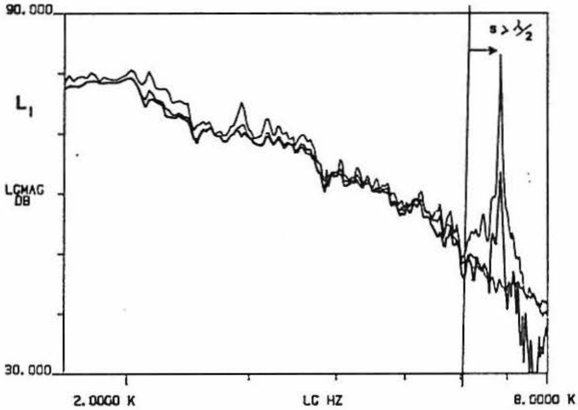
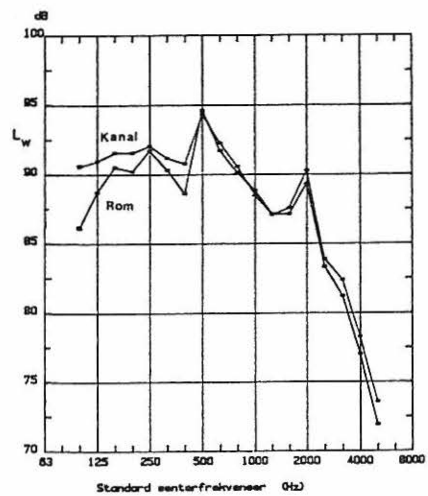


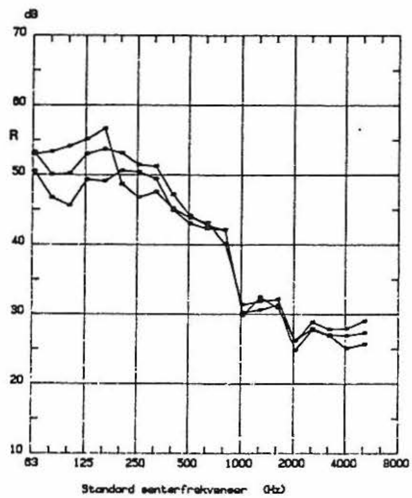
Figur 1a
Intensitetsdata for kanal med åpen ende
50-450 Hz
1 mikpos.



Figur 1b
1000-8000 Hz
30 mikpos.



Figur 2.
Effektnivå i kanal
sml. med effekt til rom.



Figur 3.
"Reduksjonstall" for
kanalvegg.

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THE RASTI METHOD FOR OBJECTIVE RATING OF SPEECH INTELLIGIBILITY

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INTRODUCTION

From the beginning of this century investigators have tried to find methods by means of which it is possible to evaluate speech intelligibility. This is quite difficult because many factors are involved, so until now only a few objective methods and subjective methods have been used. Subjective methods use different speakers who pronounce a carrier sentence in which a nonsense word is placed. The nonsense word is often a CVC-word (consonant, vowel, consonant), where the number of times a letter is used, is the same as in the language (phonetic balance, PB). Listeners try to understand the word; in this way the intelligibility is measured in a "PB-word score".

Objective methods ought to produce the same results as subjective methods, but should be much quicker. Until now these methods have usually taken only the background noise into consideration and therefore not the reverberation time in the room as well. But for the first time, it is now possible to assess speech intelligibility (by means of the RASTI method) in cases where both background noise and the reverberant field are taken into account. A measurement can be performed in less than 10 seconds. The RASTI method is standardized in a draft standard from IEC [1].

THE RASTI METHOD

RASTI is a method of quantifying the intelligibility of transmitted speech and is based upon the method of the Speech Transmission Index, STI, [2, 3, 4]. Perfect transmission of speech implies that the temporal speech envelope at the listener's position replicates the speech envelope at the speaker's mouth. Speech intelligibility can be quantified in terms of the changes brought about in the modulation of the speech envelope as a result of noise and reverberation in the room, Fig. 1. The reduction in modulation can be described by a modulation reduction factor. The modulation reduction factor expressed as a function of modulation frequency is called the Modulation Transfer Function, MTF. This function provides an objective means of assessing the speech intelligibility, and from it, the RASTI index is derived.

The test signal used in the RASTI method consists of two octave bands of noise as shown in Fig. 2. The levels for these two bands are chosen to be equal to the avarage levels found in normal speech (L_{eqA} = 60 dB), equivalent to 59 dB in the 500 Hz octave and 50 dB in the 2 kHz octave, all levels given at 1 m from the speaker.

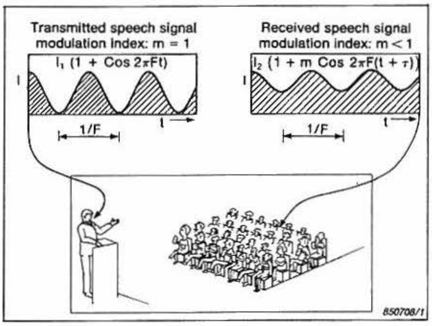

Fig. 1. Illustration of the reduction in modulation of a speech signal caused by background noise and reverberation

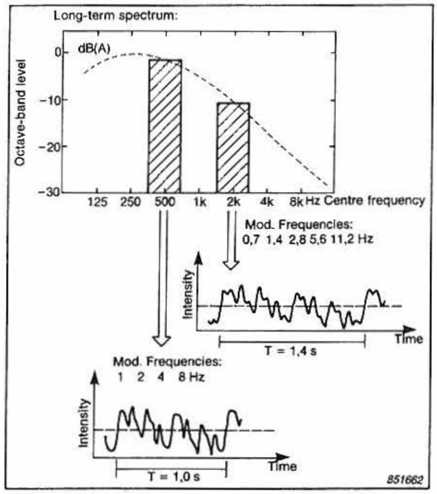

Fig. 2. Illustration of the RASTI test signal. Two octave bands of pink noise are presented at the same time, with an intensity envelope comprising four or five simultaneous modulation frequencies, and with a modulation index of 0,4 and 0,32, respectively

A RASTI measurement is made by transmitting the special test signal from the speaker's position (in the room considered or, in connection with public address system, in the control room) and analysing it at the listener's position. The reduction in modulation index for each of the nine modulation frequencies is calculated. The nine modulation reduction indices obtained are interpreted as though they were brought about by background noise alone, as indicated in Fig. 3. A qualitative interpretation of the RASTI-values is shown in Fig. 4.

CALCULATION OF THE RASTI VALUE

Nine apparent signal to noise ratios, one for each modulation frequency, are calculated as follows:

$$X_i = 10 \mathrm{Log}[m_i/(1 - m_i)]$$

where X_i is the apparent signal to noise ratio corresponding to the measured modulation reduction factor, m_i.

The X_i values are truncated at X_i = 15 dB such that:

if $X_i >$ 15 dB, then let X_i = 15 dB
if $X_i <$ −15 dB, then let X_i = −15 dB

The arithmetic mean of these 9 X_i values is obtained and normalized to yield an index which ranges from 0 to 1.

$$\text{RASTI value} = (\bar{X}_i + 15)/30$$

Fig. 3. Calculation of the RASTI-value

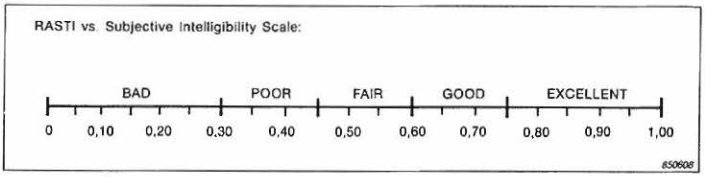

Fig. 4. Qualitative interpretation of RASTI

MEASUREMENTS

There exists many different applications for the RASTI method. By means of Brüel & Kjær Speech Transmission Meter, Transmitter Type 4225 and Receiver Type 4419 many different measurements have already been performed. One of the measurements performed was on "Fredericia Banegård".

The railway station of Fredericia is a very busy railway junction, situated at a crosspoint in the middle of Jutland. Like many other railway stations it has very poor acoustics caused by reverberant surroundings and a very high background noise level.

Fig. 5 indicates that the old sound systems was not able to provide an acceptable speech intelligibility, and many complaints were received. DSB, The Danish Railway System decided to take action, and a consulting company was hired to solve the problem. The solution chosen was to cover the entire platform with a direct sound field. Fig. 6 illustrates, this gave good results. And most important, the number of complaints went down, there was also an appreciable improvement in subjective speech intelligibility.

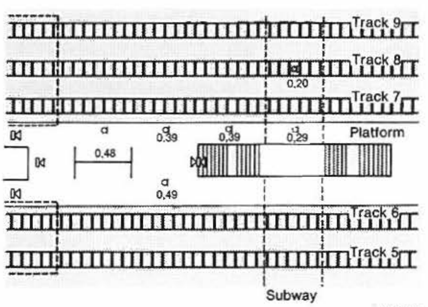

Fig. 5. RASTI measurement with Old Sound System

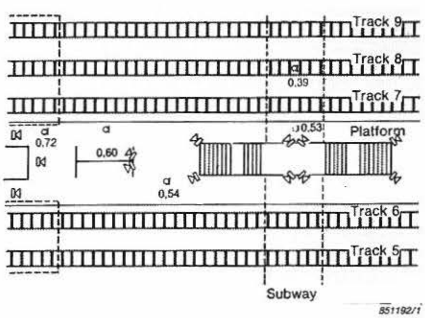

Fig. 6. RASTI Measurement with New Sound System

CONCLUSION

The new RASTI method has been shown to be a very practical and quick method for the assessment of speech intelligibility. Measurements performed with the Brüel & Kjær Speech Transmission Meter have shown good correlation between subjective assessment and the RASTI-value.

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MULTICHANNEL SAMPLING OF EXPOSURE TO NOISE AND VIBRATION

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Introduction

Usually workers are exposed to noise during exposure to local vibration. Combined exposure to noise and vibration has been shown to increase permanent or temporary threshold shift (PTS or TTS) [1,2]. In addition to the energy content of noise (ISO/DIS 1999.2) and vibration (ISO/DIS 5349.2), impulsiveness may also be a risk factor associated with PTS [3,4] and vibration-induced white finger (VWF) [5].

The purpose of the present study was to apply the digital high speed sampling technique to investigate the exposure of the workers to noise and vibration, the attenuation of noise by earmuffs, and the transmission of vibration from the tool to the wrist.

Methods of measurement

The measured pneumatic hand held power tools used in a shipyard assembly hall were a scaler (Atlas Copco RRC12), an angle grinder (Yatani YGS7GC), and a vertical grinder (Atlas Copco) with a grinding stone. The worker wore his own earmuff-helmet combination protector during all the measurements. Inside the earmuff a miniature microphone (Knowles 1785 LY072) was attached to the earlobe at the middle of the ear canal. Outside the earmuff a microphone of the same type was attached to the headband. Both signals were A-weighted in the preamplifiers.

Accelerometers (B&K 4375) were attached both to the handle of the tool with a mechanical filter (B&K, UA 0559) and to the operator's wrist (B&K 4371). The direction of the accelerometer in the handle depended on the work position and was attached parallel to that on the wrist. The signals from the accelerometers were sampled unweighted and weighted (ISO/DIS 5349.2). The preamplifiers and analog filters (Wärtsilä Oy, custom made and B&K 2634) were attached to the backside of the worker and connected via a 35 m cable to a multichannel digital sampling unit, which was controlled with a microcomputer (fig 1). The computer system was protected by a custom-made hermetic box against electric disturbances, heat and dust.

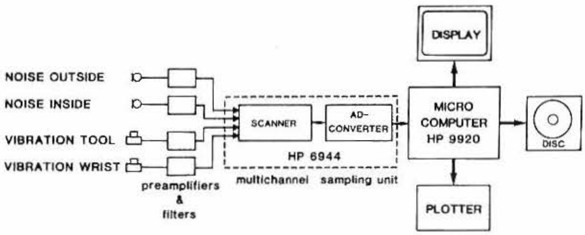

Fig.1 The microcomputer system for noise and vibration measurements

The noise and vibration channels were sampled simultaneously in pairs to analyse the attenuation of noise by the earmuff, and the transmission of vibration to the wrist. The sample rate selected was 40,000 Hz/channel, which was related to the upper frequency limit of 10,000 Hz in the preamplifiers. The duration of a time record was 200 ms which contained 8000 digital samples from each channel. Ten time records were sampled in the measurement of one tool and stored to a 20 megabyte hard disc during sampling.

The peak (L_{peak}) and root-mean-square levels (L_{rms}) were calculated from the sampled voltage values for each channel separately.

$$L_{peak} = 20 \cdot \log \frac{y_{max}}{y_o} \qquad (1)$$

$$L_{rms} = 20 \cdot \log \sqrt{\frac{1}{N} \sum_{i=1}^{N} \frac{y2}{y_o^2}} \qquad (2)$$

N = amount of digital samples in a time record
y_{max} = the maximum amplitude in a time record
y = instantaneous amplitude in a time record
y_o = reference amplitude ($p_o = 20 \cdot 10^{-6}$Pa, $a_o = 1 \cdot 10^{-6}$m/s^2)

Impulsiveness (I) was determined according to the definition, and it corresponded to the crest factor of the signal [3,4].

$$I = L_{peak} - L_{rms} \qquad (3)$$

Results

The earmuffs attenuated the peak levels and only the low frequencies were transmitted to the inside of the earmuffs. A nearly sinusoidal vibration was measured from the wrist (fig. 2).

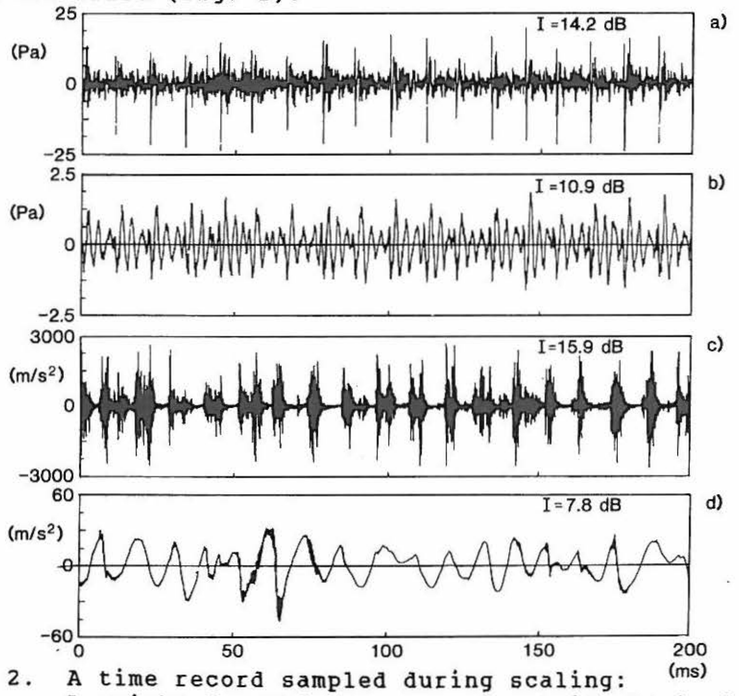

Fig. 2. A time record sampled during scaling: A-weighted sound pressure a) outside b) inside the earmuffs, unweighted vibration acceleration c) on the tool d) on the wrist

The average energy and time domain characteristics are independent factors describing the signal. The rms levels and impulsiveness of ten time records were averaged for each tool (fig. 3). The rms noise levels outside the earmuffs were about 100 dB during angle grinding and scaling and about 90 dB during vertical grinding. The impulsiveness of the noise was highest during scaling and lowest during vertical grinding. The length of the straight line is the total attenuation. The components consist of the attenuation of rms levels and the attenuation of impulsiveness. The average ($\bar{X} \pm SD$) attenuation of noise rms levels was (27 ± 5) dB. Impulsiveness was attenuated by the earmuffs (1.7 ± 1.0)dB.

The vibration acceleration rms levels were lowest during vertical grinding. The impulsiveness of vibration was highest during scaling. The rms level was (38±12) dB higher and the impulsiveness was (4.7±2.4) dB higher on the tool handle than on the wrist.

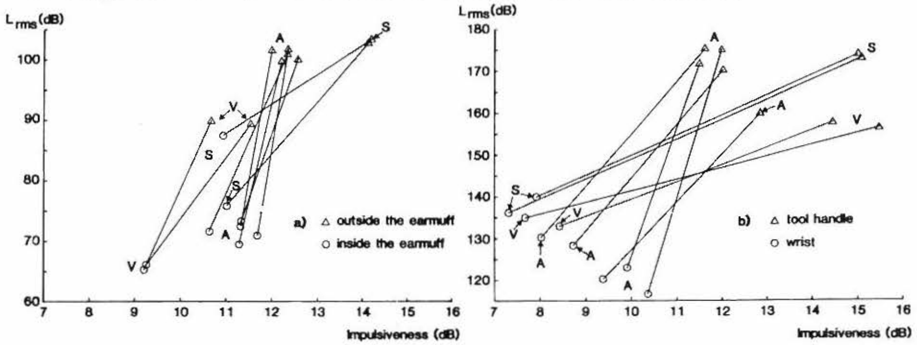

Fig.3 The rms levels and impulsiveness measured a) from noise outside and inside earmuffs b) from vibration acceleration of the tool handle and wrist.
A=angle grinding, V=vertical grinding, S=scaling

Conclusions

The results suggest that measurement of the ambient noise level is not sufficient to determine exposure of workers to noise when earmuffs are used. The earmuffs seem to protect against impulse noise at least when the impulses have a high frequency content, as is common in industrial impulse noise. The vibration from pneumatic hammering was shown to be impulsive. The vibration impulses are not transmitted to the wrist of the worker. However, it is unknown which component in the handle-hand-wrist system absorbs the major part of the impulses. Thus the vibration impulses may have harmful effects even though they are not transmitted to the wrist. Impulses may cause shock waves in the tissues, and also widely activate the central nervous system through local receptors.

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EXPERIMENTAL INVESTIGATION OF SIGNAL ANALYTICAL METHODS FOR VIBRATION CONDITION MONITORING OF ROLLING ELEMENT BEARINGS.

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INTRODUCTION

Vibration condition monitoring (VCM) is a relatively new discipline within machinery maintenance. During the past years the interest in this field has been increasing. This can probably be ascribed to the fact that it is comparatively easy to acquire information about the vibration level of a machine and that the vibration level has proved to be a fairly good indicator of the condition of the machine.

The present paper deals with the vibration signal as the source of information for on-condition maintenance of rolling element bearings, which often constitute essential components of rotating machinery. The experimental work comprises among other things comparative analysis and evaluation of methods and signal, a number of parameters which could be used for vibration condition monitoring purposes.

VIBRATION SIGNALS FROM BEARINGS

The vibration signal from a perfect rolling element bearing can be characterized as broad-band stochastic noise which is generated by friction and microscopic irregularities in the bearing surfaces. The presence of a fault in one of the bearing elements causes a change of the vibration signal. A beginning, local fault, i.e. a fault with little spreading, will cause a strong impulse to occur for every impact between the fault and a bearing element. Due to the rotation, this impulse is repeated with a characteristic frequency which can be calculated from the speed of rotation and the geometry of the bearing.

Each impulse excites natural frequencies in the nearest structure which can be observed as decaying high frequency oscillations. This effect is illustrated in Figure 1 which shows idealized sequences of impulses from local bearing faults.

In case of faults on a moving bearing element, the vibration signal is amplitude modulated due to the damaged area being moved into and out of the load zone.

MEASUREMENTS

All measurements were carried out with the bearings mounted in a specially designed test bed. The bearings were tested with variable radial load force and speed of rotation. Apart from undamaged bearings the test comprised bearings with the following damage:

a) Corrosion defects	c) Local (discrete) defects
b) Spalling defects	d) Damage arisen during long term test

Figure 3 shows a bearing after the completion of a long-term test.

During the measurements the radial acceleration amplitude was registered by means of a piezoelectric accelerometer. The vibration signals were recorded on tape for later analysis in the laboratory which gave a useful frequency range between approximately 20 Hz and at least 60 kHz ($\pm$ 3 dB).

ANALYSIS PARAMETERS

The signal parameters which have been tested are:

RMS-value	Spectrum (narrow-band)
Peak value	Cepstrum
Crest factor	Spectrum of signal envelope
Kurtosis	Spectrum of averaged signal envelope
	Autocorrelation function of signal envelope

The signal parameters have been calculated in a number of frequency bands between 10 Hz and 70 kHz. In the cases of spectrum analysis, cepstrum analysis and envelope analysis, usual baseband frequency analysis as well as zoom analysis have been applied.

MEASUREMENT RESULTS

RMS-value, peak value, crest factor and kurtosis, show a minor dependency on load and speed of rotation for undamaged bearings. When a damage is introduced, the dependency on the parameters of operation is often strongly increased.

For the long-term tested bearings the results achieved with cepstrum analysis and frequency analysis of the time envelope (envelope analysis) do not differ very much from each other. When performing similar analyses of signals from bearings with discrete faults a pronounced difference is seen.

In the spectra there are only very few and weak frequency components which can be related to the bearing damage. Hence, the corresponding cepstra give no indication of the damage. Generally the localization of the frequency range in which cepstrum analysis is performed is difficult and critical. In opposition to this, envelope analysis clearly shows the characteristic frequencies and related harmonic components. Moreover, it has been acknowledged that envelope analysis is quite uncritical with respect to the frequency range in which the analysis is performed.

CONCLUSION

Satisfactory results have been achieved with many of the tested parameters. However, rms- and peak values and kurtosis are sensitive to background noise. Moreover, kurtosis is strongly load dependent. In order to increase the signal-to-noise ratio the signal should generally be filtered before further treatment.

The envelope spectrum has proved to be an excellent tool for diagnosis of all types of local faults. The performance of the cepstrum parameter is fairly good for diagnosis of larger faults. Moreover, tests of a form of matched filtering, involving autocorrelation of the time envelope has shown encouraging results.

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Figure 1.
Idealized signals from bearings with faults on a) outer race b) inner race c) one ball.

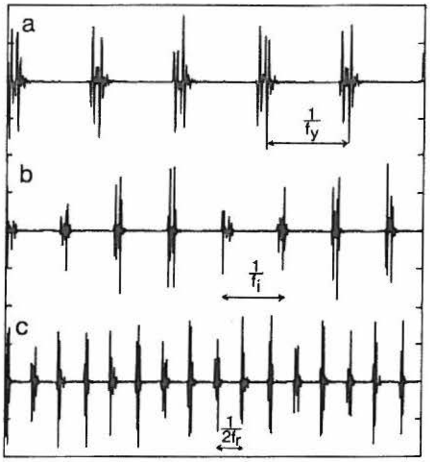


Figure 2.
Signal from bearing with discrete fault: a) raw signal b) band-pass filtered, enveloped time signal c) spectrum of time envelope.

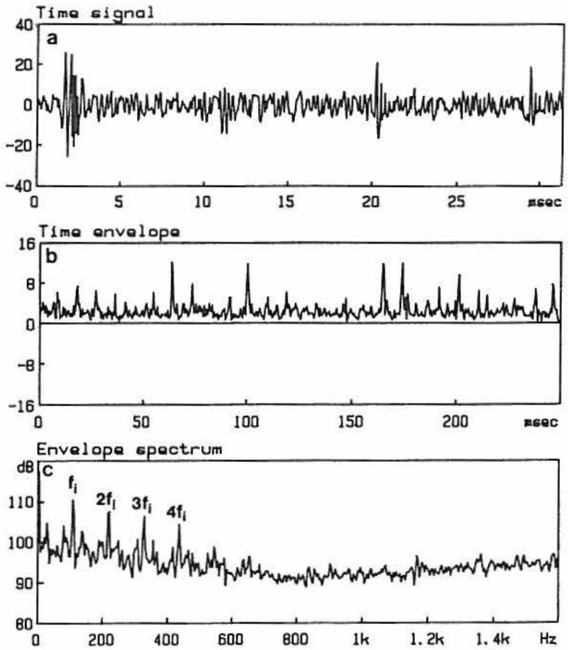


Figure 3.
Defective bearing after a long-term test (approx. 120 hours).

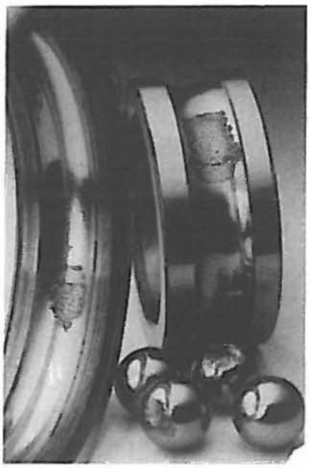
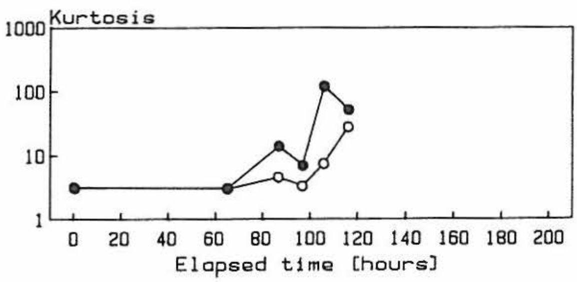


Figure 4.
The course of kurtosis during a long term test. Band-pass filtered between 10 kHz and 15 kHz.



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DATORSTYRNING AV RECIPROCITETSKALIBRERING

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1. INLEDNING

Då arbetet med riksmätplatsen för ljudtryck påbörjades baserades mikrofonkalibreringarna på Brüel & Kjaers reciprocitetskalibreringsapparat typ 4143. Denna är i normalutförande helt manuell utan någon möjlighet till yttre styrning.

Till att börja med var behovet av automatisering ganska litet. Efter det inledande utvecklingsarbetet var mängden reciprocitetskalibreringar vanligen begränsad till ca tre tillfällen per år och ett fåtal mikrofoner, typiskt tre stycken. Antalet frekvenser var litet, oftast bara 250 Hz och 1 kHz. Med denna lilla omfattning var det optimalt att fortsätta med den helt manuella metoden.

Efterhand har vi dock skaffat fler referensmikrofoner, som dels skall utgöra reserver och dels öka säkerheten i mätningarna. Dessa mikrofoner skall också mätas regelbundet och detta har medfört att mängden mätningar har ökat.

Vi deltar också i "IEC ONE-INCH MICROPHONE CALIBRATION INTERCOMPARISON" och vid dessa mätningar skall vi kalibrera vid tersbandsfrekvenserna 63 - 10 000 Hz, dvs betydligt flera frekvenser än vi brukar.

Totala mängden reciprocitetskalibreringar har alltså ökat och detta har medfört att det nu kan vara meningsfullt att automatisera dessa mätningar.

2. AUTOMATISERINGSALTERNATIV

Det finns olika möjligheter att automatisera denna typ av mätningar och flera laboratorier har gjort detta.

Vid NPL i Storbritannien [1, 2] finns en fullständigt datoriserad mätuppställning. Man använder enserieresistans för mätning av sändarmikrofonens exciteringsström och "insert voltage"-teknik för att mäta mottagarmikrofonens signal. Spänningsmätningarna görs med två digitalvoltmetrar.

Vid PTT i Finland utnyttjas Brüel & Kjaers 4143, dock med komparatordelen ersatt med en digitalvoltmeter [3].

Vid CSIRO i Australien [4, 5] används ett helt annorlunda arrangemang, med en induktiv mätbrygga. Datorstyrningen är här begränsad till signalkällan, för övrigt läses instrumenten av manuellt enligt instruktioner från programmet [5].

Även vid PTB i Tyskland görs mätningen manuellt, dock med datoriserad beräkning av slutresultatet [6]. Det elektriska arrangemanget runt mikrofonerna är mycket likt det vid NPL, men man använder sedan komparatordelen av en Brüel & Kjaer 4143 i stället för digitalvoltmetrar.

3. UTNYTTJANDE AV BRÜEL & KJAER 4143

Om man ändrade vår nuvarande mätuppställning så att den liknar NPLs skulle man få en del fördelar, bl a då man skall kalibrera enskilda komponenter i uppställningen. Exempelvis är det mycket lättare att kalibrera referensmotståndet som ingår där än att kalibrera referenskondensatorn i B&K 4143. Nackdelarna med att byta system är dock stora, det skulle innebära stora investeringar i ny utrustning och mycket utvecklingsarbete.

Eftersom vi använt B&K 4143 under lång tid och har erfarenhet av den har vi valt att bygga vidare på den. Vi har också redan två andra viktiga instrument som behövs, en tongenerator, HP3325A, och en digitalvoltmeter, HP3455A, båda med datoranslutning. Den metod vi valt är alltså att utnyttja "första delen" av B&K 4143, med referenskondensator och några förstärkarsteg. Signalerna tas sedan ut via filterutgångarna och mäts med hjälp av digitalvoltmetern. Komparatordelen i 4143 utnyttjas inte.

Vid manuell mätning med 4143 arbetar man i tre steg:
1) balansering av komparatorn
2) justering av "insert gain"-förstärkaren, som kompenserar för dämpningen i förförstärkaren
3) avläsning av känslighetsprodukten.

Vid det tredje steget mäter man kvoten mellan spänningen över referenskondensatorn och spänningen från mottagarmikrofonen. Samtliga mätningar är alltså kvotmätningar, man är intresserad av förhållandet mellan två spänningar medan deras absoluta värde har mindre betydelse.

Om man analyserar mätproceduren noggrannare finner man dock snart att det räcker med två steg. I det första steget mäter man upp kanalbalansen, skillnaden i kanalernas förstärkning, som då inkluderar dämpningen i förförstärkaren och i det andra steget mäter man känslighetsprodukten. Skillnaden mellan kanalerna kommer in som en korrektion i det andra steget.

I ett automatiserat system är man fortfarande beroende av funktionsomkopplaren i 4143 men man utnyttjar bara två lägen, "Insert Gain" för mätning av kanalbalans och "Sensitivity Product" för mätning av känslighetsprodukt. Någon justering av kanalernas förstärkning i instrumentet gör man inte utan mätprogrammet korrigerar för eventuell skillnad.

Tillverkaren erbjuder en modifiering av 4143 där man har reläer som komplement till funktionsomkopplaren och därigenom kan fjärrstyra instrumentet.

4. VAL AV KOMPARATORSYSTEM

Då man utnyttjar 4143 på det sätt som beskrivits ovan tar man ut signalerna från de två kanalerna via "External Filter"-utgångarna och man skall alltså mäta kvoten mellan dem. Eftersom vi inte har något instrument med datoranslutning som direkt kan mäta spänningskvot utnyttjar vi en vanlig digitalvoltmeter.

Den voltmeter som finns tillgänglig har 1 V som känsligaste område vid växelspänningsmätning och de spänningar man får ut från 4143 är i storleksordningen 10 mV - 1 V, beroende på mikrofontyper och kopplarvolymer. Eftersom digitalvoltmeterns upplösning utnyttjas dåligt då man mäter spänningar under några 100 mV har en mätförstärkare, Brüel & Kjaer 2636, kopplats in före voltmetern. Voltmetern kan därigenom arbeta med insignaler som ligger så att upplösningen utnyttjas optimalt.

Till mätförstärkaren är ett 100 Hz högpassfilter anslutet som externt filter. Det ersätter filtret i 4143 som ej kan användas vid denna koppling.

Valet mellan kanalerna sker med hjälp av ett relä som kan styras från datorn.

Absolutkalibreringen av förstärkare och voltmeter har mindre betydelse. De väsentliga egenskaperna är linearitet, upplösning och stabilitet.

5. PRAKTISKA ERFARENHETER

För närvarande utnyttjas en 4143 utan modifiering. Det innebär att det krävs en viss manuell övervakning under mätningarna, eftersom funktionsomkopplaren måste ställas om.

Om man mäter vid många frekvenser gör man ändå en tidsbesparing eftersom man kan låta systemet först mäta kanaldifferenser vid alla frekvenser och sedan känslighetsprodukter vid alla frekvenser. Man behöver då bara göra manuell omkoppling en gång för varje mikrofonpar.

Om man mäter ett fåtal frekvenser är tidsbesparingen däremot obetydlig. Givetvis har man fortfarande fördelen av att slippa manuella avläsningar och därmed risken för att operatören gör fel.

De värden på stabilitet som hittills kommit fram är mycket tillfredsställande. Typiska värden på medelvärdets standardavvikelse vid mätning av kanaldifferenserna är i storleksordningen 0,001 dB eller mindre.

Vid konferensen kommer ytterligare data på stabilitet och reproducerbarhet att presenteras.

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ON THE EQUIVALENT ABSORPTION AREA IN NON-SABINE ROOMS

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Introduction

In order to determine the effect of absorbing material in a room it is common to measure the reverberation time and then use Sabines formula to calculate the equivalent absorption area. The reverberation time is also used to characterize the acoustical quality of the room.

In rooms where the absorbing materials are concentrated to one surface (e.g. the ceiling) the reverberation curve will not decay linearly with time. The eigen-modes in such a room (non-Sabine) will have different decay constants which means a reverberation-time curve with a varying slope (see Fig. 1). The result will hence depend upon the portion of the decay one chooses for an evaluation of the reverberation time.

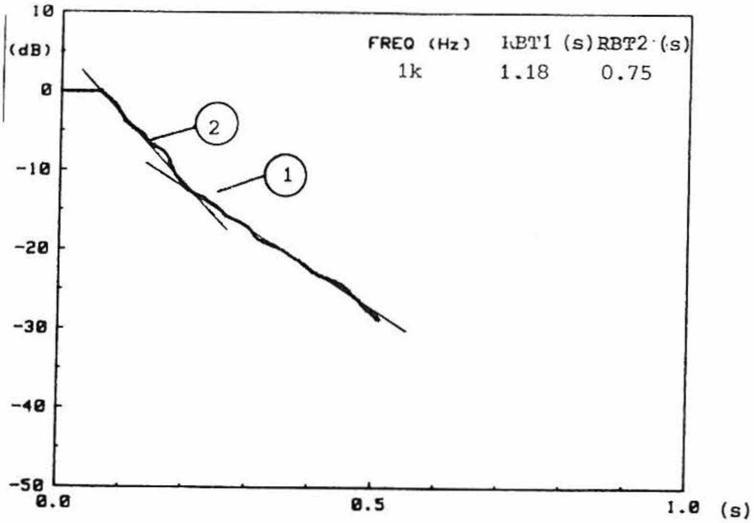
In this paper an "energy method" will be presented from which the equivalent absorption area for a room can be determined without any involvement of the reverberation time. The method is derived from Hopkins-Stryker equation [1] which like Sabines formula is based on the statistical theory.

Theory

For a reverberant sound field the following power balance is valid [2].

$$< \tilde{p}^2 > = 4\ \rho c\ W/R \qquad (1)$$

Fig. 1



where

$< \tilde{p}^2 >$	=	spatial average of the rms value squared of the sound pressure
ρc	=	characteristic impedance of air
W	=	total power supplied by the source
R	=	room constant

Expressed in levels the Hopkins-Stryker equation in its basic form, is given by

$$L_p \approx L_W + 10 \log \frac{4}{R} \tag{2}$$

where the reference value for L_p is $2 \cdot 10^{-5}$ Pa and for L_W 10^{-12} Watt. Equation (2) is valid only for air at NTP and for the aforementioned choice of reference values.

If we use an impulse with energy E and integrate equation (1) over time we get

$$< \int_0^{\infty} \tilde{p}^2 \, dt > = 4 \rho c \; E/R \tag{3}$$

This means that the room constant R can be determined if we use an instrumentation which measures the integral of the sound pressure squared and if the energy of the impulse is known.

By definition $R = S\bar{\alpha}/(1-\bar{\alpha})$ where $\bar{\alpha}$ is the average sound-absorption coefficient for the room as a whole and S is the total area of the surfaces of the room. Using $A=S\bar{\alpha}$ as the equivalent absorption area for the room

$$A = \frac{RS}{R + S} \tag{4}$$

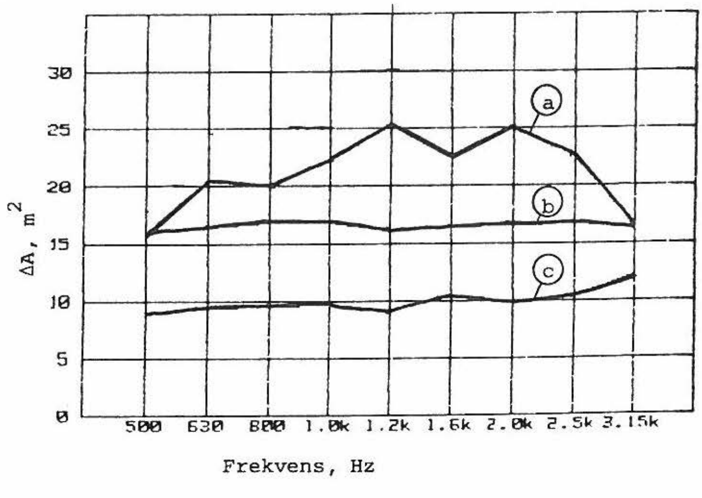
From this equivalent absorption area A, the corresponding reverberation time can be calculated.

Experiment

In order to examine the energy method proposed, the following experiments has been carried out. Throughout the experiments a start pistol was used as a sound source. The energy calibration for the pistol shot was determined from measurements in a reverberation room. The equivalent absorption area for the reverberation room was calculated from ordinary reverberation time measurements. Using a frequency analyser the integral of the sound pressure squared was measured in several positions. From equations (3) and (4) the energy for the pistol shot was determined. In order to separate the direct sound from the reverberent sound, a gating technique was used that starts the measurement with the frequency analyser first when the direct sound has passed the microphone. To avoid an influence of the reverberation of the digital filters the gate was inserted before the filters.

In a room with a volume of 104 m^3 and a total surface area of 140 m^2 the energy method was used to determine the equivalent absorption area. The equivalent absorption area was determined for the empty room and for the room with one wall covered with porous absorbing materials. The amount of added absorption area ΔA was then calculated. In Fig. 2 the results are presented in curve a. Curve b shows ΔA calculated from measurement data for the material, taken from a standard measurement in a reverberation room. From the integrated impulse response (IIR), obtained in the room with and without absorbing materials, the early decay time (EDT) was determined. ΔA was then calculated using Sabines formula. The results are presented in curve c.

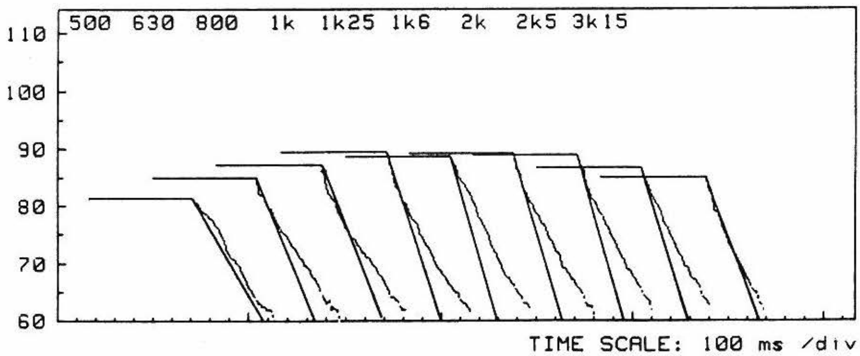
Fig. 2



The results in Fig. 2 indicates that ΔA determined with the energy method gives larger values than ΔA determined with the other methods.

Using Sabines formula and the well defined reverberation times for the empty room, the reverberation times that give the same amount of added absorption as the energy method was calculated. In Fig. 3 the corresponding reverberation curves are compared with the IIR-curves for the room with absorbents. From Fig. 3 the reverberation curves correspon-ding to the energy method seem to be tangent to the very early part of the IIR-curves. It is also worth noting that in Fig. 3 the reverberation curves bend even within the first 10 dB decay.

Fig. 3



Conclusion

The experiments indicate that measurements with the energy method give values of the equivalent absorption area that correspond to the very early part of the reverberation curve. Further, the measurement technique is practically useful and the gating arrangement makes it possible to separate the direct sound from the reverberant sound so that the influence of the room on the decay of the sound field can be extracted.

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HOW TO CONSTRUCT SIMPLE LOW FREQUENCY TRAPS

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Introduction

A hollow, boomy sound is often a problem in talk studios, sound control- and listening rooms and other small but acoustically important rooms. Room resonances are often blamed but in many cases the phenomenon is simply due to poor overall room absorption at lower frequencies. Whatever the cause, different types of resonating structures, such as membrane or Helmholtz resonators, are usually prescribed as a cure. These, however, are difficult to design and are quite expensive.

The most troublesome frequency range is often around 80...250 Hz. Above this range sufficient (often too much!) absorption can be achieved by using layers of mineral wool of reasonable thickness. Below this range windows and panels (plaster board etc.) used to provide sound insulation provide a basic level of absorption. The fundamental resonance frequency of these type of structures is typically about 50 Hz. In addition, the psycho-acoustic importance of the lowest frequencies is small (they do not contain speech frequencies, for example).

The following is a description of some simple and fairly inexpensive absorption- or trap structures which can be applied over the critical frequency range mentioned above.

Basic trap

The basic idea is to make "absorption packs" of suitable size and thickness from porous mineral wool panels and position them in the room, taking into account both acoustical and architectural considerations. The design of an absorption pack is based on the curves shown in Fig. 1.

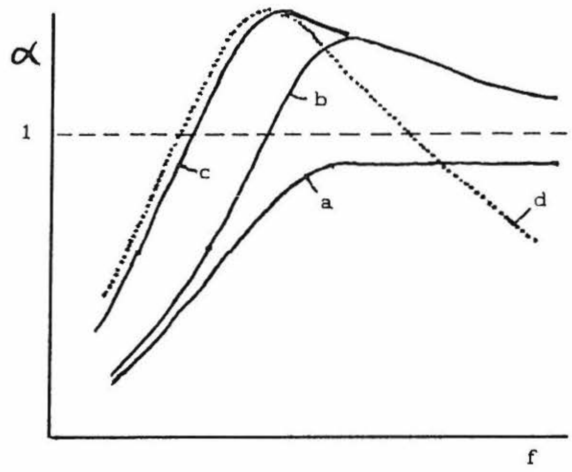

Figure 1 - Absorption coefficient of a mineral wool surface

Curve a. shows the absorption coefficient of a mineral wool panel of infinite surface area as a function of frequency. Above a certain frequency (fo) the absorption coefficient is practically constant with a value of about 0,8. Below this frequency absorption drops off rapidly. The thicker the layer of mineral wool the lower the value of fo. In the case of a finite surface the absorption coefficient determined for the geometric surface area is increased by the so-called edge effect caused by diffraction. For fairly small surfaces measured values clearly exceeding unity are obtained. The effect is greater at low frequencies, which have long wavelenghts. Curve b. in Fig. 1 shows the absorption coefficient of a fairly small mineral wool surface located in the centre of a room wall. Curve c. shows the effect of positioning the mineral wool in the corner formed by two walls, where the sound pressure has a maximum. Measurements made in a reverberation room show that the frequency range over which efficient absorption occurs is extended downwards by about one octave by the use of corner positioning. Positioning in a corner formed by three surfaces usually does not lead to any further improvement in absorption. If the mineral wool is wrapped in plastic film the absorption at higher frequencies is impaired (which is usually a desirable thing) and absorption at low frequencies improves slightly due to membrane resonance (curve d. in Fig. 1). Wrapping in plastic film also prevents the escape of dust and fibres.

Mineral wool panels are normally 60 cm x 120 cm in size. If three such panels of thickness 10 cm are stacked on top of one another the pack formed exhibits an absorption maximum in corner placement which falls more or less in the centre of the frequency range discussed above (typically around 160 Hz). Practical measurements have shown that the effective absorption area (A) of a package with a net surface area of less than 1 m2 may be greater than 3 m2. The quality of the mineral wool does not seem to have any appreciable effect, and no advantage is gained by using heavy and expensive wool.

Applications

The most practical location for absorption packages is above an acoustically transparent false ceiling (mineral wool, perforated board, grid etc.). The cross section shown in Fig. 2 illustrates a typical application. If the room contains panel walls, or if there are plenty of other light panel structures, about four "standard" absorption packs as described above will be sufficient to remove boominess from a small studio (ca 60 m3) in most cases. If the walls are heavy concrete more absorption packages will be needed (eight for example).

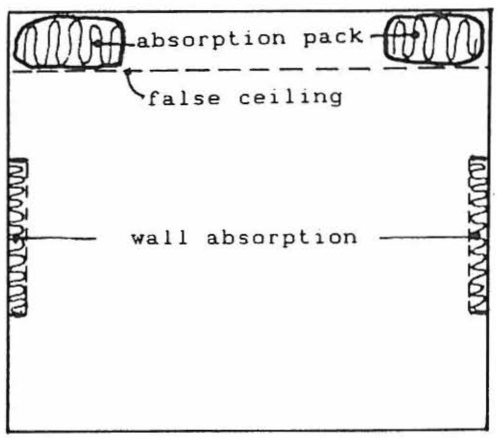

Figure 2 - Location of absorption packages above a false ceiling

Absorption structures could also be located in the vertical corners of the room, possibly integrated with wall cupboards. One type of corner trap structure suitable for use in a stereo listening room is shown in Fig. 3. The corners at one end of the room have been "cut" at an angle of about 30° by filling them with mineral wool to provide a backsurface with efficient absorption for the stereo loudspeakers. This may be beneficial to loudspeaker reproduction (some of the primary reflections disappear). The loudspeakers could also be sunk into the absorbant surface.

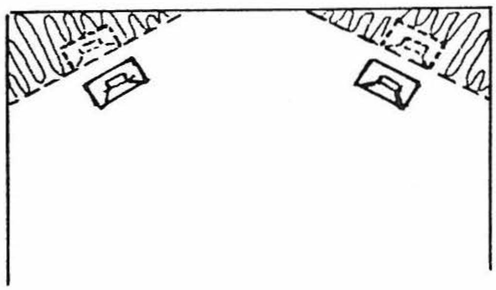

Figure 3 - Corner traps for stereo loudspeakers

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ATTENUATION OF SOUND REFLECTIONS DUE TO DIFFRACTION

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1. Introduction

Freely suspended reflectors are often used in room acoustics; however, the sound reflections from such surfaces with free edges can be attenuated considerably due to diffraction. Similar problems are met in geometric models for calculation of noise levels where reflecting surfaces of finite size are involved.

A general survey of the attenuation due to distance, absorption and curvature of the surface is found in ref. [1] or ref. [2]. In the following, only the attenuation due to diffraction is considered, i.e. the surface is assumed to be totally reflecting and plane.

2. Kirchhoff-Fresnel approximation to diffraction

A spherical sound wave emitted from a point source Q is reflected from a surface of finite dimensions as shown in fig. 1. Generally, the reflected sound field at the receiver P can be calculated from a surface integration covering all parts of the reflecting surface. Using the Kirchhoff-Fresnel approximation it appears that the intensity of the reflected sound can be expressed as a reflection coefficient K multiplied by the intensity reflected from a corresponding infinite surface, ref. [3]. Considering a rectangular surface the attenuation due to diffraction is:

$$\Delta L_{diffr} = 10 \log K = 10 \log(K_1 \cdot K_2) \quad (1)$$

where K_1 and K_2 are reflection coefficients referring to each of the two

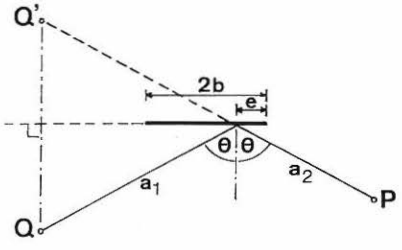

Fig. 1 *Section through reflecting surface showing the projection of source Q and receiver P. Q' is the mirror source.*

ortogonal sections through the surface. Thus, the two sections can be treated independently.

For the section shown in fig. 1 the coefficient K_1 describing the deviation from geometric acoustics can be deduced from ref. [3]:

$$K_1 = \frac{1}{2}\left[(C(v_1) + C(v_2))^2 + (S(v_1) + S(v_2))^2\right] \qquad (2)$$

where

$$v_1 = \sqrt{\frac{2}{\lambda}\left(\frac{1}{a_1} + \frac{1}{a_2}\right)}\cdot e\cdot\cos\Theta \ , \quad v_2 = \sqrt{\frac{2}{\lambda}\left(\frac{1}{a_1} + \frac{1}{a_2}\right)}\cdot(2b-e)\cdot\cos\Theta$$

λ is the wavelength and the other symbols are defined in fig. 1. C and S are the Fresnel integrals:

$$C(v) = \int_0^v \cos\left(\frac{\pi}{2}z^2\right)dz \ , \quad S(v) = \int_0^v \sin\left(\frac{\pi}{2}z^2\right)dz \ .$$

However, the general solution (2) is not easily used for practical applications.

3. Reflection at the centre of a surface

Considering the special condition $e = b$ it follows that $v_1 = v_2 = x$ and from (2):

$$K_{1,\text{centre}} = 2[C^2(x) + S^2(x)] \qquad (3)$$

where

$$x = 2b\ \cos\Theta/\sqrt{\lambda a^*} \qquad (4)$$

and the characteristic distance a^* is introduced:

$$a^* = 2a_1\ a_2/(a_1 + a_2)\ . \qquad (5)$$

The result (3) is shown graphically in fig. 2, and it appears that for low frequencies ($x < 0.7$) the diffraction gives rise to attenuation of the reflection. A very good and surprisingly simple approximation is:

$$K_{1,\text{centre}} \simeq 2x^2 \quad \text{for} \quad x \leq 0.7\ . \qquad (6)$$

For higher frequencies some fluctuations appear due to the Fresnel zones. For very high frequencies these fluctuations decrease around the asymptotic value corresponding to an infinite surface: $K_{1,\text{centre}} \to 1$ for $x \to \infty$.

4. Reflection at the edge of a surface

Another special condition, $e = 0$, leads to $v_1 = 0$, $v_2 = 2x$ and from (2):

$$K_{1,\text{edge}} = \frac{1}{2}\left[C^2(2x) + S^2(2x)\right]\ . \qquad (7)$$

This result is very similar to that above, (3), and it is also shown in fig. 2. The approximations yield:

$$K_{1,\text{edge}} \simeq 2x^2 \quad \text{for} \quad x \leq 0.35 \qquad (8)$$

$$K_{1,\text{edge}} \to \frac{1}{4} \quad \text{for} \quad x \to \infty\ .$$

Thus, at high frequencies the reflection from the edge is attenuated about 6 dB relative to reflection from an infinite surface.

5. Approximations for practical use

Considering the sound reflection in general, the distance from the geometric point of reflection to the nearest edge is denoted by e, as shown in fig. 1. Depending on the value of x it is necessary to distinguish

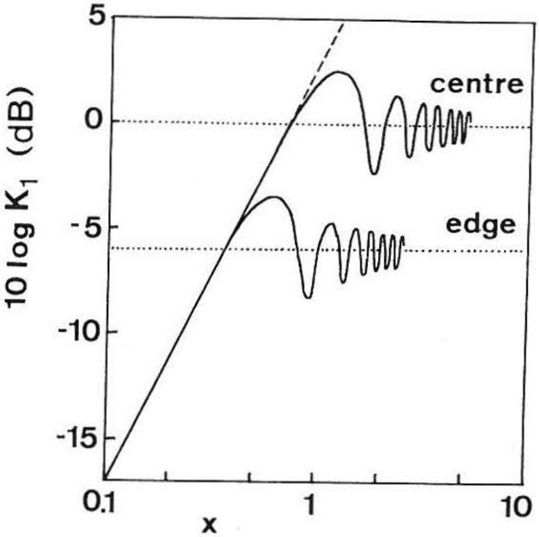

Fig. 2 *Attenuation of reflection due to diffraction. Upper curve: Reflection at centre (3). Lower curve: Reflection at edge (7). Dashed line: Approximation for low frequencies (6) and (8).*

between three regions.

a) $x \leq 0.35$: $K_1 \simeq 2x^2$, (9)

i.e. independent of the value of e.

b) $0.35 < x \leq 0.7$:

$$K_1 \simeq \frac{1}{4} + (e/b)(2x^2 - \frac{1}{4}) \qquad (10)$$

using linear interpolation between the approximate values.

c) $x > 0.7$: In this region the concept of an edge zone is introduced. A measure for the width of the edge zone is e_0:

$$e_0 = \frac{b}{\sqrt{2}\,x} = \frac{1}{\cos\Theta}\sqrt{\frac{1}{8}\lambda a^*} \,. \qquad (11)$$

If $e > e_0$ the reflection can be treated geometrically with reasonable accuracy, but if $e < e_0$ the attenuation due to diffraction must be taken into account. Thus, the following approximations are suggested for $x > 0.7$:

$$K_1 \simeq \begin{cases} 1 & \text{for } e \geq e_0 \qquad (12a) \\ \frac{1}{4} + \frac{3e}{4e_0} & \text{for } e < e_0 \,. \qquad (12b) \end{cases}$$

The simple approximations (9)-(12) have been compared with the more exact solution (2) for a number of values of x, and the agreement has been found to be good for practical use, see fig. 3.

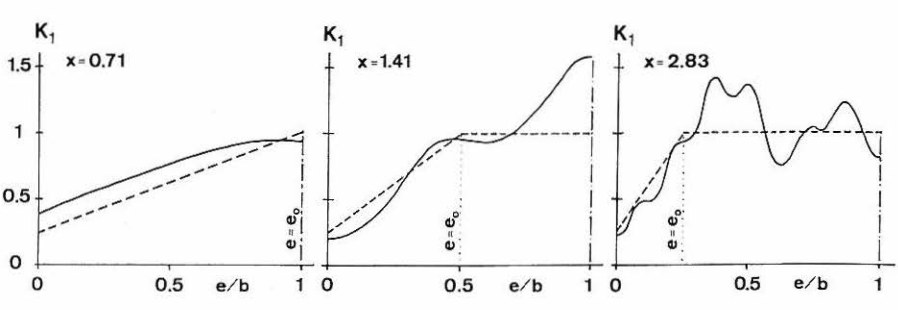

Fig. 3 *Calculated values of K_1 as a function of the distance e from the edge to the geometric point of reflection. —— : Exact (2), - - : Approximations (10)-(12).*

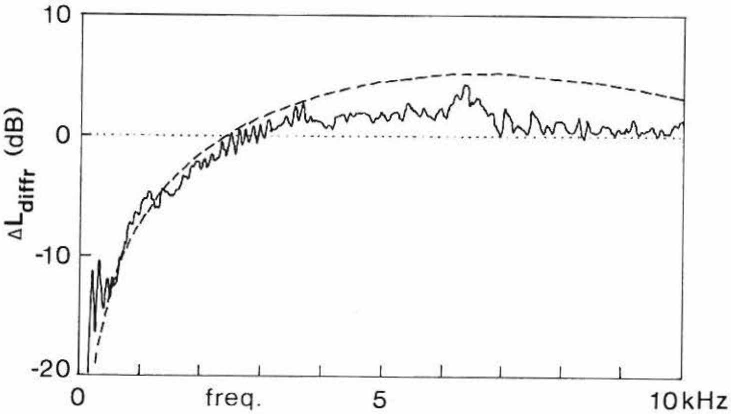

Fig. 4 *Measured and calculated attenuation of sound reflection from a square surface. ——— : Measured, - - : Calculated.*

6. Experimental results

Measurements of sound reflection were carried out in a large anechoic chamber using impulse gating technique and an FFT-analyzer B & K 2033. An example is shown in fig. 4. The measuring object was a 22 mm hardboard plate 0.60 m x 0.60 m. Distance to source a_1 = 6.00 m, distance to microphone a_2 = 4.00 m and angle of incidence $\Theta = 0^o$. The geometric point of reflection was at the centre of the surface. The attenuation of the sound reflection was found using a free-field measurement in the total distance $a_1 + a_2$ = 10.00 m as a reference.

In fig. 4 the measured attenuation is compared with the calculated attenuation due to diffraction using (1), (3), and the fact that $K_1 = K_2$ in this case. The results seem to be in good agreement, especially at lower frequencies. At higher frequencies the reinforcement is not so pronounced as expected from theory. This makes the proposed approximations even better.

7. Conclusion

The attenuation of a sound reflection from a hard surface with one or more free edges can be described by a special reflection coefficient due to diffraction. It has been shown that reflections near an edge can be treated easily by introducing an edge zone.

It follows from both theoretical and experimental results that the attenuation due to diffraction is of minor importance above a limiting frequency: $f > \frac{1}{2}ca^*/(2b \cos \Theta)^2$, corresponding to $x > 0.7$. It should be noted that the limiting frequency found here is one octave lower than the very often quoted result in ref. [4].

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PREFERRED DISTANCE TO A WALL BEHIND TALKERS

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Introduction

The investigation presented here is an extension of a previous paper concerning the acoustical conditions preferred for talkers presented at the 12th International Congress on Acoustics ICA-86 [1]. In the previous experiment the talking comfort of six different synthetic sound fields was judged by 10 talkers. The sound fields were simulated for rooms of different cross sections with no reflections in front of or behind the talkers. The results showed that the lateral-vertical ratio does not have the same important influence on talking comfort as on listening quality in auditoria. In this new experiment a wall behind the talker has been added and the distance to the back wall has been varied.

In all previous experiments with musicians and singers [2, 3, 4] only a very limited number of early reflections were simulated. This limitation will probably cause the test to be "oversensitive", i.e. the subjects will perhaps detect differences between simulations that will never occur in reality. In real sound fields the great number of reflections will in some cases mask differences that would be detected if for instance, only reverberation and the first order reflections were present. Therefore all early reflections with a delay time of up to approx. 125 ms were simulated in our tests.

Stimuli

One of the cross sections from the previous study ($w \times h = 10 \times 10\ m^2$) was chosen. In this section the source-receiver position was asymmetrical as indicated in fig. 1. The cross section was left invariant and only the distance to the wall behind the talker was varied according to table 1 and fig. 1.

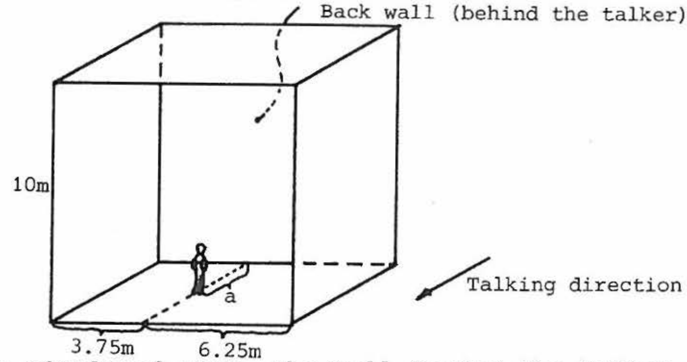

Figure 1. The simulated room. The wall facing the talker was totally absorptive ($\underline{r} = 0$).

Stimuli	1	1.5 m
	2	3 m
	3	6 m
	4	12 m
	5	∞ (no wall)

Table 1. Distance a in fig. 1.

Since we were only interested in the early reflections the front wall was eliminated from this study. In practice the front wall (facing the talker) is often so far removed that its reflections are part of the general reverberation. Although no reverberation was added the stimuli are relevant as a rough approximation of the "theatre situation" with reflecting scene surfaces and an auditorium with a very short reverberation time. The remaining five surfaces were totally reflecting ($\underline{r} = 1$).

Stimuli, containing reflections with a delay time of up to 125 ms, corresponding to approx. 100 image sources, were calculated. In order to avoid rebuilding the simulator, the lower half plane image sources were created by reflection in a reflecting floor plane. This introduces some errors in the lower half plane reflections. The actual propagation path lengths become slightly longer than the theoretical, however these errors are probably negligable compared with the differences between stimuli.

Simulation of stimuli

The speech signal from the talker was picked up using a directional microphone (AKG 451 & CK1) 50 cm away from the mouth at an angle of approx. 90° laterally and 45° vertically. The signal from the microphone entered a 1/3-octave equilizer, to compensate the loudspeaker frequency response, and was then fed to the delay unit. From the delay unit the 13 delayed signals entered the mixer. In the simulations 29 output signals from the mixer were amplified and fed to 29 loudspeakers. The overall frequency response of the system was within ± 2 dB from 100 Hz to 5 kHz.

Experiments

Because of the rather small perceptible differences the method of paired comparisons was chosen. Fifteen trained talkers were used, 9 male and 6 female, varying from 20 to 50 years of age. The subjects were told that they were standing in a lecture hall with appr. 200 - 300 listeners and were asked to judge the "talking comfort". They were allowed to talk for unlimited time and were forced to choose one of the alternatives of the pair (A, B). The pairs were presented in random order and four replications (including AB and BA) were used. This gave totally 40 comparisons per subject equivalent to approx. 45 minutes test time. The test was therefore subdivided in 3 blocks.

Statistical analysis

When trying to scale the judgements according to the assumptions of Thurstoneś case V [6, 7, 1] two groups with different mean values of the discriminal differences were detected. Therefore these groups were also analysed separately giving two different preference scales as shown below. Nevertheless the χ^2-test showed that the response variable was one-dimensional as is assumed in the Thurstone case V model.

Results

The significant differences obtained are indicated in table 2, 3 and 4. A X in the matrixes indicates a

	1	2	3	4	5
1					
2	X		X		
3	X				
4	X				
5	X	X	X	X	

Tab.2. All 15 subjects

	1	2	3	4	5
1					
2	X		X		
3	X				
4	X				
5	X	X	X	X	

Tab.3. Group 1 (10)

	1	2	3	4	5
1		X	X		
2					
3					
4		X			
5				X	

Tab.4. Group 2 (5)

significant (5 %) preference for the column stimulus compared with the row stimulus. From the preference scales in figs 2-4 we can observe that subject group 1 judged stimulus 1 to have the best "talking comfort" while subject group 2 thought this stimulus was the worst case.

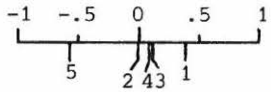

Fig. 2. Pref. scale for all (15) subjects

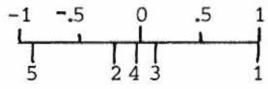

Fig. 3. Pref. scale for group 1 (10 subj.)

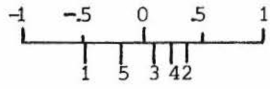

Fig. 4. Pref. scale for group 2 (5 subj.)

Discussion

The opposite opinions concerning stimulus 1 shows that the optimal distance to the wall is different for different groups. Probably this is mainly caused by different speech levels. However, the "average subject" seems to prefer the shortest distance to the back wall which gives high level and more correlated reflections (the two image source planes are close together).

Acknowledgement

The author wish to thank M. Kleiner and J. Kirszenstein for useful discussions and also A. Granhäll and B. Wijk for technical assistance.

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SUBJECTIVE SURVEY OF ACOUSTIC CONDITIONS ON ORCHESTRA PLATFORMS

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Introduction

After having worked on the problem of musicians' acoustic conditions for a number of years, we have recently conducted a questionnaire survey among orchestra players during rehearsals in nine Danish concert halls. The aims of this survey were the following:

1. to test the validity in 'real life' of laboratory results [1] concerning sound field properties (objective parameters) responsible for musicians' judgements of room acoustic quality.
2. to find the dimensionality of these judgements, i.e. the number of relevant independent subjective parameters.
3. to get an impression of musicians' evaluation of Danish halls used for symphonic concerts.

Method

Questionnaires The subjective evaluation was registered on questionnaires containing scales for a number of subjective aspects (see Table 1). According to an earlier study [2] these aspects cover the main factors in musicians' room acoustic concern - apart from background noise and echoes. (These mere faults were also included in the survey, but will not be dealt with in this paper.)

Subjects and Orchestras In each of the nine halls the questionnaires were filled in immediately after the rehearsal by about 20 musicians evenly distributed in the orchestra.

To cover the nine halls it was necessary to cooperate with three orchestras: The Sealand Symphony Orchestra (covering 6 halls), The Danish Radio Symphony Orchestra (2 halls) and Aalborg Symphony Orchestra (2 halls). The need to employ more orchestras may have introduced an 'orchestra effect' which, unfortunately, could only be weakly estimated since, only in one case, two orchestras played in the same hall (the Sealand and Radio Orchestras in the Tivoli hall: TI).

Objective Measurements Acoustical data for the nine empty halls were already available from an earlier objective survey of 21 Danish halls [3].

The objective parameters used to characterize the acoustics of the orchestra platforms had been selected on the basis of the subjective laboratory experiments already mentioned [1]: Reverberation Time (RT), Early Decay Time (EDT), Point-of-gravity Time (t_s), Clarity (C), and Early Ensemble Level (EEL) were all measured on the platform five to eight m from the sound source. Besides, Clarity (CS) and Support (ST1 and ST2) were measured with a source/microphone distance of one meter. At that distance the direct sound is so dominating that CS = $10 \log[E(0\text{-}80\ \text{ms})/E(80\ \text{ms}\text{-}\infty)] \approx 10 \log[E_{dir}/E(80\text{-}\infty)] = -10 \log[E(80\text{-}\infty)/E_{dir}]$, i.e. CS is a measure of the reverberation level relative to energy emitted (proportional to the direct sound level measured at a fixed distance). Likewise ST1 = $10 \log[E(10\text{-}100\ \text{ms})/E_{dir}]$ and ST2 = $10 \log[E(10\text{-}200\ \text{ms})/E_{dir}]$ measure the level of early reflections and early reflections plus early reverberation, respectively.

To describe the tonal character of the hall the low/high frequency ratio of EDT was formed: EDTF = [EDT(250 Hz)+EDT(500 Hz)]/[EDT(1 kHz)+EDT(2 kHz)].

Results

Dimensionality of Judgements An analysis of variance indicated substantial differences in judgements between subjects in the same hall, and factor analyses showed a substantial difference in dimensionality of the data before and after averaging over subjects in each hall. However, these differences between individuals/groups in the orchestra remain to be analysed and the following will concentrate on the subject averaged data corresponding to one 'orchestra judgement' for each hall.

A two factor space fits the averaged data very well (explain 91% of the variance), which means that 'an orchestra' judges hall acoustics in only two dimensions. Factor 1 (76% of the variance) is closely connected to all aspects in the questionnaire except Timbre which comprises factor 2 (15%), i.e. judgements on all scales except Timbre are closely interrelated. It is interesting to note that exactly the same factor pattern was found in the laboratory study ([1], section 4.3) which also predicted a lack of distinction between soloist (support, reverberance, dynamics) and ensemble judgements ([1], chapter 5).

Correlations with Objective Data The immediate result of correlating the subject averaged responses and the position averaged objective parameter values indicated only few significant relationships of which the most important was the perception of reverberance being related to RT and EDT (r being 0.63 in both cases).

However, plotting the subjective data against objective parameters, with which a certain relationship had been expected, often revealed pronounced outlier tendencies as shown in figures 1 and 2. In both cases it is seen that a rather firm linear relationship is predicted by all data points except one point: 'AH' in Fig. 1 and 'AS' in Fig. 2. These two halls happen to be the ones judged by the Aalborg Symph. Orch., which suggests the existence of a serious orchestra effect. Concerning the two other orchestras - both resident in Copenhagen - their judgements of the TI hall do not differ noticeably. This leads to the plausible hypothesis that the room acoustic frames of reference might be particularly different between the Aalborg and the two Copenhagen orchestras. (The subjects were instructed to interpret the scale extremes as experience limits.)

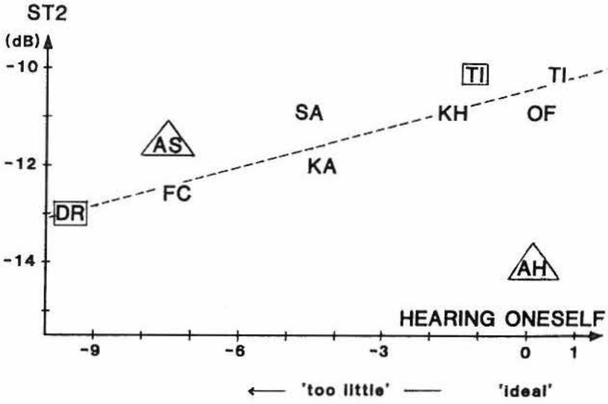

Fig. 1 *Ease of hearing oneself versus ST2 for nine halls. '∆' indicates subjective judgement by the Aalborg Symph. Orch., '□' judgement by the Danish Radio Symph. Orch. - the others were judged by the Sealand Symph. Orch. Units up the x-axis refer to mm in the questionnaire scale. The regression line for all data except the AS and AH halls has been drawn dashed. r(all data) = 0.38; r(all -AH,AS) = 0.87.*

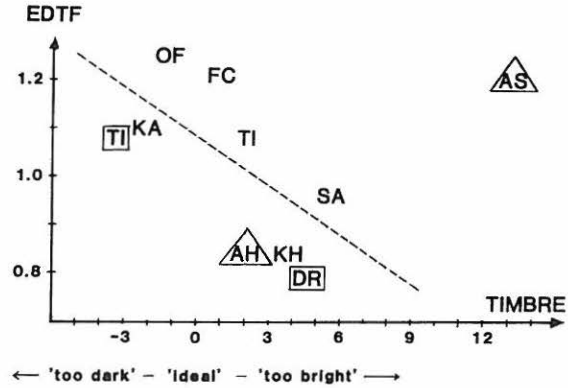

Fig. 2 *Timbre versus EDTF for nine halls. r(all data) = -0.11; r(all -AH,AS) = -0.66. See also legend to Fig. 1.*

Consequently, it seemed natural to analyse the data of the Copenhagen orchestras separately which resulted in a number of significant correlations as shown in Table 1. It is noted that the subjective responses correlate mainly with those parameters which measure reflected energy relative to the emitted energy (the direct sound): CS, ST1 and ST2.

The correlation patterns for the ensemble and the support judgements are very similar as expected from the factor analysis results. The fact that the highest correlations occur with ST2, i.e. with the level of early reflections plus early reverberation (≈ total energy) suggests that hearing oneself/support may have been the aspect(s) focussed on among this sub-group of factor 1 aspects.

Reverberance and dynamics comprise another factor 1 sub-group which is - not unexpectedly either - related to the reverberation level: CS.

It is very interesting that Timbre (= factor two) shows some correlation with nothing but EDTF. This demonstrates that this factor was not just statistical noise. This had been feared since Timbre was not significant - and the only scale not being so - according to the analysis of variance.

Subjective aspects:	Objective parameters								
	RT	EDT	t_s	C	EEL	CS	ST1	ST2	EDTF
Ensemble, generally						-0.68	0.69	0.82	
Ensemble, hearing others							0.69	0.77	
Ensemble, hearing oneself						-0.67	0.76	0.91	
Support	0.64					-0.66	0.67	0.81	
Reverberance		0.71				-0.92		0.63	
Timbre									-0.69
Dynamics						-0.93		0.66	

Table 1 *Correlation coefficients between subjective evaluations and objective measurements for the halls evaluated by the Danish Radio and the Sealand Symphony Orchestras. Only coefficients significant at a 10% level are shown; 5% levels have dashed underlinings, 1% full underlining.*

Optimal Parameter Values From significant regression plots like Figs. 1 and 2 it is possible to predict objective values, which will be judged as'ideal' or on optimal compromise by the (Copenhagen) symphony orchestras since 'ideal' corresponds to the value 0 on the subjective scales. This results in the following set of optimal values for the significant objective parameters:

$$EDT \approx 1.4\ s, \quad CS \approx 13\ dB, \quad ST2 \approx -10\ dB, \quad EDTF \approx 1.1\ .$$

Concluding Remarks

There may be substantial differences in judgements of room acoustic quality within an orchestra as well as between different orchestras. Still, it has been found that averaging over individuals results in a general orchestra consensus which can characterize differences between halls, and - even better - these hall differences show surprisingly high correlations with certain objective parameters.

The positive results of this field experiment are fully in accordance with the earlier laboratory results. Only was it disappointing to see that no correlations with EEL appeared as it had done in the laboratory study. One reason may be that the ease of ensemble judgement was subordinated the impression of support, although this was not expected to be the case for orchestra players according to [2].

Acknowledgements

The members and staff of the three orchestras deserve warm thanks for their most willing participation in this survey. Expenses in connection with the project have been covered by a grant from 'A.N. Neergård og Hustrus Fond'.

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AKUSTIKEN I STOCKHOLMS FYRA KONSERTSALAR - EN JÄMFÖRELSE

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Den akustiske konsultens ständiga fråga är vilka projekteringsmetoder som skall användas då t ex en konsertlokal skall byggas? Vilka mätmetoder skall användas i den färdiga lokalen? Den stora, och i sammanhanget intressanta, frågan är då givetvis hur den aktuella projekteringsmetoden stämmer med den subjektiva upplevelsen av akustiken i den färdiga lokalen.

För att utröna detta har Ingemansson Akustik i samarbete med Kungliga Tekniska Högskolan i Stockholm, som examensarbete gjort en undersökning i fyra av Stockholms konsertlokaler, nämligen Berwaldhallen, Cirkus på Djurgården, Nacka Aula samt Stockholms Konserthus. Arbetet har utförts av Anna Pålsson och Anne Wetserlund med Ulf Rosenberg som handledare från Ingemansson Akustik.

Som projekteringsmetod i undersökningen användes strålgångssimulering med dator. Med hjälp av detta har sedvanliga egenskaper såsom efterklangstid, deutlichkeit, clarity och så vidare beräknats. Mätningarna i färdig lokal har dels omfattat sedvanliga efterklangstidsmätningar dels pulsmätningar. Dessa har utvärderats med FFT-analys och som pulssvar på oscilloskop.

En konsertlokal skall inte bara vara bra för publiken. Den är arbetsplats för ett hundratal musiker och det är väsentligt att arbetsmiljön för dessa är så god som möjligt, inte minst akustiskt. Musikerna måste ha rimliga möjligheter att uppfatta sitt eget spel i relation till den övriga orkestern, både de individuella stämmorna och den totala klangen. Det missas ofta att detta faktiskt är viktigt även för publikens musikupplevelse. Om arbetsförhållandena för musikerna inte är optimala kan de inte heller spela optimalt, något som givetvis negativt påverkar publikens upplevelse. För att även få med dessa förhållanden i undersökningen så lades provytor och mätpunkter in även på podiet.

För att få ett visst grepp om den i sammanhanget viktigaste biten - nämligen musikernas och publikens omdömen om de olika lokalerna - så gjordes också en mindre intervjuundersökning. Denna undersökning begränsades till personer med anknytning till och erfarenheter av de olika lokalerna. Fyra kategorier intervjuades, musiker i de aktuella orkestrarna, recensenter, ljudtekniker med vana vid de olika lokalerna samt musikadministratörer med anknytning till lokalerna samt medlemmar i orkestrarna.

Urvalet av lokaler för undersökningen var lätt. Samtliga större konsertlokaler i Stockholm skulle ingå. Det rör sig om totalt fyra salar, nämligen:

Berwaldhallen som invigdes 1979, är Sveriges Radios konsertlokal/studio och Sveriges Radios Symfoniorkesters ordinarie hemvist. Akustisk konsult var Vilhelm Lassen Jordan. Lokalens volym är 13.000 kubikmeter och den rymmer 1.300 publikplatser.

Cirkus på Djurgården - färdigställd 1892 - som just cirkuslokal. Under ett flertal år före invigningen av Berwaldhallen, användes lokalen för konserter och inspelningar av Radions symfoniorkester. Lokalens volym är 9.500 kubikmeter och den inrymmer cirka 3.000 sittplatser.

Stockholms Konserthus invigdes 1926 och disponeras av Stockholms filharmoniska orkester. I början av sjuttiotalet genomgick lokalen en genomgripande ombyggnad då bland annat ett helt nytt podiehus byggdes. Akustisk konsult vid denna ombyggnad var Stellan Dahlstedt. Lokalen har en total volym på 10.000 kubikmeter och rymmer cirka 1.800 personer.

Nacka Aula är samlingssal till Eklidens Skola i Nacka. Den är ritad som skolaula i första hand. Den har dock blivit en synnerligen uppskattad konsert- och inspelningslokal. Salen blev färdig i början av 1960. Akustiker var Ove Brandt. Lokalens volym är 5.800 kubikmeter och den rymmer drygt 600 sittplatser.

Sammanfattning

Efterklangstiden för två av lokalerna, Berwaldhallen och Stockholms Konserthus ligger strax under 2 sekunder. För de båda andra ligger efterklangstiden kring 2,5 sekunder. För alla lokalerna utom Stockholms Konserthus faller efterklangstiden för frekvenser under 500 Hz. För Konserthuset är däremot efterklangstidskurvan i basen "handboksmässigt" rak.

Beträffande de subjektiva omdömena om lokalerna så varierar dessa relativt mycket mellan olika personer, delvis förmodligen beroende på personliga erfarenheter och på personliga referenser. Undersökningen gav dock ett entydigt resultat -

den enda lokalen som inte projekterats för musik rankades av nästan samtliga intervjuade som nummer ett. Rangordningen mellan de övriga varierade däremot så att ingen av dessa kan rankas som bättre eller sämre än någon av de två övriga.

Beträffande strålgångssimuleringarna och mätningarna i de färdiga lokalerna överensstämde dessa resultat relativt väl. Det framgick till exempel att det finns vissa problem i Berwaldhallen på grund av ljudkällan/orkestern där sitter nära salens mittpunkt med i stort sett lika avstånd till alla reflexytor. Detta är problem som också kan upplevas i salen. Både mätningar och strålgångssimulering i Stockholms Konserthus och Nacka Aula talade för att detta var konsertlokaler utan allt för stora brister, något som också stämmer med den allmänna uppfattning som kom fram av intervjuundersökningarna.

Beträffande Cirkus var det däremot svårt att ur mätningar och simulering få fram att det här rörde sig om Stockholms kanske bästa konsertlokal. En trolig förklaring är att strålgångssimuleringen bygger på att efterlikna resultaten från de bästa klassiska konsertsalarna med sina rektangulära former. Cirkus med sin nästan kvadratiska grundform och med sin kupol ligger långt från dessa klassiska former. Detta innebär i och för sig inget positivt och substansiellt svar på varför de nu gängse metoderna inte lyckades ge resultat som överensstämmer med musikers och publiks allmänna uppfattning. Resultatet ger dock en ny frågeställning för en kommande undersökning!

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GLASOVERDÆKKEDE RUMS AKUSTIK

Opfordring til forsøg, måling og analyse

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Glas er et unikt byggemateriale med dobbelte egenskaber. Det lukker sol, lys og syn igennem samtidig med, at det beskytter mod vind, kulde og varme. Traditionelt har glasset mest været anvendt til isætning, d.v.s. det har siddet i rammer som vinduer. Nu holder denne begrænsning ikke mere - anvendelsen af glas stiger i byggeriet.

Fra isætning til omslutning

I mange nybyggerier søges en forbedring af det fysiske miljø og en højere kvalitet. Samtidig søges løsninger som minimerer energiforbruget og giver et billigere byggesystem. Ved at anvende glas i områderne mellem bygningerne eller i facader skabes en ny type af rum. Et rum mellem hus og hus. Et rum mellem ude og inde, som hverken er ude eller inde, eller som er både udenfor eller indenfor. Et rum i glas - på en gang åbent og lukket. Glasrummet åbner for en ny energiteknik, nye byggemetoder og året-rundt-aktiviteter mellem og omkring bygningerne. Glasset ser ud til at blive lige så betydningsfuldt som omsluttende som til isætning.

Det nye

Tanken om det store glasrum er gammel. Den engelske gartner Joseph Paxton opførte Chrystal Palace (70.000 m2) til verdensudstillingen i London 1851. Det nye ligger i materialet, glasbyggeteknikken, energisparebehovet og den moderne drøm om et rigtigt og brugbart bymiljø. Nu findes hærdet, lamineret og energibesparende glas. Nu findes aluminium-konstruktioner, standardbyggesystemer, automatik for ventilation og solafskærmning. Nu findes data fra anvendelse af passive solfangere. Desuden findes der nu teorier/modeller om integrerede by-glasbyggeri. Glasrummet er kommet for at blive.

Væksthuskonceptet

De som ved mest om glashuse i dag er væksthusbyggerne. Fra deres byggesystemer og byggeerfaringer har vi hentet det bedste. Gartnerne behersker styringen af klimaet, lys, fugtighed og varme. Planter er meget sarte og må behandles som spædbørn. Kan man klare tomater og andre grøntsager under glas, så går det også med mennesker. Glas holder i århundrede, hvis det ikke udsættes for overlast. Det behøver ikke males. Og der er ingen risiko for, at det rådner, ruster, skaller eller ældes. Men - Hvordan gør vi det akustisk bedst? Et godt akustisk miljø er ofte afgørende for et glasrums komfort-vurdering, d.v.s. lavt lydniveau, tilpasset efterklangstid og lydisolering mod omgivelserne.

Beregningsmodeller til dette ønskes

Giver glasrum anledning til specielle foranstaltninger for at få et akustisk miljø? De store glasrum indeholder ofte store arealer med væsentligt hårde byggematerialer, glas, beton, fliser etc. Norges Teknisk-Naturvitenskapelige Forskningsråd (NTNF) siger direkte i bogen: "Glassgårder" "Det bør derfor vurderes å utarbeide retningslinjer for akustisk dempning i disse lokaler". I en af de internationalt oftest citerede bøger om glasbyggeri - Richard Saxon's "Atrium Buildings" - nævnes overhovedet ikke noget om akustik!!

Værktøjet

I eksisterende byggerier er benyttet mange forskellige regulerings- og tilpasningsmuligheder. Nogle med sigte på at opnå en bestemt akustisk virkning, menoftest af helt andre årsager. Mange spørgsmål kan rejses om mulige forsøg, målinger og analyser af akustik i glasrum. Efterfølgende emner er ikke prioriteret eller vurderet for deres betydning.

1) Glas som absorbent

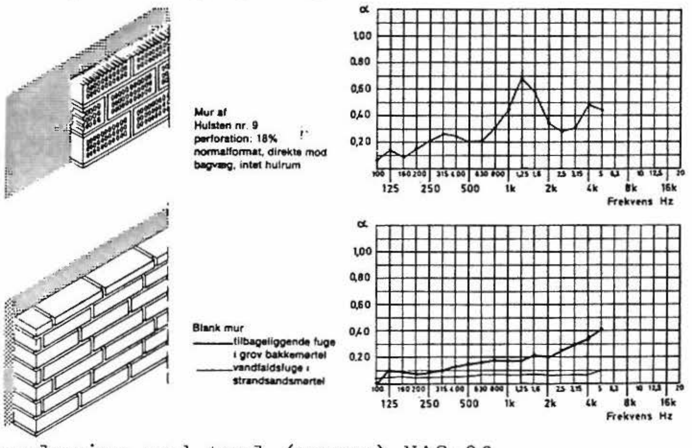
I flere anvisninger og lærebøger angives absorptionskoefficienter for vinduer og glas. Men kan disse anvendes, når det drejer sig om store glasarealer i tage eller i facader? Termoruders effekt som membranabsorbent i henholdsvis 1, 2 og 3 lags konstruktioner må variere væsentligt med glassets tykkelse og areal.

Materiale	Absorptionskoefficient ved frekvens, Hz					
	125	250	500	1000	2000	4000
Vindue med termorude opbygget af 3-4 mm glas	0,10	0,07	0,05	0,05	0,02	0,02

Uddrag fra SBI-anvisning 137: Rumakustik 1984.

2) Øvrige absorbenter

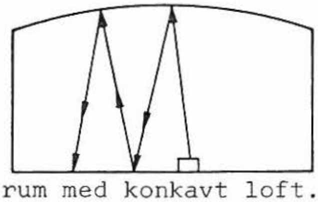
Efterklangstiden er en hensigtsmæssig målestok for rummets akustiske kvalitet. Det er for de fleste typer af rum muligt at fastlægge den efterklangstid, som vil være bedst egnet for en påtænkt anvendelse. Lydabsorberende materialer kan indbygges i bygningernes facader f.eks. i brystningsfelter, hvor mineraluld lægges bag perforerede plader af gips, stål eller hultegl. Kan eksisterende forsøgsmodeller og beregningsmetoder anvendes i glasrum for at finde ud af, hvordan rumakustikken bliver i forhold til placering og mængde af absorbenter?



Lydregulering med tegl (paper) NAS-80

3) Tagudformningens betydning.

Kan der i glasrum forekomme svagt dæmpede egensvingninger og flutter-ekko eller andre uønskede reflektioner? Glastage oplægges sjældent parallelt med modstående, reflekterende flader, men ved glasfacader ses det ofte.



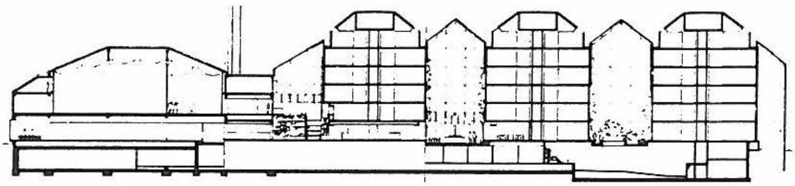
Flutter-ekko i rum med konkavt loft.

4) "Akustisk beplantning"

En velplaceret stedsegrøn beplantning anvendes i mange tilfælde til at give en akustisk dæmpning, men findes der et beregningsgrundlag for denne praksis? "Et nærmere studie af planters akustiske virkning ville kunne få stor betydning" - citat fra BUR-rapport: Glasoverdækkede uderum. Beplantningernes bunddække med brændte lette lerklinker som Leca har måske en upåagtet virkning?

5) Inder-facadens lydisolerende egenskaber

Ofte bliver rum som ligger ind mod et glasrum udstyret med kun et lag glas i eventuelle vinduer, hvis der ikke stilles brandtekniske krav. Man må derfor være opmærksom på lydgennemgang fra glasrum. Til ydervægge stilles der krav til lydreduktion, men kan kravene opfyldes, når "ydervæggen" ind mod glasrummet udføres enkelst muligt? Brug af glasfacader kan også have en positiv effekt i form af støjskærm mod veje og jernbaner. Brug af glastilbygninger foran facader vil derfor kunne være et aktuelt støjskærm-middel i forbindelse med renovering af ældre bebyggelser i støjudsatte områder.



- Ved projektering, bygning og brug af glasrum kræves indsigt i en række forhold. Man kan i lille udstrækning bygge på erfaringer, fordi glasrum hidtil ikke har været brugt i større omfang ved de klimaforhold, som hersker i Norden. Lad rummene - også - blive en akustisk oplevelse.

Aktuel litteratur

Glasoverdækkede uderum, BUR-rapport, København 85

Glassgårder, NTNF, Oslo 85

Överglassade rum, Svensk Byggtjänst, Stockholm 85

Akustisk prosjektering av Royal Garden Hotel (paper), NAS 84

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Hvordan opnås bedre lydforhold ?

Bent Christensen

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Erfaringer fra praksis viser, at akustikeren ofte først bliver hidkaldt, efter at brugeren er flyttet ind og gennem nogen tids brug af bygningen har konstateret en mangel på lydområdet.
Jeg har selv i sådanne situationer tænkt: "Hvis de havde spurgt tidligere, så kunne lydforholdene næsten uden ekstraomkostninger have været helt tilfredsstillende, men nu er opgaven næsten uløselig".

Enhver der arbejder med lyd ved, i hvor høj grad brugerne vil opleve fejl og mangler på lydområdet. Men er akustikere opmærksomme nok på, at de er de eneste der kan forebygge fejlene, og er akustikernes indsats på dette område stor nok ?

Der har i de seneste år været talt en hel del om kvalitetsstyring indenfor byggeriet. I den forbindelse fik jeg den opgave, at udarbejde en kvalitetsstyringshåndbog for den virksomhed jeg på det tidspunkt var beskæftiget i.
Resultatet blev en kvalitetsstyringsmodel med 7 faser.
Meget kort fortalt omhandler modellen følgende:

1. Behovsanalyse.

* Lovkrav.
* Brugerønsker.
* Dine erfaringer om brugerbehov.

Analysen foretages på "brugerens betingelser". Det er vigtigt,at de forventninger som brugeren først bliver bevidst om efter ibrugtagning tages med i betragtning.

2. **Objektiv kvalitetsaftale.**

* Eentydig og målbar angivelse af hvad der skal opfyldes.

Information til bruger om betydningen af den objektive kvalitetsaftale.

3. **Projektering/optimering.**

* Samkøring af kvalitetsparametre.
* Udførelsesvenlighed.
* Økonomi.

Det handler om at opnå et tilfredsstillende resultat billigst muligt.

4. **Kobling projekt/praktik.**

* Undersøgelse af prøver/prøveopstillinger.
* Information om projektideer til de udførende.
* Information om "risikoområder" til de udførende.
* Justeringer af projekt udfra den udførendes synspunkter.

Det gælder om at "fange" eventuelle fejl så tidligt som muligt, så er de billigst at rette.

5. **Aktiv kontrol.**

* Information til de udførende om audio/visuel kontrol.
* Kvalitet gennem udførendes selvkontrol.
* Løbende stikprøvekontrol.
* Afsluttende kontrolmålinger.

Det gælder om at "fange" eventuelle fejl så tidligt som muligt, så er de billigst at rette. Løbende erfaringsopsamling dygtiggørelse af medarbejderne.

6. **Brugs-/vedligeholdelsevejledning.**

* Information til brugeren om rigtigt brug.
* Information til brugeren om rigtig vedligeholdelse.

Det gælder om at bevare den opnåede kvalitet.

7. **Erfaringsopsamling.**

* Blev brugerens forventninger opfyldt ?
* Gentag successer.
* Undgå/forebyg fiaskoer/fejl.

Indsamling af erfaringsmateriale til den næste behovsanalyse.

Ved gennemlæsning af foredrag fra tidligere Nordisk Akustiske Møder kan det konstateres, at akustikerens hovedinteresse koncentrerer sig om fase 3. projektering, den del der handler om at finde tekniske løsninger på lydopgaverne.
Der er måske grund til at påpege, at den teknisk set geniale løsning først bliver noget rigtigt værd, når den anvendes i praksis.

Jeg mener , at der er et meget stort behov for, at akustikere interesserer sig lidt mere for de andre 6 faser i kvalitetsstyringsmodellen.

I det følgende vil jeg specielt beskæftige mig med fase 1. Behovsanalysen.

Lovkrav.

Det er ofte hørt, at forbedringer på lydområdet kun kan opnås gennem en skærpelse og udvidelse af lovgivningen på området.
Det givetvis rigtigt, at lovgivningen har betydning for kvalitetsniveauet, specielt hvis der gennemføres kontrol af om lovgivningen er opfyldt, og det har konsekvenser hvis den ikke er opfyldt.

Brugerønsker og akustikerens erfaringer.

Men lovgivningsmetoden er ikke den eneste vej til bedre lydforhold, det er muligt, gennem bevidsthedsgørelse af brugerne, bygherrene og deres nærmeste rådgivere arkitekterne, at skaffe lydområdet en højere prioritering.

Der er derfor behov for, at akustikeren søger indflydelse på byggeriet allerede i behovsanalysefasen.

Dette kan f.eks. opnås ved et mere snævert samarbejde med arkitekten.

Tradiotionelt er det arkitekten der har den første kontakt til brugeren/bygherren, det er arkitekten der gennemfører behovsanalysen.

Arkitekten gør normalt dette meget grundigt og godt på det visuelle område. Beskrivelser, tegninger, materialeprøver, farveprøver og eventuelt en model af bygningen giver brugeren/bygherren et klart indtryk af, hvordan huset kommer til at se ud.
Jeg er helt sikker på, at arkitekten også gerne ville lade behovsanlysen omfatte lydområdet. Jeg har været ude for flere arkitekter der er mere end villige til at tage spørgsmålet op, under forudsætning af, at behovsanalysen bliver gennemført på et for bruger/ bygherre forståeligt niveau.

Det nytter ikke at tale om R'_w værdi, $L'_{n,w}$ værdi eller T_e i fase 1. hvor man har med mennesker at gøre, der "kun" bruger lyden til at lytte på. Det er nødvendigt at "illustrere" budskabet om betydningen af gode lydforhold på en for brugeren/bygherren forståelig måde.

Efter min mening er det en meget værdig opgave for akustikeren, at være med til at formidle budskabet, enten ved at gøre det selv, eller ved at give arkitekten "værktøj" så han kan gøre det.

Jeg har forsøgvis anvendt lyd til at illustrerer lydforholdene med. Det lyder måske lidt banalt, men det har virket hver gang.

Det gør et dybt indtryk på kommende kontorhusbrugere, at høre forskellen på en skillevæg der går til undersiden af akustikloftet uden overlukning mellem akustikloftet og den overliggende etageadskillese, og en skillevæg der afsluttes imod underside af etageadsikillese eller udføres med overlukning. Det er langt billigere at lade kontorhusbrugeren høre denne forskel på planlægningsstadiet end når alle skillevæggene er opstillede og kontorhuset er taget i brug.

Og beboere i en ejendom hvor der skal skiftes vinduer ud får et bedre beslutningsgrundlag,når de inden beslutningen om hvilke vinduer og ruder der skal anvendes har hørt forskellen imellem et normalvindue og et lydvindue.

Demonstrationen med lyd har hver gang resulteret i, at den lydmæssigt bedste løsning er blevet valgt, også selv om den er lidt dyrere.

Jeg ser gode muligheder for at højne kvalitetsniveauet på lydområdet, specielt for de områder hvor der ikke er lovgivet, ved at fremstille auditive modeller af lydforholdene med forskellige løsninger.
Det er betydeligt nemmere, at fremstille en auditiv model på et lydbånd, end at fremstille en visuel model af balsatræ, pap og akryl.

Men for at fremstille den auditive model fordres indsigt med forskellige konstruktioners lydmæssige formåen.
Så vidt jeg kan se, vil det derfor være naturlig om akustikeren fremstiller disse auditive modeller.
Om det så er en akustikeropgave at foretage behovsanalysen på lydområdet, eller om man vil overlade dette til arkitekten, er ikke så væsentlig, de auditive modeller virker i sig selv meget overbevisende.

Jeg mener, at akustikerne har et ansvar for, at der bliver gennemført en behovsanalyse på lydområdet, og at akustikeren bør fremstille "værktøj" til denne behovsanalyse i form af auditive modeller.

Jeg er sikker på, at kvalitetsniveauet på lydområdet ville blive højere, dersom brugere, bygherre og arkitekter gennem sådanne behovsanalyser med auditive modeller blev bevidstgjort om lydforholdenes betydning allerede på planlægningsstadiet.

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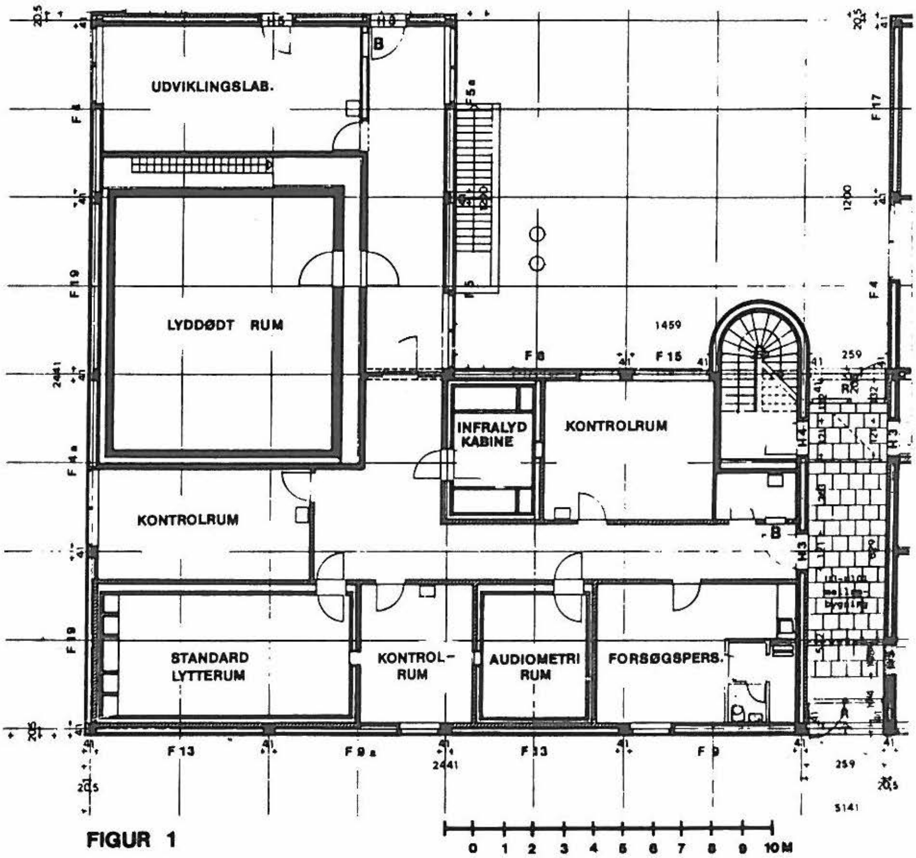
OPFØRELSE AF AKUSTISKE MÅLERUM VED AALBORG UNIVERSITETS-CENTER

JØRGEN PEDERSEN

SKANDINAVISK LYDTEKNIK A/S
SOHNGÅRDSHOLMSVEJ 2
9000 AALBORG
DANMARK

INTRODUKTION

I april måned 1986 startede opførelsen af 5. afsnit af Aalborg Univercitetscenter (AUC). Afsnittet skal huse Institut for Elektroniske Systemer, som flytter ind i august måned 1987.
Byggeriet er projekteret af arkitektfirmaet Dall & Lindhardtsen A/S, Helsingør, det rådgivende ingeniørfirma Carl Bro A/S, Glostrup og med Skandinavisk Lydteknik A/S som akustisk konsulent.
Afsnittet opføres som tre 2 etagers karrébygninger hver bestående af 4 L-formede bygninger, der er i indbyrdes forbindelse med glasoverdækkede mellemgange.
Stueetagen i en af disse bygninger indrettes med en række akustiske målerum med tilhørende kontrolrum.
Stueplanen er vist i fig. 1.
Bygningskonstruktionerne for målerummene var fastlagt ved byggeriets udbydelse i offentlig licitation.
Den akustiske aptering og alle installationer til målerummene blev imidlertid udbudt i en særlig akustikentreprise på grundlag af funktions- og materialekrav og efter gennemførelse af en prækvalifikationsrunde for indbudte entreprenører.
Målerummene, der består af lyddødt rum, infralydkabine, standard lytterum og audiometrirum er nærmere beskrevet i det følgende.



FIGUR 1

LYDDØDT RUM

Det lyddøde rum skal anvendes til målinger og forsøg inden for elektroakustik, psykoakustik, audiologi, medicinsk akustik, musikakustik og akustisk støj.
Fra brugerne blev der stillet en række akustiske funktionskrav. Rummet skulle have en grænsefrekvens, der var lavest mulig inden for det bygningsrumfang, der var til rådighed. Under normalt forekommende eksterne støjniveauer skulle baggrundsstøjen i rummet, her og i det følgende forstået som støj fra uvedkommende støjkilder, være mindre end -10 dB(A) re $2 \cdot 10^{-5}$ Pa. Dette sidste krav blev senere ændret således, at baggrundsstøjniveauet målt i 1/1 oktaver skulle ligge 10 dB under den i DS/ISO/R 226 angivne kurve for høretærskelen.
Endvidere skal rummet være vibrationsisoleret, så egenfrekvensen bliver 7 Hz $\pm$ 2 Hz.
Rummet er projekteret som en dobbeltkonstruktion med en indvendig jernbetonkasse med 25 cm vægtykkelse. Ved 2 af kassens sider er der et dæmpet hulrum på 20 cm, mens hulrummet ved de 2 øvrige sider er min. 75 cm. Mellem kassens

loft og det ydre tag, er der en fri højde på ca. 1,00 m. Herfra kan mikrofoner m.v. sænkes ned i rummet gennem 35 mm rør indstøbt i loftet. Udfra et fastlagt typisk eksternt støjspekter, er den nødvendige reduktionstalskurve og dermed konstruktionens dimensioner bestemt. Adgangen til rummet sker via dobbelte lyddøre.
Rummets akustiske beklædning bliver mineraluldskiler med en base på 23 cm x 30 cm x 30 cm og en total længde på 128 cm. Den teoretiske grænsefrekvens er beregnet til ca. 65 Hz. Rummets indre dimensioner mellem kilespidserne bliver 604 cm x 484 cm x 564 cm.
Gulvet udføres som et netgulv af ø 3 mm koldtrukket tråd og med en maskevidde på 5 cm. Over netgulvet kan udlægges et ristegulv, så tungere måleobjekter kan placeres i rummet. Yderligere kan der over ristegulvet udlægges en lydreflekterende flade, så rummet kan bruges som et semilyddødt rum.

INDFRALYDKABINE

På Institut for Elektroniske Systemer forskes der i infralydens påvirkning af mennesket. Der er derfor i det nye byggeri projekteret en kabine, hvor forsøgspersoner kan udsættes for kontrollerede infralydpåvirkninger. Infralyden frembinges af 48 elektrodynamiske højttalere, der vil kunne give et lydtryk på op til 137 dB re $2 \cdot 10^{-5}$ Pa i frekvensområdet 0-35 Hz og op til 125 dB i området 30 - 100 Hz. De 48 højttalere er placeret med 16 stk. i loftet og 16 stk. i hver af de to sidevægge. Bag højttalerne er højttalerkamre med et samlet rumfang svarende til kabinens.
Da infralydkabinen skal fungere som et trykkammer, er den udført som en 10 cm tyk jernbetonkonstruktion. For at sikre, at kabinekonstruktionen er tæt, bliver der på alle indvendige betonoverflader opsat en membran, der sikrer en akustisk resistans > 10^6 Nsm^{-5}.
Forsøgspersonerne må kun udsættes for ren infralyd, hvorfor der er krav om en meget stor lydisolation for støj i det normale hørbare område. Baggrundsstøjen skal ligge mindst 10 dB under høretærskelkurven målt i 1/1 oktavbånd. Løsningen blev en dobbeltkonstruktion, hvor kabinen er anbragt svømmende uden forbindelser til den øvrige bygning. Kabinens indvendige dimensioner er 260 cm x 240 cm x 230 cm.
Kabinen skal være meget dæmpet, og der er stillet krav om, at middelabsorptionskoefficienten i frekvensområdet 31,5 - 4000 Hz skal være > 0,4, svarende til en beregningsmæssig middelefterklangstid på 0,2 sek. Lydabsorbenterne udføres som stofbetrukne rammekonstruktioner og i en kombination mellem høj- og mellemfrekvensabsorbenter og lavfrekvensabsorbenter.

STANDARD LYTTERUM

Standard lytterummet skal anvendes til forsøg med og afprøvning af højttalere.
Rummet er projekteret efter de anvisninger, der er givet i: "Draft: IEC Publication 268-13: Sound System Equipment; Part 13: IEC Report on Listening Test on Loudspeakers".

Derudover har brugerne stillet krav om, at baggrundsstøjen målt i 1/1 oktaver skal være lavere end høregrænsekurven efter DS/ISO/R 226.
Rummet udføres som en svømmende dobbeltkonstruktion med vægge af pudset murværk. Rågulv og -loft udføres i jernbeton. Dimensionerne på det nøgne rum bliver 779 cm x 413 cm x 280 cm. I den ene endevæg udføres der 5 nicher til placering af højttalere.
Loftet nedhænges og udføres i felter, hvor der kan anbringes højttalere og forskellige absorberende loftplader.
Gulvet bliver et trægulv på strøer.
Efter færdigindretning af rummet foretages efterregulering af efterklangstiden til følgende værdier:

100 Hz	125 Hz	160 Hz	200-4000 Hz
0,55±0,20 sec.	0,5±0,15 sec.	0,45±0,10 sec.	0,4±0,05 sec.

AUDIOMETRIRUM

Audiometrirummet skal benyttes til såvel frifelt- som hovedtelefonaudiometri.
Kravet til baggrundsstøj er, at denne målt i 1/1 oktavbånd skal ligge mindst 10 dB under høregrænsekurven.
Rummet skal være meget dæmpet, og kravet er en middelabsorptionskoefficient > 0,4 i frekvensområdet 31,5 - 4000 Hz. Rummet udføres som en svømmende dobbeltkonstruktion i pudset murværk.
Rummets dimensioner vil blive 334 cm x 393 cm x 295 cm.
Lydabsorbenterne udføres som i infralydkabinen.

Ved siden af de omtalte målerum placeres kontrolrum, hvorfra der vil være visuel kontakt til forsøgsopstillinger og forsøgspersoner. Der indrettes endvidere et omklædningsrum med badefaciliteter til forsøgspersoner.

VENTILATION

Alle fire målerum bliver ventileret.
Der installeres et luftkonditioneringsanlæg med køling fælles for de fire rum.
Det fælles teknikrum er placeret i kælderetagen ved siden af det lyddøde rum. Kanaler føres frem som betonkanaler under gulv.
Ventilationsstøjen dæmpes dels ved effektive baffellyddæmpere i teknikrummet og dels ved støjabsorberende beklædninger indvendig i kanalerne. For at minimere generering af aerodynamisk støj vil lufthastigheden blive holdt på ca. 0,5 m/sek.

SAMMENFATNING

Med denne udbygning af Aalborg Universitetscenter får Institut for Elektroniske Systemer udbygget og samlet sine akustiske målerum under ideelle betingelser.
Dette vil indebære store undervisnings- og forskningsmæssige fordele, og vi, der er placeret på den udførelsesmæssige side i den akustiske branche, vil håbe, at det fører til store og praktisk anvendelige resultater i de kommende år.

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ANECHOIC CHAMBER AT BRÜEL & KJÆR

Carsten Fog
AS IKAS, Copenhagen Denmark

In connection with the extension of the premises of Brüel & Kjær, Ikas' division for noise and vibration control has in 1985 finished the erection of an anechoic chamber.

Types of Anechoic Chambers

In an anechoic chamber all the surfaces must be lined with a material which absorb at minimum 99% of the sound.

To fulfil this criterion, sound wedges made from mineral fibres are normally used. However, there are some alternatives.

First dented slabs can be used for cut off frequencies above 150 Hz, as shown in Fig. 1.

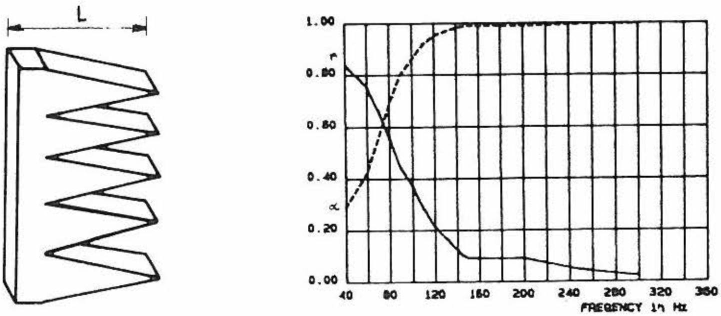


Fig. 1. Dented slabs. 83 kg/m^3, L = 800 mm

Due to the manufacturing process this type of lining will be cheaper compared to the lining of a traditional wedge. However, there is a limit in use for very low cut-off frequencies.

Another alternative type of lining are the not often used sound cubes, which have been chosen for this room, especially because there was no demand to the cut-off frequency.

Description of the Room

The room is built as a double shell construction with the inner box in prefabricated wall elements and the outer in reinforced concrete.

The inner box is 7.7 x 6.5 x 6.6 m and it is placed on resilient mountings, which will give a frequency of resonance below 10 Hz.

The cut-off frequency fo will be approx. 100 Hz.

The Lining Principle

The internal lining of the room is made of glass wool cut into cubes and blocks of different dimensions and density.

This type of lining was proposed by Dr. L. Cremer, Technical University, Berlin in the early sixties.

The basic idea is that from an acoustic point of view, it represents a large irregular surface, a socalled "acoustic jungle".

The acoustic effect is the same as that of the wedges, and the final result measured in terms of deviations from the law of distance or reflection is about the same.

However, the cube room seems to be better at high frequencies above 10 kHz.

Also the reflection coefficient is smaller at large degrees of incidence.

The cubes will submit a "diffuse" absorption (irregular reflection). The wedges will submit a more "uniform" absorption (regular reflection) due to the symmetry of the material.

Finally, the price for the cube rooms should be 3/4 of the price for a similar wedge room, but this is not clearly verified.

The Cubes

Since the cut-off frequency fo should be approx. 100 Hz, the lining should have a depth of at least 1/4 of the wave length, i.e. 70-80 cm.

As shown in Fig. 2 we were using different types of glass wool for each layer of absorption material, hanging from the ceiling and statistically even distributed over the surface.

At all inner surfaces heavy weight material (110 kg/m^3) is stacked in layers.

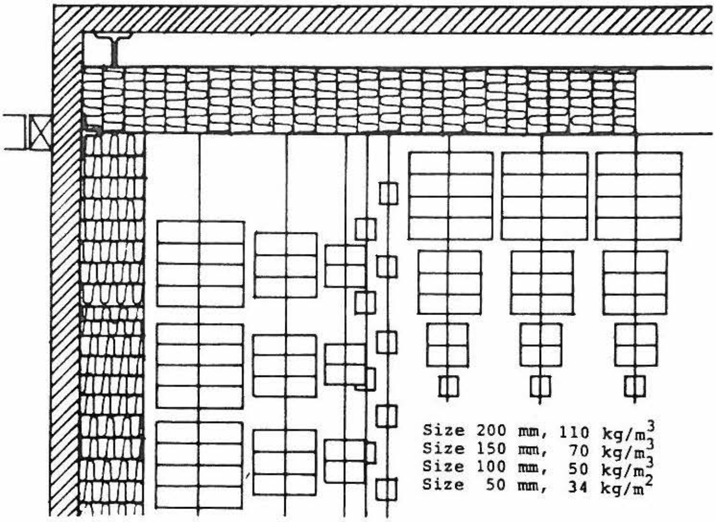


Fig. 2. Principle of lining.

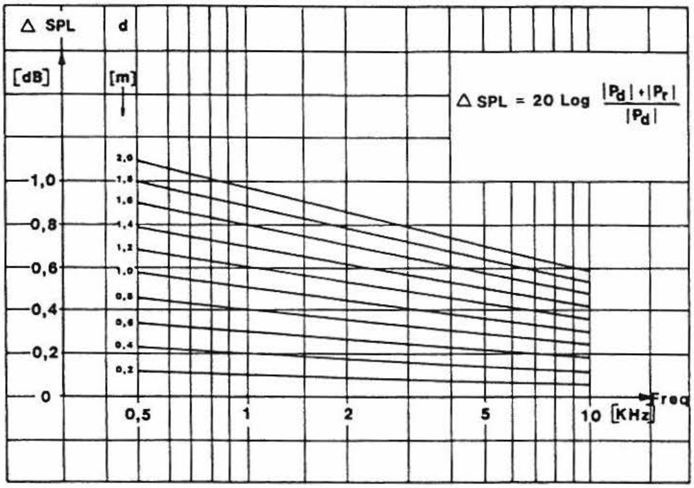
We have used

* 65,000 pcs. of different cubes and blocks
* 6,000 m of suspension wire
* 22,000 pcs. of customs seals

Measurements

For the time being a project is running for a technical university to measure the uncertainty due to reflections from the walls of the new chamber.

We can, however, present the characteristics for the old chamber. We expect these to be fulfilled for the new one since the size and the principle are the same.



d = Measuring distance

Δ SPL= Max. change in Sound Pressure Level caused by reflections from the walls.

$|P_d|$ = Pressure of direct sound field at indicated distance, d.

$|P_r|$ = Max. Pressure of reflected sound field.

Fig. 3. Measuring uncertainty due to reflections.

Future use

In the future the room will be used for measurements on
- microphones and loudspeakers
- equipment for telecommunication
- audiological test equipment

while measurement on industrial devices such as electric motors, fans etc. more and more takes place using the sound intensity techniques, which do not demand anechoic chamber facilities.

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SOUND INSULATION FOR SEALED TRIPLE GLAZINGS

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INTRODUCTION

The Danish Acoustical Institute is carrying out a project series concerning optimization of sound insulation for windows. The total project comprises investigations of hermetically-sealed double and triple glazings and frame/sash constructions. Until now, double and triple glazings have been investigated. The influence of glass thicknesses, laminating, glass spacing(s), and gas filling on the sound reduction index has been examined. The aim of the experiments is to give manufacturers of windows and glazings better possibilities of optimizing the sound insulation in relation to total weight, thickness, and price of the glazing. The investigations are described in detail in the project reports [1] and [2] (in Danish). The complex of problems and the conclusions are summarized in [3]. The main results for double-glazed windows were presented at NAS-84 [4]. The present paper presents the main results for triple-glazed windows.

DESIGN OF EXPERIMENTAL INVESTIGATION

In order to achieve maximum usefulness of the project results, "realistic" test specimens were chosen for the eperiments, i.e. the glazings were commercially produced hermetically-sealed units and were not extremely heavy, thick, nor expensive. Some samples are shown on Fig. 1.

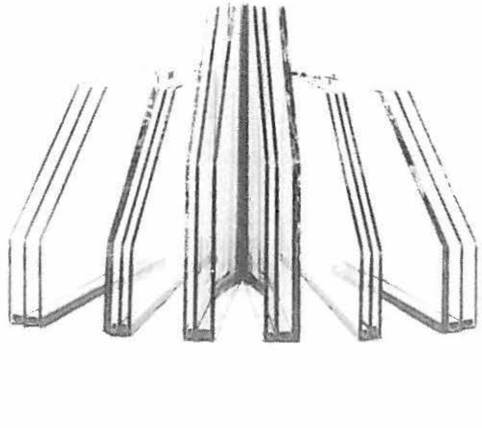

Fig. 1. Samples of different hermetically-sealed triple-glazed units

The tested types of glazings are found in Table 1. The full measurement programme for the test series and descriptions are found in [2]. At the design of the experiments special importance was attached to examining glazings with small spacings (6–9 mm) and glazings with gas filling. Further it was considered essential to examine the effect of different fillings in the two cavities of triple-glazed units.

The size of the windows was 12 M × 12 M corresponding to a test opening of 1,21 m × 1,21 m. As this size of facade windows is the most widespread size in Denmark, it is recommended as test size in the Danish Standard DS 1084 for classification of sound insulating windows. Most measurements were performed with glazings mounted in a firm frame (wood), but selected types of glazings were also tested in an openable window (wood). A sectional view of the windows and their mounting in laboratory are shown in [3]. The measurements are carried out in the transmission rooms at the Technical University of Denmark. The test facility, measurement procedure and instrumentation are described in [2]. The test method is ISO 140/3–1978.

SOME MEASUREMENT RESULTS

In [2] R_w–values (ISO 717/3–1982) are found in tabular form. Decimal values are found, too, even if the R_w–value is defined as an integer value. This implies an easier assessment of the significance of the differences. Besides there are presented a large number of comparisons of sound reduction index curves, which illustrate the influence of the examined parameters one by one and some examples of mutual influences are also found.

A table with R_W–values for glazings mounted in a firm frame is found below. Further, a few diagrams are presented, showing some selected measurement results. The glazings are described by a "code" specifying the glass thicknesses and the spacings in the indicated order. The letters *GG*, *GL*, *LG* and *LL* specify the cavity fillings in the same order, *G* = gas filling (SF_6) and *L* = atmospheric air. The mark 4/2/4 defines a laminate consisting of 4 + 4 mm glass with a 2 mm thick intermediate layer of soft polymethacrylate. The "*F*" before the glazing code in the diagrams means that the result is found for the glazing in a firm window.

Triple Glazing [mm]	Weight [kg/m^2]	Thickness [mm]	R_W / Δ_{max} [dB / dB] Cavity Fillings in the Sealed Unit: GG	GL	LG	LL
4-6-4-6-4	30	24	30 /10,2	32 / 9,4		32 / 8,6
4-9-4-9-4	–	30	31 / 8,2	34 / 8,7		31 / 6,7
4-12-4-6-4	–	–	32 / 8,6	35 / 8,4	32 / 7,1	32 / 6,6
4-12-4-12-4	–	36	32 / 7,4	35 / 7,8		32 / 6,0
6-6-4-6-4	35	26	34 / 7,7	37 / 9,6		36 / 7,0
6-9-4-9-4	–	32	*			35 / 6,4
6-12-4-6-4	–	–	37 / 7,2	38 / 8,0		36 / 5,8
8-6-4-6-4	40	28	36 /10,0	38 / 7,2	37 / 8,8	37 / 6,7
8-9-4-9-4	–	34	37 / 6,3	38 / 4,2		37 / 6,0
8-12-4-6-4	–	–	38 / 3,9	39 / 4,3	38 / 5,9	38 / 4,8
8-12-4-12-4	–	40	38 / 6,2	39 / 4,8	39 / 5,1	39 / 6,5
8-20-4-6-4	–	42	41 / 5,5	41 / 4,5		40 / 5,2
4/2/4-6-4-6-4	42	30	36 / 9,4	40 / 7,4		38 / 7,4
4/2/4-9-4-9-4	–	36	38 / 9,3			38 / 4,9
4/2/4-12-4-6-4	–	–	40 / 6,6	42 / 6,4		40 / 6,9
4/2/4-12-4-12-4	–	42	41 / 9,3	41 / 6,4		40 / 8,0
4/2/4-20-4-6-4	–	44	43 / 5,0	42 / 4,4		42 / 6,5

* The planned measurement is missing due to a wrong supply

T01199GB0

Table 1. R_w–values for sealed triple glazings mounted in a firm window (1,21 m × 1,21 m)

The influences of glass thicknesses and laminating are similar to the results for double glazings. A small asymmetry in glass thicknesses has a great effect, whereas the effect of a further change in asymmetry can be rather small. Use of laminated glass increases the sound reduction index at the high frequencies. The influence on the R_w–value is typical 2 dB, but varies between 0 and 3 dB, depending on the type of glazing. With deep resonance dips at the low frequencies there might be no increase in R_w, cf. Fig. 2 and 3 (full lined curves).

On Fig. 2 it is shown that changes in glass spacings can influence the sound reduction index considerably at some frequencies. It should be noticed, that asymmetric spacings are advantageous. The glazings are gasfilled, and the significance of spacings is less pronounced with atmospheric air in one or both cavities.

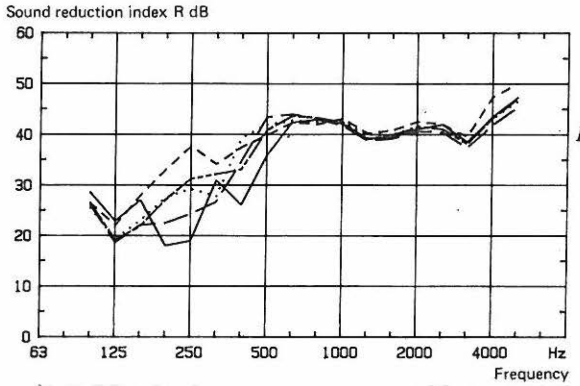

Fig. 2. Comparison of sound reduction index curves for triple-glazed units with different combinations of spacings, but all with gas-filling and identical glass combination

————	F 8-6-4-6-4 GG	Rw = 36 dB	Δmax =10.0 dB ;	Rw(dec) = 36.1 dB
— — —	F 8-9-4-9-4 GG	Rw = 37 dB	Δmax = 6.3 dB ;	Rw(dec) = 37.0 dB
-·-·-·-	F 8-12-4-6-4 GG	Rw = 38 dB	Δmax = 3.9 dB ;	Rw(dec) = 38.8 dB
···········	F 8-12-4-12-4 GG	Rw = 38 dB	Δmax = 6.2 dB ;	Rw(dec) = 38.9 dB
- - - - - -	F 8-20-4-6-4 GG	Rw = 41 dB	Δmax = 5.5 dB ;	Rw(dec) = 41.0 dB

For some types of glazings the choice of gas filling is important to the sound insulation as illustrated on Fig. 3. A better sound insulation is obtained with gas filling in one cavity than with gas filling or air in both cavities. It is assumed that the main reason for this phenomenon is mismatch of modes in the two cavities due to different sound velocities for SF_6 and atmospheric air. It should be noticed that the order of glass thicknesses, spacings and fillings are not unimportant. As an instructive example it could be mentioned that for the glazing 4–12–4–6–4 with fillings GG, GL, LG and LL the measured R_w-values are 32, 35, 32 and 32 dB, respectively.

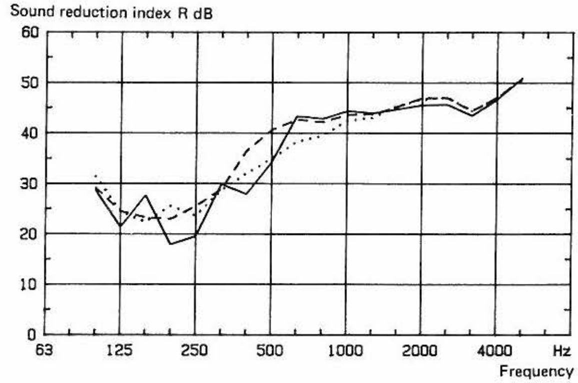

Fig. 3. Comparison of sound reduction index for a triple glazing 4/2/4-6-4-6-4 with different combinations of cavity fillings

————	F 4/2/4-6-4-6-4 GG	Rw = 36 dB	Δmax = 9.4 dB ;	Rw(dec) = 36.7 dB
- - - - - -	F 4/2/4-6-4-6-4 GL	Rw = 40 dB	Δmax = 7.4 dB ;	Rw(dec) = 40.5 dB
···········	F 4/2/4-6-4-6-4 LL	Rw = 38 dB	Δmax = 7.4 dB ;	Rw(dec) = 38.6 dB

It should be mentioned that in some literature the upper resonance frequency of a triple glazing is considered unimportant. However, if both cavities are small and gasfilled, it is found that both resonance dips are pronounced. Obviously, this is the case for the lower curves on Figs. 2 and 3.

CONCLUSIONS

The experimental results indicated below should be evaluted in the light of ordinary triple glazings being symmetric as regards both glass thicknesses, spacings and fillings. In Denmark typical glass thicknesses are 3 to 4 mm and typical spacings in triple-glazed units are 6 to 9 mm.

Based on R_w (ISO 717/3–1982) as a measure of the sound insulation property of a window, the following main results have been achieved from the experiments with triple-glazed windows (the gas used is SF_6). The sound insulation is improved by:

- Asymmetry in glass thicknesses
- Using laminated glass
- Increasing the glass spacings – this applies especially to glazings with gas-filled cavities
- Gas filling in one of the two cavities of the glazing

The last-mentioned result is particularly interesting because it is a rather simple way of improving the sound insulation. Gas filling in both cavities of a triple glazing causes only in a few cases an increase in sound insulation. With small spacings (6 to 9 mm) the sound insulation is reduced by the gas filling. The experiments have been carried out using SF_6, which is the type of gas mostly used. The results cannot be applied right away when using other types of gas with substantially different acoustic properties.

The experiments showed that an optimization of glazing details often results in a 3 to 5 dB higher sound insulation (R_w-value), alternatively a lower weight, thickness or price. The results from Fig. 3 can be considered as an example of optimization. The thickness and weight is the same for all three glazings. The price differences are small, but in fact the most expensive glazing is the poorest one with R_w = 36 dB. If "forgetting" the gas filling in one cavity, the R_w-value is improved with 4 dB!

The importance of each construction detail is connected with different frequency ranges, and different changes cannot replace one another right away. In practice, the effect of glass thicknesses is primarily connected to frequencies below 1000 Hz and above 2000 Hz, lamination to the frequency range above 1000 Hz, spacing to the frequency range below 800 Hz and gas filling to frequencies below 1000 hz.

The importance of the frame/sash construction depends on a number of factors which are not yet completely examined. However, some aspects are indicated in [2] and [5]. It has been established that the effect may be favourable in one frequency range (below approx. 1000 Hz) and unfavourable at other frequencies. When using a glazing with a high sound insulation, the sound reduction index of the total construction can be reduced by sound transmission through the frame and sash material, primarily in the frequency range of 500–2000 Hz. This was also found for some of the windows in the present investigation. Further it is well known that leaks are of special importance at frequencies above approx. 500 Hz. The problems connected to frame/sash construction are being examined more closely in a project which will be finished by the end of 1986.

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SOUND INSULATION OF WINDOWS IN THE FIELD

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1. INTRODUCTION

This paper deals with the difference between laboratory and field measurements and with the correlation between different A-weighted level differences and R_w. The differtent measurement methods and possible improvements will be discussed in the oral presentation.

2. MEASUREMENTS

5 different modern Swedish 3-pane windows have been tested in the laboratory and in the field. The 38 field measurements have been carried out either by using traffic noise or a loudspeaker with 45 degrees angle of incidence.The outdoor microphone was placed on the facade giving +6 dB reflection. The flanking transmission through the facades was estimated to be negligible. The results are summarized in Figure 1.

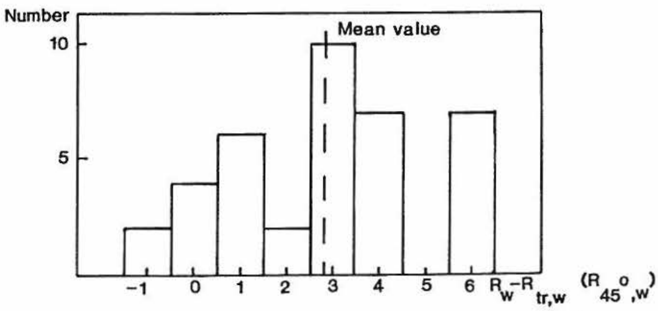

Figure 1. The measured difference between laboratory and field measurements

The results indicate lower field values, probably due to the difference in measurement methods and mounting conditions. The difference in mounting was probably the same for each window in each of the measurement objects.The standard deviation of the measurements in each of the 5 objects was between 0,5 and 1,5 dB. An estimate of the difference between laboratory and field measurements could be 3 dB.

3. RATING

3.1 Defining a traffic noise index

In a proposal for Nordtest method (1) a traffic noise reduction index, $R_{A,tr}$, is defined as

$$R_{A,tr} = -10 \lg 10^{(L_{ui}-R_i)/10} \qquad (1)$$

where i=index for the 1/3 octave band,
R_i=sound transmission index measured in the laboratory and L_{ui}=traffic noise spectrum normalized to 0 dBA.

In the Nordtest method the index can either be calculated for the frequency range 100-3150 Hz or 50- 5000 Hz. Unless a standardized urban road traffic spectrum is used in the range 100-3150 Hz (Spectrum S1 below) the index should be called special traffic noise reduction index and denoted $R_{A,tr,s}$. Using the definition of sound reduction index in ISO 140/5 it can be shown, assuming that 10 lg(S/A) is constant over the whole frequency range used, that $R_{tr,A}$ relates to the outdoor-indoor difference in A-weighted sound pressure levels DL_A as

$$DL_A = R_{A,tr} - 10 \lg(S/A) \qquad (2)$$

It is here assumed that the outdoor sound pressure level is measured 2 m in front of the facade and that the level is 3 dB higher than the free field level. With S=2 m^2 and A=10 m^2

$$DL_A = R_{tr,A} + 7 \qquad (3)$$

3.2 Different traffic noise spectra

7 different spectra have been studied including two spectra for road traffic, 3 for rail traffic and two for air traffic. All spectra are based on recent measurements carried out during the years 1982-1985.

The spectra are all A-weighted and normalized to 0 dBA. The same normalization can be used both for 100-3150 Hz and for 50-5000 Hz. The spectra, numbered S1-S7, are presented in Figure 2-4 and briefly described as follows:

S1: Reference spectrum. Mixed urban road traffic at 50 km/h and with about 10% heavy vehicles.
S2: Mixed highway road traffic at 90 km/h and with 10% heavy vehicles. Mean values of measurements on smooth and rough textured surfaces.

S3: Normal railway traffic at high speeds.
S4: Railway traffic not belonging to S3 or S5.
S5: Normal railway traffic at low speeds.
S6: Aircraft noise representing starting DC-9s.
S7: Aircraft noise representing propeller aircraft. Mean value of 10 different types of planes.

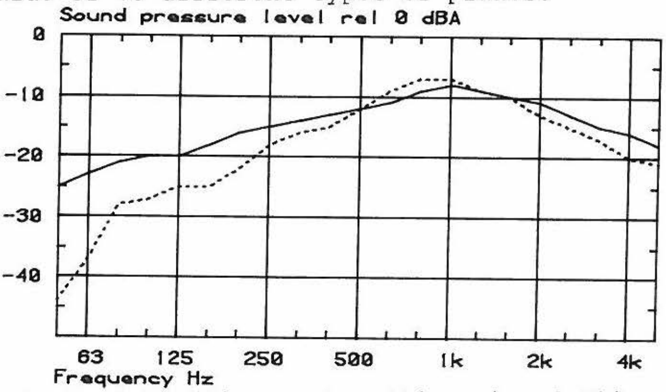

Figure 2. Road traffic spectra S1(———) and S2(········)

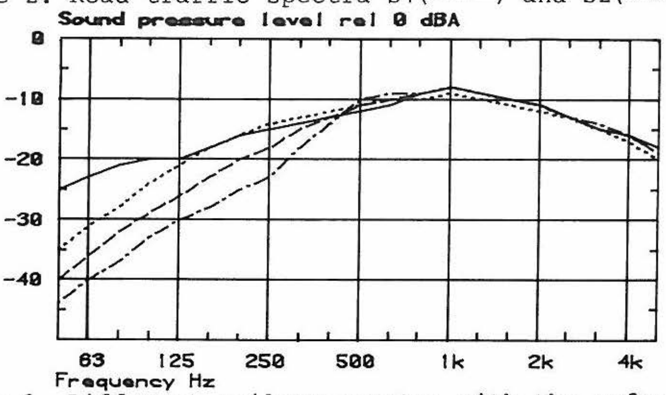

Figure 3. Different railway spectra with the reference S1. S3(·······), S4(– – – –), S5(–·–·–)

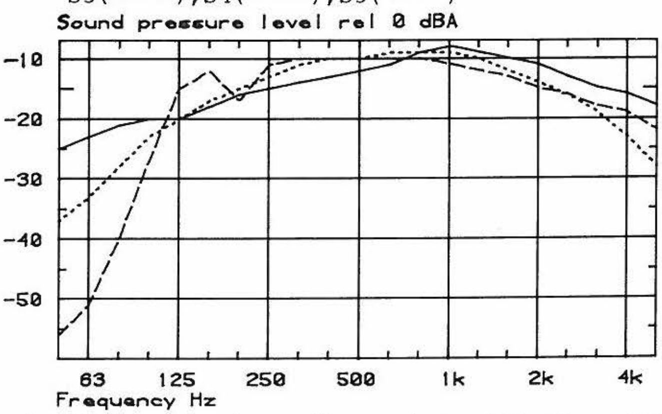

Figure 4. Different aircraft spectra with the reference S1. S6(·····), S7(– – – –)

3.3 Correlation between R_w and $R_{A,tr,s}$

The different spectra have been used to calculate the indices of 47 different 3 pane windows with R_w ranging from 27 to 45 dB. The correlation coefficient r is at worst 0,96 for S1 and S7 and at best 0,99 for S2, S4 and S6. Figure 5 indicates that R_w is as good as the new traffic noise reduction index as long as we are only interested in a relative ranking and the frequency range 100-3150 Hz. However, it is obvious that R_w is not a very good measure when we want to estimate the noise reducing effect. If we expand the frequency range to 50-5000 Hz the correlation will become worse. With the same traffic noise spectra and windows the correlation coefficent will drop to about r=0,5.

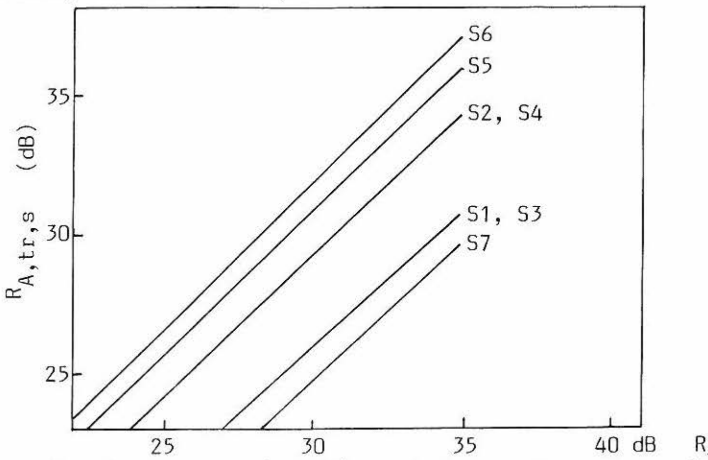

Figure 5. The regression lines between $R_{A,tr,s}$ and R_w.

4 CONCLUSIONS

Field tests according to ISO are likely to yield lower(3 dB) sound reduction than laboratory tests. R_w can normally be used for a relative ranking of windows irrespective of the type of traffic noise. However, if we are interested in the actual dB reduction R_w is not very suitable. The same window will reduce different types of traffic noise differently(1-7 dB). For these purposes a traffic noise index as outlined above should be a useful complement to R_w.

5 ACKNOWLEDGEMENTS

This work has been financed by the Swedish Building Research Council and Nordtest.

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REQUIREMENTS ON SOUND INSULATION IN BUILDINGS AND VERIFICATIONS BY ACCREDITED TESTING LABORATORIES IN GERMANY

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Introduction

The question for possibilities to reduce obstructions in international trade caused by different requirements, measurement methodes and accreditation procedures of testing laboratories is getting importance also in the field of sound insulation in buildings. In particular this counts for the membercountries of the EC, as in June 1985 an according aim in view was passed. It states that until 1992 free trade and commernce of goods, people, services and capital shall be realized as well as the technical obstructions between the members shall be cancelled. Parts of this strategy are the mutual approval of national standards, accreditations, testing laboratories, testing results and testing certificates /1/. Therefore inventory seems to be necessary in order to analyze to which extent handy solutions are already possible under given circumstances. By this it is obvious that the field of requirements on sound insulation in buildings is not at all immovably determined in Germany, as in the latest past substantial changes were made and are to be expected in the near future. These will not hinder the efforts in international harmonization, to some extent they will immediately stimulate them.

Requirements on sound insulation

There are different kinds of requirements on sound insulation to be found in practice: a) minimum requirements fixed by law and recommended proving of the realization, b) independent arrangements by contracts, c) demanded sound insulation based on the realized type of building construction respectively on the state of the art. In the following only point a) will be discussed.

Considering requirements on sound insulation in buildings determined by law one must pay regard to the fact that the Federal Republic of Germany is the only state in the EC governed federatively. Therefore the legal basis of sound insulation requirements is laid down in 11 different building regulations of the federal states with additional regulations of execution. However, the codes on this subject have an extensive conformity and efforts in reducing the number of regulations and in standardization were successful in the last years. Thus since 1981 a new model building regulation exists by which the different states are designing revisions of their own building regulations or have finished that already (recently e.g. Berlin in 1985 and Niedersachsen in 1986). These building regulations are the basis of legal requirements on sound insulation, but no numerical values or instructions to proceed are to be found here. They are specified in DIN-standards or others as the VDI-instructions. Each state determines by orders concerning the building regulations which standards or even which part of them to be valid in public as a "technical buildung rule". In this field there are still differences to some extent between the states. Without exception in all the states established, however, is the comprehensive relevant standard, the DIN 4109.

A co-operation in this field of all federal states happens in the "Fachkommission Baunormung". Furthermore the "Institut für Bautechnik" in Berlin has got the competence to coordinate public tasks regarding building matters and to issue besides others approvals for new building materials, building elements and building methods to be valid in all the states.

Requirements on sound insulation stated in DIN 4109

At present the legal minimum requirements on sound insulation in buildings are fixed in DIN 4109 "Schallschutz im Hochbau" (issue 1962). Essentially they are referred to the properties regarding sound insulation of walls and floors installed between dwellings. As the single-number quantity for the airborne sound insulation the term "Luftschallschutzmaß LSM" is used in the standard. But today the weighted sound reduction index R_w according to ISO 717 is in common use. The following relation exists: R_w = LSM + 52 dB (LSM equals M_a in ISO 717). The minimum requirements on impact sound insulation are given as the single-number quantity "Trittschallschutzmaß TSM". With sufficient accuracy the relation to the weighted normalized impact sound pressure level $L_{n,w}$ (ISO 717) is valid with $L_{n,w}$ = 63 dB - TSM. These simple relations became possible by introducing a new version of DIN 52210, Teil 1 (August 1984), where the measuring methods for tests in building acoustics have been harmonized to ISO 140. Also the other parts of DIN 52210 have been revised and most of them became valid already.

It is important to be noticed that in Germany requirements are not determined regarding the sound insulation between dwellings but the building elements (walls and floors) between them have to meet minimum requirements on sound insulation. The discussion on the question of necessity of fixing airborne sound insulation requirements between rooms in buildings by quantities like weighted standardized level difference $D_{nT,w}$ (ISO 717) is also in Germany not yet finished (see for instance /2/).

In the standard DIN 4109 of 1962 different parts today are true for modification especially regarding the impact sound insulation requirements. The standard contains also recommended values for improved sound protection which can serve as basis for contracts. 1984 a new draft of a revision has been published substituting a draft of 1979. But at present the minimum requirements laid down in DIN 4109 of 1964 are still the valid legal regulations.

Only when the fulfilling of the minimum requirements are called in question an acceptance test has to be certificated. From this different handling by the different authorities of the different states may occur. Furthermore mostly complaints are leading to acceptance tests. Regarding buildings with a great number of dwellings sometimes the financial backer demands random samples of acceptance tests.

Qualification and acceptance tests by testing laboratories

Measuring procedures for qualification and acceptance tests are fixed in DIN 52210. There also the laboratory test facilities necessary to execute the qualification tests are standardized. Regarding international comparability of test results in this field substantial difficulties occur as a part of the tests have to be done in specified test rooms which are standardized in this form only in Germany (test rooms for walls and floors with flanking transmission similar to a building situation, test rooms for windows and panes and for doors). Testing institutes intending to carry out qualification and acceptance tests according to DIN 4109 in official proceedings of building authorities may pass through an "accreditation procedure". They are then incorporated in a list which is published by the "Institut für Bautechnik, Berlin" /3/ and are acknowledged in all the federal states as competitive testing institutes regarding DIN 4109. At present this list comprehends 6 institutes for qualification and acceptance tests as well as 54 institutes for acceptance tests only.

Accreditation procedure for testing laboratories

The "Institut für Bautechnik" consulted by experts besides others of Physikalisch-Technische Bundesanstalt (PTB) and Bundesanstalt für Materialprüfung (BAM) is organizing this "accreditation". The "official approval" is declared by the "highest building authority" (the competent minister or senator) of the state where the institute is located. Approval by the delegates of the other federal states in the "Fachkommission Baunormung" is necessary. For this procedure principles have been fixed: The applying institute has to be independent. Demands regarding the qualification of the staff and the relevant education of the responsible person are made. By test reports originated of the institutes work the qualification has to be ascertained. The institute also has to take part in comparative measurements according to PTB rules. Testing institutes applying for qualification tests have to prove that all needed special test rooms and equipment are existing and are proved to fulfill the relevant requirements.

Regarding comparative measurement on sound insulation in buildings rules of PTB have been published /4/. There the minimum of needed instrumentation, the comparison procedure and tolerances which have to be met by the measurement results are determined. Measurements of airborne sound insulation and of impact sound insulation of a test object have to be performed. In addition to the comparative measurements which have to be executed by the institute staffs with equipment of its own tests of instruments take place (filters, standard tapping machines and loudspeaker for airborne sound insulation measurements). The sound level meter has to meet the requirements of IEC 651 or 804, class 0 or 1, approved by a certificate of an "Eichamt" or PTB. Comparative measurements have to be repeated in a three years turn. In the past they took place in PTB, Braunschweig, in the future they will be done by "Staatliches Materialprüfungsamt Nordrhein-Westfalen" in Dortmund.

International relationship

It can be pointed out, that in Germany the major part of relevant measuring methods are now harmonized with international standards. But in the area of qualification tests which especially seems to be important relating to international trade substantial difficulties may occur regarding the acceptance of test results of foreign institutes as the tests in Germany have to be performed partly in very special test rooms. An attempt has been made to find a solution to some extent. A German and a French state institute agreed to accept measurement results mutually from such kind of tests where comparative measurements had been successful and to issue in such cases national test certificates /5/. In the field of acceptance tests a few foreign testing institutes have founded "sisters" in Germany which passed through the German procedure. One Danish institute participates in the PTB comparative measurements and equipment tests. It is now applying for being in the above mentioned list. Here especially the problem occurs that according to the present valid procedure test reports of the institute which are in conformity with German sound insulation standards should be available. But also in this case a mutual acceptable answer will be found.

The realization of the aims of the EC at this time surely is not possible completely but solutions especially in parts of practical importance are existing at least in single cases.

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EXTERNAL SOUND INSULATION OF BUILDINGS: DEVELOPMENT OF A FINNISH CODE PROPOSAL

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Introduction

Despite the apparent and growing importance of outdoor noise intruding into houses, few countries have settled specific requirements for the sound insulation of external wall components. In some others, some kind of lower level practice serves the purpose. In Finland, the Ministry of the Environment, Physical Planning and Building Department, has initiated the work towards Finnish regulations. This paper describes the proposal /1/ submitted to the ministry for review in late 1985. An open formalism was preferred in the proposal, in order not to restrict its usefulness together with the Finnish Noise Abatement Act which is now under preparation.

What is used today?

Even in the present situation with no written regulations at all, land use planners constantly set requirements on properties near roads etc. The values are in "decibels" with no other definition. The problem is thus passed on to the building inspection.

The work started with a brief review of foreign examples of regulations and "decibel" descriptors. In Denmark, Bygningsreglement BR-82 relates in a straightforward manner the weighted apparent sound reduction index R'_w of both windows and walls to the 24-h equivalent level L_{Aeq} outdoors /2/.

The same principle is used in the German DIN standard /3/ with, however, a more explicit structure of a dimensioning (massgeblich) outdoor level related directly to the R'_w. The dimensioning level of various noises is left to be defined in another context.

Another approach is to formulate the regulations to be based on the difference of A-weighted levels outdoors and indoors, which is used in Norway /4/ and in the French NF standard /5/. The concept is also supported by the recent NORDTEST project on developing a Nordic method for rating windows /6/. With this approach the form of the outdoor spectrum must be agreed upon.

Problems to be solved

In addition to defining acoustical quantities, many non-acoustical parameters and problems are involved in formulating the regulations.

Starting point: outdoor noise level data The problem here is twofold: how to define the external sound level, and where the level data is available to the end user.

It is probable that the joint Nordic calculation models for outdoor noise will be in central position in the future Finnish noise assessment practice. It was considered advisable that the insulation regulations could use these despite the fact that various sound level quantities are used for different noise types. A practice in which the numerical values are given by local authorities was considered preferable when shaping the insulation regulations.

The basis for regulations: indoor noise level In effect, the dimensioning of the sound insulation against outdoor noise must be founded on the desired noise level inside. When referring to indoor noise level, one usually means either the average level in the room (easy to measure) or the diffuse field level (easy to calculate). However, some indoor noise limits (e.g. the existing Finnish health and building regulations) refer to a third possibility, the worst-location level. The nearest proximity (to within 0.5 m) of room boundaries is excluded.

The use of the latter has the advantage that the sound level difference and the power reduction are often numerically equal. Then the window area and room absorption have no influence on the insulation requirements. The inclusion of the room absorption to the code system could lead in situations where variations in room furniture affect the acceptability of the external wall.

Noise spectra No international or even Nordic agreement seems to have been reached of representative spectra for different noise types. Numerous alternatives may be found in e.g. /6,7/, some of which are shown in Fig. 1.

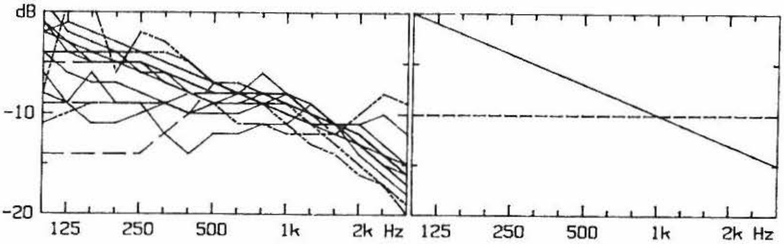

Fig. 1. Unweighted reference spectra. **(a)** examples from /1,4,5,6/: (——) road traffic, (---) rail traffic, (-·-) aircraft noise; **(b)** proposed: (——) road traffic, (---) all other noise types.

The code proposal

Dimensioning outdoor level L_{Au} is A-weighted, time-integrated free-field level outside the building in general. A value stated by the commune is used primarily, a result from a calculation secondarily, and measured values only exceptionally. The noise type is either "road traffic noise" or "other noise". The widest-spread noise type, road traffic noise, is used primarily. The calculations shall be made using latest authority guidelines.

Dimensioning indoor level L_{As} is a computational quantity for noise 0.5 m from external wall. Only noise coming from outside is considered. The choice of the 0.5-m level as the dimensioning indoor level leads to a very simple procedure where each building component must individually fulfil the requirement, unless they are smaller than 1 m^2. Larger components are dimensioned directly by their sound reduction properties and smaller ones are weighted by their areas.

Outdoor noise reduction index R_A is defined in accordance with /6/ as the difference between the above mentioned outside and inside A-weighted levels obtained from the laboratory-measured sound reduction values, using a reference spectrum which depends on the noise type. The code prescribes two versions of R_A: R_{At} for road traffic case and R_{Av} for the cases of all other noise types. The former is selected if road traffic noise is not more than 5 dB below the total noise level.

The unweighted reference spectra are very simple, as shown in Fig. 1(b). These choices were motivated because the importance of the shape details was found very weak.

The requirements External insulation R_A shall be at least equal to the difference between the dimensioning outdoor and indoor levels. L_{Au} is limited upwards to 75 dB and downwards to 55 dB. Table 1 shows the L_{As} values and Table 2 the requirements for R_A.

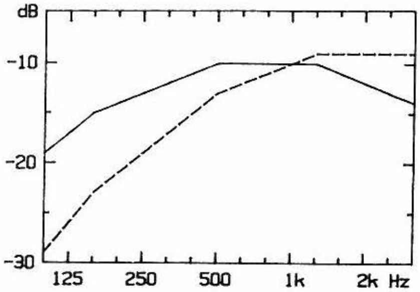

Fig. 2. Proposed A-weighted reference spectra: (—) road traffic, (---) other noise types.

Table 1. Proposed dimensioning indoor level.

L_{As} = 30 for residential houses, hotels etc and hospitals
35 for schools, kinder gartens and offices
40 for offices in industrial premises

Table 2. Proposed requirements for the outdoor noise reduction index R_A.

	L_{As} = 30	35	40
L_{Au} = 55	25	25	25
60	30	25	25
65	35	30	25
70	40	35	30
75	45	40	35

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LABORATORY MEASUREMENTS OF SOUND REDUCTION INDEX – CONFIDENCE OF TEST RESULTS

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INTRODUCTION

The quality or performance of a product is often assessed or documented by means of tests carried out according to test methods, which are standardized.

Test results have different applications:
- Documentation for performance – in general or for a specific purpose
- Comparison with test results for other products
- Classification

Unfortunately, it is a fact, that tests performed on presumably "identical products" do not, in general, yield identical results. Regarding the above mentioned applications it is obvious that when a product is tested twice by the same laboratory (or organization) or by different laboratories, the difference between the results should be within certain limits. So, some specifications are necessary concerning the test facility, and the precision of the test procedure. To a certain degree differences between test results must be accepted, because it is impossible to specify completely all factors that influence a test result.

The concepts repeatability (r) and reproducibility (R) are used as confidence measures for test results, within a laboratory and between different laboratories, respectively. Both give a value, below which the absolute difference between two single test results may be expected to lie with a specified probability. Normally a confidence level corresponding to 95% probability is used. Determination of repeatability and reproducibility is described in ISO 5725 [1], which is not related to a specific test property or a specific type of products, but is intended to be applied to standardized test methods in general.

The present paper deals with some Nordic Round Robin experiments comprising measurements of sound reduction index for the same type of building component in five laboratories. The aim of the experiments was to provide repeatability and reproducibility values for sound reduction index measurements carried out according to the test method outlined in ISO 140 [2]. The present version of the standard includes requirements to the repeatability values, but reproducibility requirements have been postponed due to the lack of experimental data.

ROUND ROBIN EXPERIMENTS

The experiments were carried out within two Nordtest-projects NT 235–80 and NT 360–82. The test series are described in detail in the project reports [3] and [4].

The main aim of the project NT 235–80 was to achieve repeatability and reproducibility data for measurements of sound reduction index. The measurement programme was set up, so it was possible to estimate some sources of errors and their relative contributions to the total variance of a measurement result. For economical reasons the number of test objects had to be limited to one even if more than one type of test specimen and one level of the test property were desirable, since the repeatability and reproducibility can be expected to depend on both factors. A glazing was chosen as test specimen because the largest discrepancies in measurement results were found for windows. The test specimen for the project NT 235–80 was a glazing mounted in a flat test opening, and 4 test series were planned within this project.

Before the accomplishment of the experiments, a proposal for a supplementary test series was brought about. Due to an international discussion on the shape of test openings for windows, it was decided to carry out – within a new project NT 360–82 – measurements for the same type of glazing mounted in a simulated staggered test opening. The aim was to find out, whether the reproducibility is considerably influenced by the type of test opening.

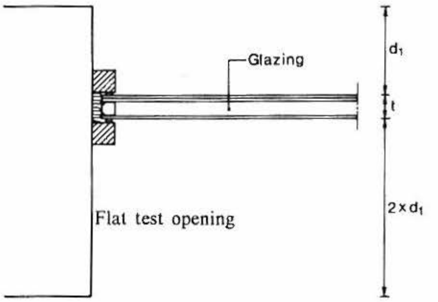

Fig. 1. Mounting of glazing in a flat test opening 1,21 m × 1,21 m

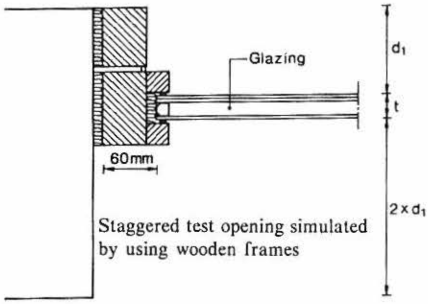

Fig. 2. Mounting of glazing in a staggered test opening, 1,09 m × 1,09 m

The total measurement programme for the projects included 5 series of measurements. Series 1–4 belong to NT 235–80 (glazing in a flat test opening), and series 5 belongs to NT 360–82 (glazing in a staggered test opening):

Measurement series	Number of measurements in each laboratory
Series 1: Influence of stochastic noise signal	5
Series 2: Remounting of glazing	1
Series 3: Repeatability/reproducibility	6
Series 4: Reproducibility within laboratories (long-time variations within a laboratory simulated by changing test procedure and/or conditions)	5
Series 5: Repeatability/reproducibility for measurements of a glazing in a staggered test opening	6

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STATISTICAL ANALYSIS

The statistical analysis is carried out according to the guidelines in ISO 5725 [1]. A single test result obtained with a standardized test method is described as a sum of three components: $y = m + B + e$, where y is a single test result for the tested material, m is the average of several test results for the material tested in many laboratories, B is a term representing the deviation for the actual laboratory from m, and e represents a random error occurring in every test.

When the repeatability r or the reproducibility R is used to check whether a difference between two measurement results is significant, a bias of m will have no influence and can be ignored, and the calculations of r and R are based on the variances of B and e. For further analysis of sources of errors (test series 1–4) the terms B and e are subdivided into some individual components:

$B = B_o + B_s + P$ and $e = u + d + k$, see Fig. 3. A more detailed description of the statistical analysis is found in [3].

TEST RESULTS

The significance of the various error components is illustrated in Fig. 3. It is seen that the important factors are the laboratory components B_o and B_s.

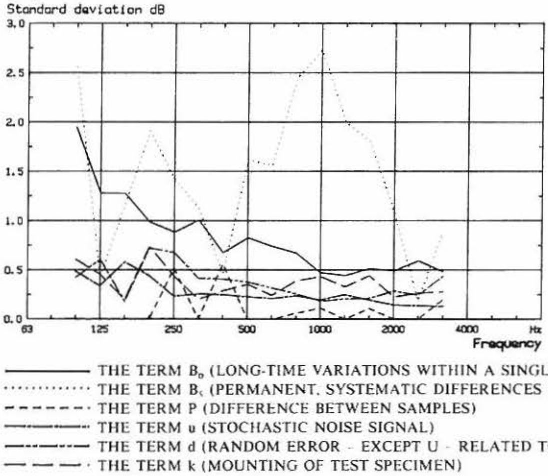

Fig. 3. Standard deviations for the various error components calculated from test series 1 – 4 (NT 235–80)

In short, the repeatability depends on the terms u, d and k and the reproducibility depends on all the terms shown in Fig. 3. Consequently, the repeatability values are, as expected, small compared to the reproducibility values. The repeatability values fulfill the requirements in both the present and proposed ISO 140/2. The term B_S has a very pronounced peak around 1000 Hz, which implies poor reproducibility values, and the proposed reference values for ISO 140/2 are exceeded considerably in this frequency range. This is illustrated in Fig. 4, which also shows the reproducibility for a glazing in a simulated staggered test opening.

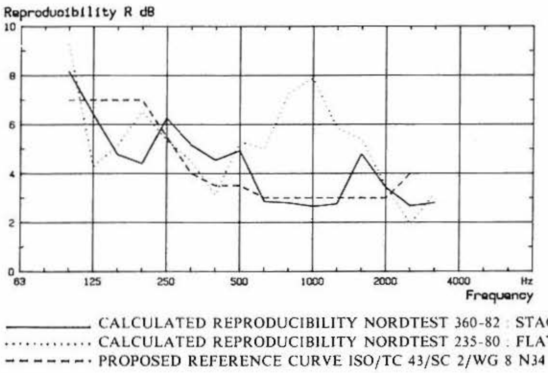

Fig. 4. Reproducibility values for measurements of sound reduction index for a glazing mounted in two different types of test openings (see Figs. 1 – 2). The stretched curve indicates proposed values for a revised ISO 140/2

In the frequency range 630–1250 Hz the reproducibility values are considerably better with the test specimen mounted in a staggered test opening. Possibly, the main reason is not the shape of the test openings, but the improved uniformity due to identical materials in the test openings. However, the test values of reproducibility do still not strictly fulfill the proposed reference values at all frequencies, even if the character of the two curves are rather similar.

Besides different reproducibility values for the two types of test openings, it is found that the measured sound reduction index is systematically influenced by the type of test opening. This might be caused by different lateral modes in the airspace within the glazing. The measurement results (mean of five laboratories) are shown in Fig. 5. The best sound insulation is obtained with the staggered test opening. It should be noted that both the systematic difference and the reproducibility are frequency-dependent and are further expected to depend on the type of test specimen. It is obvious that a systematic difference added to other reproducibility variations complicates a comparison between measurement results from different laboratories.

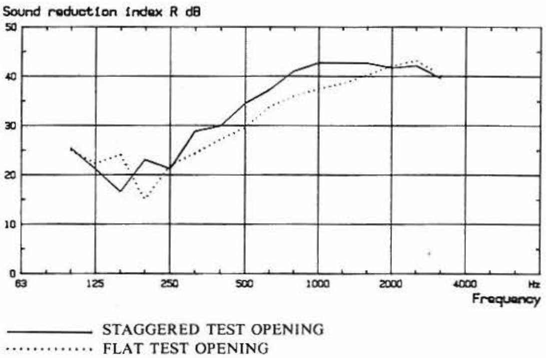

Fig. 5. Measured sound reduction index for a glazing mounted in two different types of openings. Mean values of five laboratories

CONCLUSIONS

The variances for the various components contributing to the total variance of sound reduction index measurements have been estimated. The results show that variances due to other factors than differences between laboratories are relatively small.

As indicated in ISO 5725 [1], pronounced differences between test results reported by different laboratories may indicate that the standard is not yet sufficiently detailed and can possibly be improved. As regards the specific problems of glazings this is also realized internationally, and an ISO working group is revising ISO 140/3, (test method). Another working group is revising ISO 140/2 (precision). However, the problem of glazings is somewhat complicated, as the choice of a standardized method should be based on considerations taking into account both the systematic influence due to the type of test opening, the practical aspects concerning mounting and the applications of the test results.

The inter-laboratory test described deals specifically with precision problems related to laboratory measurements of sound reduction index according to the test method ISO 140 [2]. However, problems are likely to exist for several test methods. The results illustrate that the precision of test results to a high degree are related to the experimental details specified in a test method.

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LJUDISOLERINGEN HOS GAMLA TRÄBJÄLKLAG

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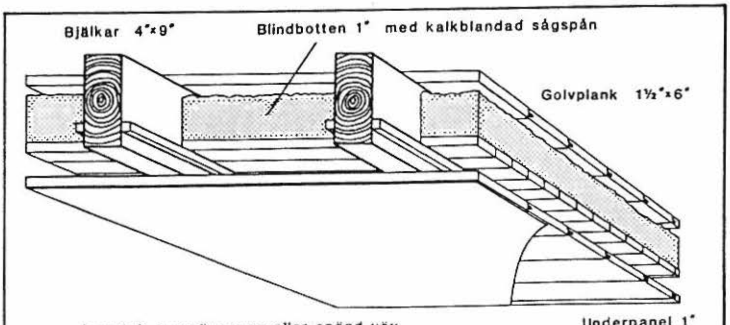
Bjälklagens historiska utveckling

Alla riktigt gamla hus är av naturliga skäl byggda med träbjälklag. Bjälklag av betong började först användas i de stenhus som byggdes under 30-talet. Efter 2:a världskriget konstruerades i allmänhet samtliga bjälklag av betong. Betongbjälklagen gav ett bättre brandskydd, förbättrade hållfasthetsegenskaper och gav även bättre ljudisoleringsegenskaper än de gamla träbjälklagen. Eftersom detta föredrag enbart berör flerfamiljshus med träbjälklag kommer det alltså att handla om hus som är mer än 40 år gamla. Anledningen till att vi intresserar oss för dessa bjälklag är att detta husbestånd idag är föremål för omfattande renoveringsåtgärder i Sverige.



Figur 1.

Ett tidstypiskt trähusbjälklag från perioden 1900-1930 visas i figur 1. Tidigare användes skrädda bjälkar i dimensioner från 6" x 8" till 7" x 10" men sågindustrins utveckling i början på 1900-talet medförde att man successivt övergick till klenare dimensioner.

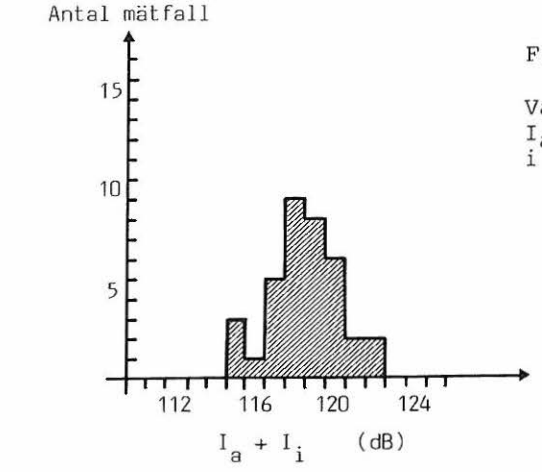
Bjälklagsdimensionerna i stenhusen utvecklades på samma sätt som i trähusen. De fyllnadsmaterial som förekom i stenhusens bjälklag var dock tyngre än i trähusen. Vanligt förekommande material var kalkgrus, sand, koksaska och tegelskrot. Under bjälklaget spikades en spräckpanel eller s k glespanel som försågs med en vassrörsmatta plus puts. Putsade ytor var regel även för stenhusens innerväggar medan det förekom i trähus där krav ställdes på gott brandskydd.

Gamla mätresultat

De mätresultat som återfunnits i den tillgängliga litteraturen har sammanställts i referens [1]. Materialet är magert och mätningarna har i de flesta fallen genomförts med gamla instrument och med metoder som inte överensstämmer med dagens standarder. Någon övergripande analys av ljudisoleringen hos dessa bjälklag har inte kunnat återfinnas i litteraturen.

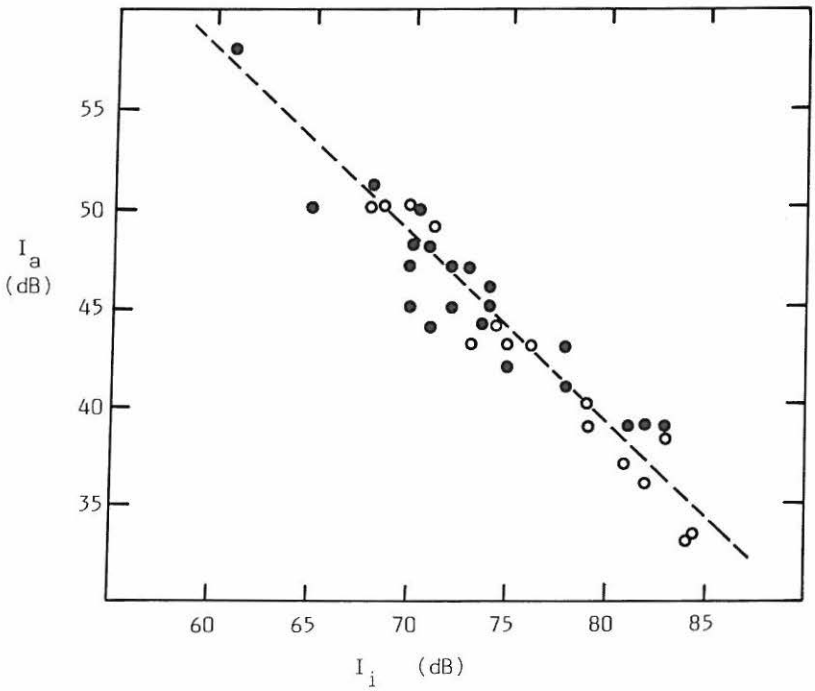
Sex ombyggnadsobjekt

Provningsanstalten har följt sex ombyggnadsobjekt. Ljudisoleringsmätningar genomfördes på konventionellt sätt dels före och dels efter ombyggnaden. Ljudisoleringsresultaten före ombyggnad återges i figur 2 och 3. Figurerna visar att det är stor spridning på värdena men att det föreligger en utmärkt korrelation mellan luftljudsisolerings- och stegljudsnivåindexen. Jämför man med de svenska byggnormskraven så finner man vidare att det är stegljudsvärdena som är dimensionerande.



Figur 2.

Variationsområdet för $I_a + I_i$ för mätfallen i figur 3.

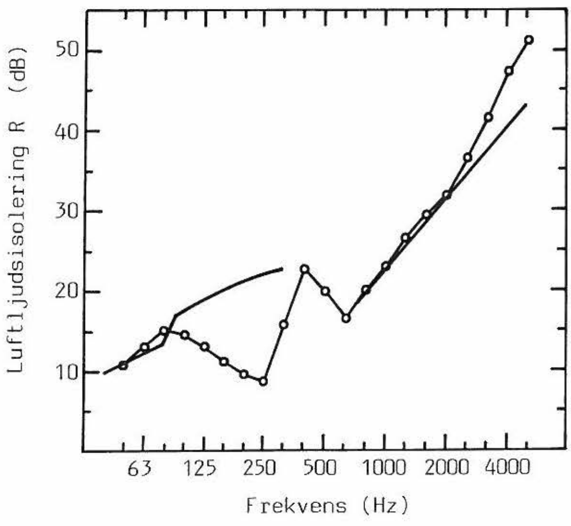


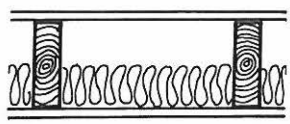
Figur 3. Sambandet mellan I_a och I_i för ett antal träbjälklagsfall. Resultaten härstammar från de sex fältobjekten (●) samt från ett antal litteraturreferenser (○).

Laboratorieexperiment

Systematiska laboratorieexperiment har genomförts för att lära känna ljudisoleringsmekanismerna och vilka konstruktionsparametrar det är som är viktiga för dessa bjälklag. Dessa experiment och resultat återges i detalj i referens [1].

I figur 4 jämföres de tillgängliga dubbelväggsteorierna med luftljudsisoleringsresultaten för det bjälklag som närmast överensstämmer med förutsättningarna för teorierna. Denna jämförelse visar att det finns två problemområden vid 250 Hz och 630 Hz. Den kraftiga ljudisoleringsförsämringen vid 250 Hz saknar motsvarighet i de tillgängliga teorierna. Även 630 Hz-problemet är svårt att förklara helt, även om detta problem sammanhänger med att takskivan har sin koincidensfrekvens vid 600 Hz.





Figur 4. En jämförelse mellan uppmätta värden (-o-o-o-) för bjälklagskonstruktionen ovan och motsvarande ljudisoleringsvärden beräknade med hjälp av de tillgängliga teoretiska sambanden (———).

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SUBJEKTIV OPPLEVELSE AV TRINNLYD I FLERFAMILIEHUS MED LETTE ETASJESKILLERE AV TRE MELLOM LEILIGHETER.

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Det har lenge vært kjent at den internasjonalt standardiserte bankemaskinen (ISO 140/VI) gir trinnlydnivåer på lette trebjelkelag som overensstemmer dårlig med opplevd sjenanse når folk går på slike etasjeskillere.

Oslo helseråd har de siste årene mottatt en rekke klager over plagsom trinnlyd fra folk som bor i flerfamiliehus der trebjelkelag er leilighetsskillende konstruksjon. Kontrollmålinger med bankemaskin har vist, at til tross for at folk er plaget av trinnlyd er byggeforskriftenes krav til trinnlydnivå stort sett tilfredsstilt. (Krav: $I_i = 63$ dB).

For å få kartlagt trinnlydproblemene i hus med lette trebjelkelag nærmere, gjennomførte Oslo helseråd i 1985 en undersøkelse i 9 boligområder, i samarbeid med en hovedfagstudent ved Samfunnsvitenskapelig fakultet, Universitetet i Oslo. Undersøkelsen ble lagt opp som en boligmiljøundersøkelse, der spørsmål om trinnlyd utgjorde en del. I introduksjonen til spørreskjemaet ble det sagt at: "Hensikten med undersøkelsen er å få inn informasjon om enkelte sider ved bosituasjonen slik den oppleves av beboerne." Spørreskjemaer ble lagt i postkassene til samtlige husstander i boligområdene, og samlet inn ved besøk av prosjektmedarbeidere.

BESKRIVELSE AV HUSTYPER.

Syv av de ni undersøkte boligområdene hadde flerfamiliehus med trebjelkelag som leilighetsskillende konstruksjon, mens hus i to kontrollområder hadde etasjeskillere i betong. En antok at områdene hadde relativt ensartede demografiske og sosiale forhold.

Det kom inn svar på spørreskjemaet fra 376 leiligheter i de ni boligområdene. Oversikten i tabell I på neste side viser hvilke hustyper som forekom i de forskjellige boligområdene, og antall innkomne svar på spørreskjemaet.

TABELL I: BESKRIVELSE AV HUSTYPER.

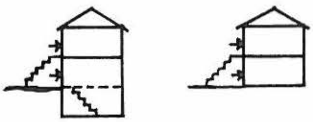
HUS TYPE	BOLIG OMRÅDER	ANTALL SVAR PÅ SPØRRESKJ.	ETASJESKILLER MELLOM LEIL.	ETASJESKILLER INTERNT I LEIL.
I	6 ABS,R/H LT, AÅ AT, RBV	252	Trebjelkelag Nedforet himling opphengt i sekundærbjelkelag	Trebjelkelag
II	1 FB	38	Trebjelkelag Nedforet himling opphengt i elastiske metall-bøyler	Trebjelkelag
III	2 NF, AL	86	Betong	Betong

Skissene nedenfor (figur 1) illustrerer (i snitt) etasjeinndelingen av de forskjellige hustypene. Etasjeskillere mellom leiligheter er vist med heltrukne linjer, mens etasjeskillere internt i leilighetene er vist med stiplede linjer.

FIGUR 1.

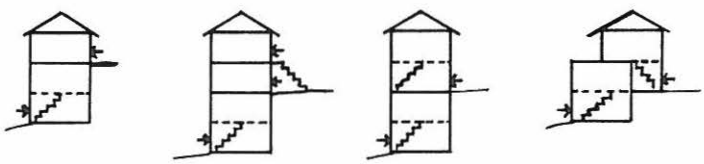
HUSTYPE

I og II



HUSTYPE

III



FYSISKE MÅLINGER.

Trinnlydmålinger ble dels foretatt med bankemaskin plassert på aktuelle etasjeskillere, dels ved at ekvivalentnivå (Leq) og maksimalnivå (Lmax) ble registrert i boligrom under etasjeskillerne mens en person på 75 kg gikk på gulvet i etasjen over.

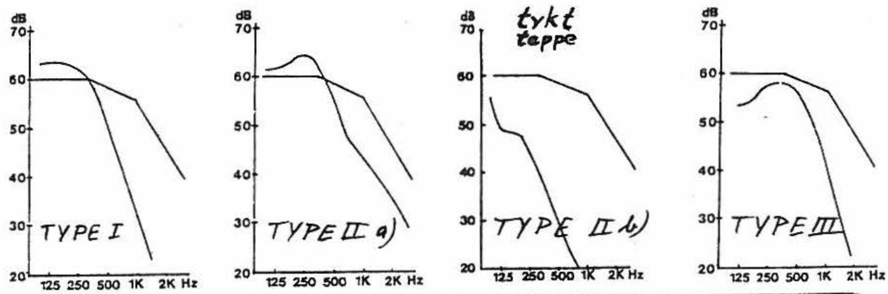
Trinnlydmålingene med bankemaskin ble foretatt med en Bruel&Kjær bankemaskin type 3204, og nivåene registrert med et Norwegian Electronics type 823 "Sound Measuring System". Trinnlyd fra en person som gikk på gulvet i etasjen over ble tatt opp i 4 mikrofonposisjoner på en NAGRA IV-SJ tospors båndopptager. Leq og Lmax i dBA og i 1/3-oktavbånd ble registrert i perioder på 3 sekunder ved hjelp av en Norwegian Electronics "Real Time Analyser" type 830.

TABELL II TYPISKE VERDIER FOR TRINNLYDINDEKS Ii

HUSTYPE	Ii-verdier (dB) Typisk	Var.omr.	Antall målinger	Gulvbelegg
I	63	60 - 65	12	Vinyl, Teppe
II a)	63	62 - 64	3	Vinyl, Kork, Parkett
II b)	52	50 - 54	2	Tykt teppe
III	57	54 - 60	6	Vinyl, Teppe

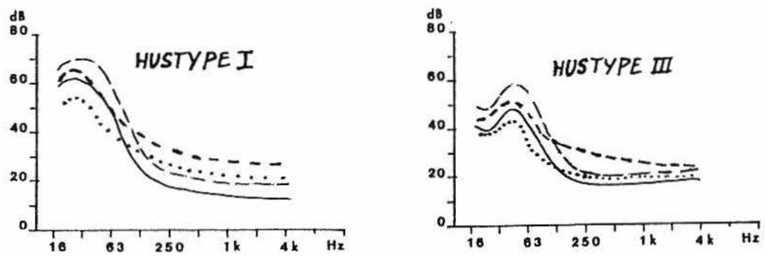
For hustype II er målingene delt i a) og b) på grunn av store variasjoner mellom resultatene. (Selvbyggerleiligheter).

FIGUR 2. TRINNLYDNIVÅER MÅLT MED BANKEMASKIN.
TYPISKE KURVEFORLØP AV TRINNLYDNIVÅ I 1/3-OKTAVBÅND 100 - 3150 HZ



TABELL III TYPISKE NIVÅER AV REELL TRINNLYD

FOTTØY:	Sko Leq	Sko Lmax	Sokker Leq	Sokker Lmax (dBA Fast)
HUSTYPE: I	31	38	33	43
II	30	35	33	38
III	23	34	28	34



FIGUR 3. REELL TRINNLYD. FOTTRINN AV EN PERSON PÅ 75 kg.
TYPISKE 1/3-OKTAVNIVÅER FRA 20 Hz TIL 4 kHz.

- - - - - - - Lmax med sko —·—·— Lmax med sokker
· · · · · · · · · · Leq med sko ———— Leq med sokker

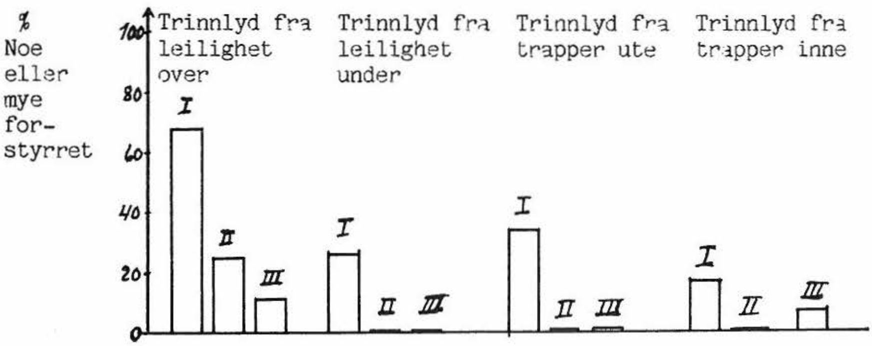
SUBJEKTIV OPPLEVELSE AV TRINNLYD.

Totalt 40 % av de 376 som besvarte spørreskjemaet var plaget av støy.

Trinnlyd ovenfra var den støykilden som flest (24 %) var forstyrret av (noe eller mye forstyrret). Deretter fulgte trinnlyd fra trapper ute (23 %), støy fra veitrafikk (20 %), lekeplass/lek (15 %), trinnlyd fra trapper inne (13 %), naboens TV (12 %) og fly (12 %). Andre støykilder ble oppgitt i mindre enn 10 % av spørreskjemaene, herav trinnlyd nedenfra (7 %).

Figur 4 nedenfor viser hvor mange prosent av dem som hadde bebodd leilighet henholdsvis over eller under seg som var plaget av trinnlyd ovenfra eller nedenfra. Figuren viser også hvor mange som svarte at de var plaget av trinnlyd fra trapper ute og fra trapper inne.

FIGUR 4. PROSENTVIS ANTALL SOM FORSTYRRES AV TRINNLYD.



DISKUSJON OG KONKLUSJONER.

Resultatene av boligmiljøundersøkelsen viser at ca. 2/3 av beboerne i en type flerfamiliehus med lette trebjelkelag (type I) er noe eller mye plaget av trinnlyd, selv om byggeforskriftenes krav stort sett er tilfredsstilt ved måling med bankemaskin.

Støyspektrene av reell trinnlyd (Fig. 3) viser systematiske forskjeller mellom ulike hustyper. Dette gjelder særlig i frekvensområdet 20 - 50 Hz, der 1/3-oktavnivåene for hustype I (trebjelkelag) ligger betydelig høyere enn for hustype III (etasjeskillere i betong).

For å rette opp misforholdet mellom byggeforskriftenes krav og opplevd sjenanse av trinnlyd i hus med lette trebjelkelag bør det stilles tilleggskrav som rammer slike konstruksjoner spesielt. I forbindelse med revisjon av byggeforskriftene er det foreslått å sette en maksimalgrense på 65 dB for 1/3-oktavnivåer. Med et slikt krav ville 9 av 12 etasjeskillere i hustype I ikke blitt godkjent, mens samtlige kontrollmålte etasjeskillere i hustype III ville blitt godkjent.

Resultatene som gjelder hus av type II er noe usikre. Boligmiljøundersøkelsen ble gjennomført kort tid etter innflytting, og innredning og gulvbelegg varierte svært mellom leilighetene (selvbyggerleiligheter).

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EN ALGORITM FÖR ELIMINERING AV BANDBREDDSFEL VID BERÄKNING AV TRANSMITTERAT BULLER.

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INTRODUKTION.

Det produceras tyvärr en mångfald av beräkningar i oktavband utan att det funderas över precisionen i beräkningarna.

Sytena med oktavbandsberäkningar är vanligen att

1) beräkna ljudnivån, LA (dB(A)) i mottagarpositionen.

2) bestämma åtgärder för att innehålla ett uppställt krav, vanligen också formulerat som ljudnivå (dB(A)).

Beräkning t ex av ventilationsbuller i ventilerade rum utförs vanligen just i oktavband. Här som i andra beräkningsfall är det av stor vikt, inte minst ekonomiskt, att man inte överdämpar i onödan eller att anläggningen överskrider gällande krav, trots omfattande åtgärder.

I samband med oktavbandsberäkningar uppkommer s k bandbreddsfel. BB uppkommer när spektrumformen hos en källa och ett dämpningselement har olika lutning. Värdena på nivå resp dämpning vid bandgränserna kan då avvika kraftigt från värdena vid mittfrekvenserna inom resp band. Målsättningen vid hanteringen av BB bör vara att hålla detta litet relativt andra (ofrånkomliga?) fel i beräkningen enligt principen att alla eliminerbara fel bör elimineras.

VAD MENAS MED BANDBREDDSFEL?

Oktavband har som bekant en bandbredd på 71 % av mittfrekvensen och utgör alltså ett tämligen stort frekvensområde. Vid beräkningar utförda med så stor bandbredd kan betydande **bandbreddsfel** uppstå. Bandbreddsfel uppstår:

a) när energin hos en ljudkälla faller kraftigt mot högre frekvenser, (som t ex för många fläktar) medan dämpningen hos transmissionselementet stiger mot högre frekvenser (som t ex för många ljuddämpare).

b) som ovan men när ljudkällans energi stiger och dämpningen faller med frekvesnen.

c) när ljudkällan innehåller toner som ligger vid bandgränserna och dämpningen hos transmissionselementet varierar inom frekvensbandet.

I Fig 1 visas bandbreddsfelet som funktion av skillnaden mellan källa och dämpning i "slope" i dB/oktav. Som synes kan man få stora fel även i en enda beräkningsoperation.

Populärt kan man säga att felet uppkommer genom att man räknar med dämparens egenskaper i den övre delen av oktavbandet medan den huvudsakliga källenergin ligger i den nedre delen av bandet (man jämför alltså fel saker med varandra). Ett annat sätt att utrycka saken är att en dämpare vars dämpning ökar med frekensen får sämre dämpning relativt katalogen om källspektrum faller med frekvensen (förutsatt att dämparen provats med vitt brus).

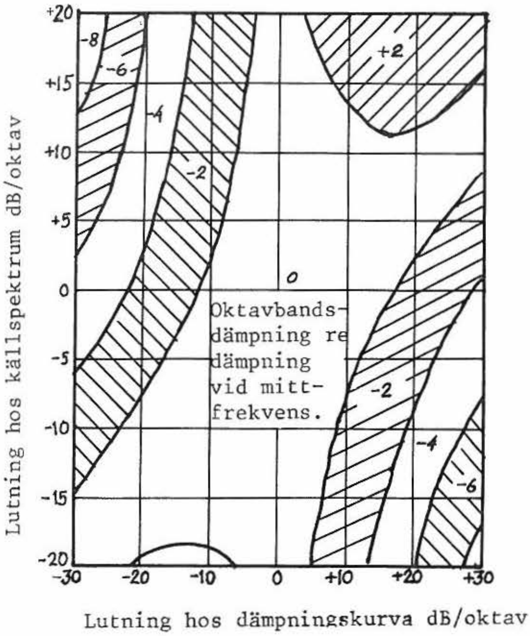

Fig. 1. Diagram för bestämning av dämpning i oktavband för bredbandsexcitering relativt dämpning för sinus-excitering vid mittfrekvenserna. /1./

HUR ELIMINERAR MAN BANDBREDDSFEL?

Det mest självklara sättet att eliminera bandbreddsfel är att minska bandbredden i beräkningarna. Vi har, för vissa industribullertillämpningar, genomfört beräkningar i tersband som jämförts med motsvarande beräkningar i oktavband. Resultatet visar att man ofta får en skillnad på ca 3 dB(A) i slutresultatet.

För vissa transmissionsberäkningar måste man dock driva bandbreddsminskningen mycket långt för att få felet tillräckligt litet. Detta gäller t ex för beräkningar på reaktiva ljuddämpare. Sådan långt driven bandbreddsminskning kräver datorberäkning. Här krävs stor minneskapacitet och god tillgång till maskintid.

VIAKs ALGORITM FÖR ELIMINERING AV BANDBREDDSFEL.

På VIAKs Akustikavdelning har utvecklats en egen algoritm för bandbreddsfelseliminering som ger tillräckligt litet bandbreddsfel för de flesta tillämpningar.

Målsättningen är ju som tidigare nämnts att bandbreddsfelet skall vara litet i förhållande till andra fel i beräkningen, som t ex brister på kvaliteten i indata, uteslutande av vissa källtyper generaliseringar i katalogdata eller osäkerheter i mottagarrummets ekvivalenta ljudabsorptionsarea.

VIAK-algoritmen för eliminering av bandbreddsfel programmerats i BASIC och länkats till beräkningsprogram för industri- och ventilationsbuller. Algoritmen visar sig ge tillräcklig minskning av bandbreddsfelet till priset av en mycket måttlig ökning av exekveringstiden i berörda program.

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MYNNINGSDÄMPNING VID ÖVERHÖRNING

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Inledning

S.k. mynningsdämpning uppstår genom reflexion vid mynningen. Denna beror på att strålningsresistansen avviker från vågimpedansen i kanal eller rör. Missanpassningen blir utpräglad med stor mynningsdämpning som följd om kanaltvärsnittet är väsentligt mindre än våglängden. Vid tillämpning på överhörning via kanal mellan rum har ibland viss tveksamhet rått om mynningsdämpning skall ingå för mottagarrummet. Vi skall här visa att normering på tillgänglig effekt respektive infallande intensitet gånger yta avgör vilket som gäller.

Teori

Den största effekt som kan gå in i kanalen kallas tillgänglig effekt och uppstår om inimpedansen är konjugerat komplex till strålningsimpedansen (Se Appendix A).

Tillgänglig effekt kan vara betydligt större än infallande effekt beräknad som intensitet gånger kanalens tvärsnittsyta. Det är detta som leder till den stora absorptionsytan hos en Helmholtzresonator. Vid resonans kan denna under gynnsamma omständigheter absorbera hela tillgängliga effekten. I överhörningsfallet är tillgänglig effekt även den största effekt som kan transmitteras till mottagarrum via kanalen (se figur 1).

Det totala dämpningen D som svarar mot tillgänglig effekt/transmitterad är alltid större än noll. De första två termerna är mynningsdämpning. De sista är längsdämpning resp inverkan av multipelreflexer mellan mynningarna.

$$D = D_{myn,1} + D_{myn,2} + D_1 + D_{mult} \qquad (1)$$

Mynningsdämpningarna påverkas något av fri rymdvinkel vid mynningarna. Multipeldämpningen kan sättas = 0 om längsdämpningen är > 10 dB.

För ljudisoleringsberäkning används reduktionstal R från infallande intensitet gånger yta och transmitterad effekt. Vi använder kanalens tvärsnittsyta och får följande reduktionstal om tvärsnitt hos kanalen är mindre än våglängd/4. Ω_1 är fri rymdvinkel vid mynning i sändarrum.

$$R = D_1 - 10 \log \frac{4\pi}{\Omega_1} + D_{myn,2} + D_{mult} - 6 \text{ dB} \qquad (2)$$

Om mynningen är fri blir första termen = 0. R ger enkelt sammansatt reduktionstal R' om man känner väggens reduktionstal R_v och yta S_v. Kanalens yta är S_k

$$S_v \cdot 10^{-R'/10} = S_v \cdot 10^{-R_v/10} + S_k \cdot 10^{-R/10} \qquad (3)$$

Man kan därför vilja ha fram R om D känd:

$$R = D - 10 \log \frac{4\pi}{\Omega_1} - 10 \log \frac{\rho c}{S_k \, Re[Z_{r,1}]} \qquad (4)$$

Vid definition av D med hjälp av tillgänglig effekt kommer strålningsimpedansen $Z_{r,1}$ för mynningen med. Denna ingår ju inte i infallande intensitet vilket förklarar skillnaden mellan R och D givet av formeln ovan (jfr Appendix A).

Vid låga frekvenser är realdelen av akustiska strålningsimpedansen betydligt mindre än specifika vågimpedansen i luft ρc dividerad med ytan S_k. Sista termen tenderar därför att göra R < D.

Slutsatser

Medtagandet av mynningsreduktion på sändarrumssidan beror på hur man normerar. Genom normering på tillgänglig effekt får man symmetri i uttrycken. Vid den vanliga typen av normering för reduktionstalsberäkning får formeln emellertid mynningsdämpning bara på mottagarrumssidan i lågfrekvensområdet.

Appendix A - Normering på tillgänglig effekt

Antag att innan mynningen till en kanal öppnas råder det blockerade ljudtrycket p_0 vid platsen för mynningen. Blockerat tryck har samma betydelse som elektromotorisk kraft i en tvåpol när denna anses ekvivalent till rum - mynning som akustisk enport. Med volymhastighet q (effektivvärde $\tilde{q}$) i mynningen får man trycket p vid mynningen genom superposition av inverkan från p_0 och q. Z står här för akustiska impedanser med "r" för utstrålning (se figur 2).

$$p = p_0 - q \cdot Z_r \tag{A1}$$

Dessutom gäller

$$p = q \cdot Z_{in} \tag{A2}$$

A_1 och A_2 ger

$$q = p_0 \;/\; (Z_r + Z_{in}) \tag{A3}$$

Mynningens akustiska strålningsimpedans blir tydligen inre impedans i den ekvivalenta tvåpolen - enporten. Effekt lämnas till inimpedansen i mynningen:

$$W_{in} = \tilde{q}^2 \cdot \text{Re}\,[Z_{in}] = \frac{\tilde{p}_0^2 \cdot \text{Re}\,[Z_{in}]}{|\,Z_r + Z_{in}\,|^2} \tag{A4}$$

Effekten blir maximal och lika med tillgänglig effekt om inimpedansen är konjugerat komplex till Z_r:

$$W_{till} = \frac{\tilde{p}_0^2}{4\ \text{Re}\,[Z_r]} \tag{A5}$$

Kvoten mellan effekterna blir symmetrisk m a p Z_r och Z_{in}. Man skulle få samma kvot om mynningen strålade ut till Z_r och Z_{in} vore då inre impedans i enporten - tvåpolen.

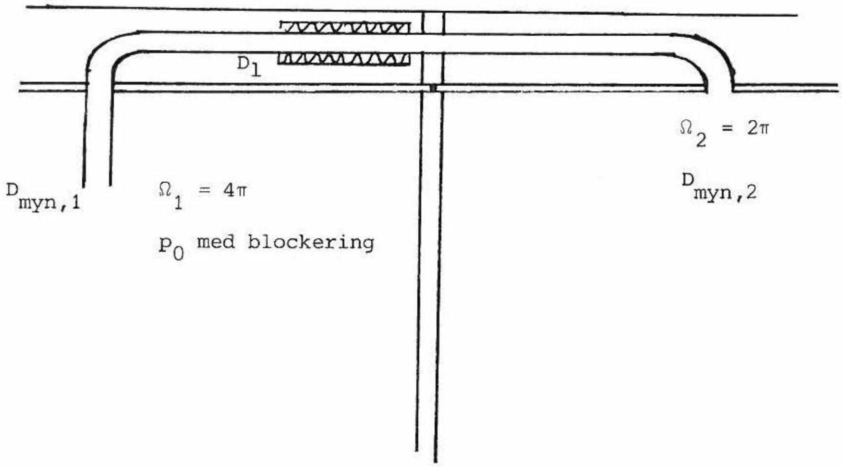
Detta leder till att mynningsdämpningen kommer med för instrålning vid normering på tillgänglig effekt. Vi sätter $Z_{in} = Z_k$, kanalimpedans:

$$\frac{W_{in}}{W_{till,rum}} = \frac{W_{ut}}{W_{till,kanal}} = \frac{\text{Re}[Z_r]\text{Re}[Z_k]}{|\,Z_r + Z_k|^2} = 10^{-D_{myn}/10} \tag{A6}$$

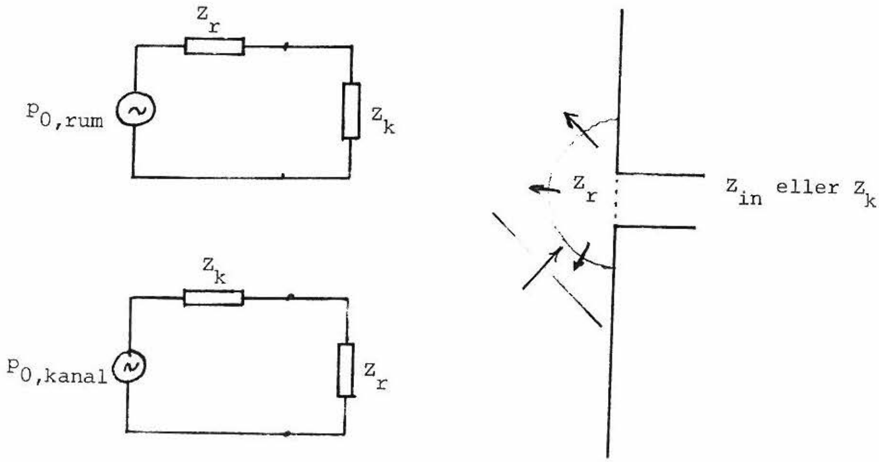
För beräkning av transmitterad effekt eller för övergång till reduktionstal behövs $\tilde{p}_0$. Medelvärde för $\tilde{p}_0$ i diffust fält beror av fri rymdvinkel Ω_1 så att man får följande där p är diffust tryck på godtycklig plats och < > betyder rumsmedelvärde.

$$\tilde{p}_0^2 = \frac{4\pi}{\Omega_1} \cdot \langle\, \tilde{p}^2 \rangle \tag{A7}$$

Figur 1.



Figur 2.



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FLANKETRANSMISJON VIA LIMTREBJELKER, ET FELTEKSEMPEL

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Arbeidet som her er utført i forbindelse med undersøkelse av flanketransmisjon via limtrebjelker startet opprinnelig som et rådgivningsoppdrag. Et utstillingslokale var ombygget til musikkskole, og man hadde fått problemer med for dårlig lydisolasjon mellom to øvingsrom. Årsaken viste seg å være flanketransmisjon via limtrebjelkene. Vi valgte å føre undersøkelsene litt lenger enn det selve oppdraget omfattet.

PROBLEMSTILLING

Opprinnelig problemstilling var å finne årsaken til den dårlige lydisolasjonen og foreslå utbedringstiltak. Da lydtransmisjonsveien var bestemt ønsket vi også å belyse forhold som har med overføring av lyd via limtrebjelker av mer generell karakter. Det er imidlertid viktig å være klar over at konklusjonene vi prøver å trekke skriver seg fra et eksempel og at de ikke kan betraktes som forskningsresultater.

MALESTED

Øvingsrommene ligger på hver sin side av en lagerromsone, se isometrisk figur. Sender- og mottaker-rom har volumer på h.h.v. 269 m^3 og 176 m^3. Skilleflatens areal defineres som vegg mot lagerrom og er på 30 m^2. Hvert rom har separat flytende sponplategulv. Himlingene er også separate (brutt ved veggene), og er spikret til lekter under et bjelkelag som ligger oppå limtrebjelkene. Himlingsmatrialet er pressede mineralull-plater. Veggene er gipsplater med enkelt stenderverk og en gipsplate på hver side. Fra rom til rom er det en tett vegg, lagerrom og så en vegg til med lyddør av klasse 35dB. Ventilasjon av rommene blir ført via lagerromsonen, og kanalsystemet er utført med et rikelig antall lydfeller og bend. Det er i alt fire limtredragere synlig fra rommene. To av disse ligger fritt og de to andre ligger over hver sin vegg. Limtrebjelkenes synlige dimensjoner er b x h = 150 x 900 mm.

LYDISOLASJON

Lydisolasjonen ble målt fra øvingsrom til lager (over tett vegg) og fra øvingsrom til øvingsrom. Resultatene ble h.h.v. Ia = 40 dB og 50 dB. Reduksjonstallene er vist i diagram 1. Utfra teoretiske betraktninger av de forskjellige lydveiene burde det være mulig å oppnå in Ia-verdi godt over 60 dB dersom det ikke forekom flanketransmisjon via limtrebjelkene. En sammenligning med lydisolasjon andre steder i bygget hvor det ikke forekommer limtrebjelker på samme måte, var med på å gi grunn til å tro at limtrebjelkene var årsak til den dårlige lydisolasjonen. Man kunne imidlertid stille spørsmål om det var limtrebjelkene alene, eller disse koblet med f. eks. himlingene som utgjorde lydtransmisjonsveien.

VIBRASJONSMALINGER

For å få en bedre oversikt over hvilke flater som var de effektive lydutstrålerne ble det målt akselerasjonsnivå på aktuelle flater i mottakerrommet. (Måling av akselerasjon i stedet for hastighet ble valgt for å sikre et tilstrekkelig signal over hele frekvensområdet). Det ble også målt på bjelker og takflater i senderrommet for å kunne sammenligne vibrasjonsnivåene og vurdere bl.a. knutepunktsdemping. Måleresultatene er vist i diagram 2.

AVSTRALING

Man ser allerede direkte av kurvene for vibrasjonsnivået at lydavstrålinger kommer fra taket. For å beregne lydstrålingen må man kjenne strålingsfaktorene for de aktuelle matrialene med tilstrekkelig sikkerhet. De mest aktuelle flatene viste det seg å være vanskelig å finne teoretiske strålingsfaktorer for:

a) -limtrebjelker med fri underkant.

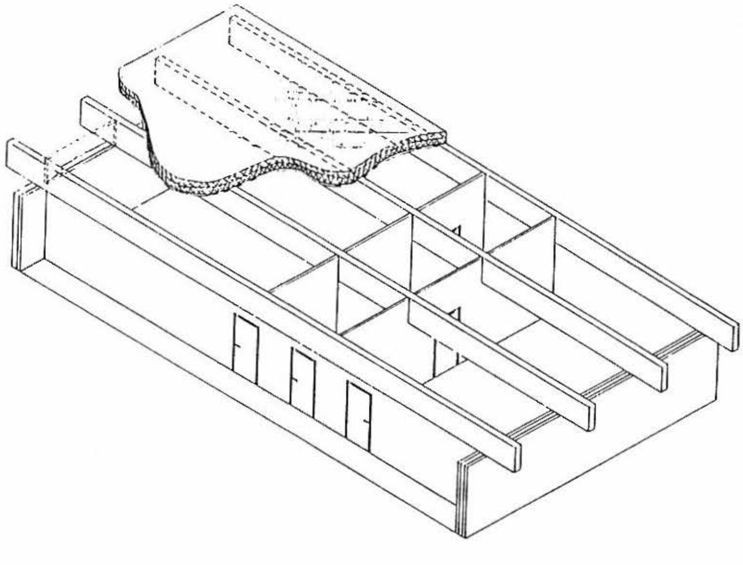
b) -pressede mineralullplater (absorbent = 0,6)

Utfra mulige egenvekter og stivheter for tre, ble grensefrekvensen for bøyningsbølge om svake akse i bjelken beregnet til å ligge mellom 100 og 160Hz. Strålingsfaktoren, ble etter dette satt til 1 for hele frekvensområdet 100-3150 Hz. Dermed ble det også sett bort fra kopling mellom bjelkenes for- og bakside. Strålingsfaktoren fra de pressede mineralullplatene ble beregnet som for tette plater med fg = 2000 Hz og dimensjon 0,6 x 0,6 m. Vi følte oss sikre på at dette ga en strålingsfaktor som er lik eller større enn den reelle. Beregnet avstrålt lydtrykknivå fra endel av flatene er vist i diagram 3, hvor også målt lydtrykknivå er tegnet inn. Utfra kurvene kan man se at det er avstråling fra limtrebjelkene som dominerer lydnivået i mottakerrommet. Kurvene viser også at beregnet avstrålt lydtrykknivå er høyere enn målt lydtrykknivå i mottakerrommet.

Dersom man antok at limtrebjelkene var alene om å stråle lyd til mottakerrommet ville strålingsfaktorene bli sett som vist i diagram 4.

Knutepunktsdempingen i vårt litt spesielle tilfelle fremkommer ved subtraksjon av kurvene for bjelkene i sender- og mottakerrommet i diagram 2.

Det synes å være grunn til å etterlyse mer data for avstråling fra absorberende matrialer. Her hadde vi et konkret tilfelle hvor det ville vært aktuelt å vite noe om strålingsegenskapene til pressede mineralullplater. Det er ikke vanskelig å tenke seg andre situasjoner hvor det er tilsvarende data kan være aktuelle.



Over: Isometrisk figur som viser øvingsrommenes plassering, lagerromsonen og limtrebjelkene.

FORENKLET BEREGNINGSMÅTE

Vi foretok en forenklet beregning av flanketransmisjonen ved å betrakte limtrebjelkene som enkeltvegger med strålingsfaktor lik 1 fra 100 - 3150 Hz og knutepunktsdemping lik 0. Ved å korrigere for arealforhold fant vi et tilsynelatende redukstall for skilleveggen med en Ia-verdi lik 47 dB. (Se diagram 1).

UTBEDRINGSTILTAK

For å forbedre lydisolasjonen foreslo vi å kle inn limtrebjelkene med 1 lag gips på 50 x 50 mm stenderverk i avstand 25 mm fra bjelken, og med 75 mm mineralull i hulrommet. Med usikkerheten i avstråling fra andre flater, spesielt taket, antok vi en mulig forbedring på 5-8 dB i Ia-verdien. Etter at tiltaket var utført ble Ia-verdien målt 58 dB (en forbedring på 8 dB).

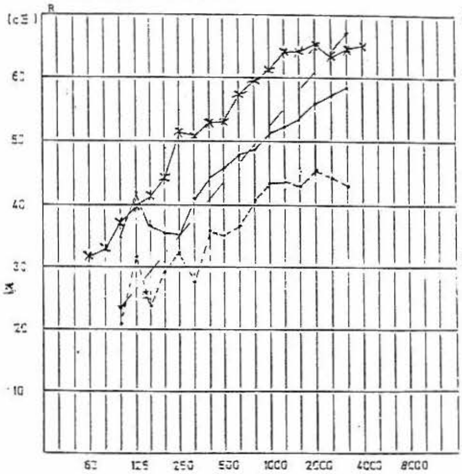
KONKLUSJON

Erfaringene fra dette prosjektet kan sammenfattes i følgende punkter:

- Strålingsfaktor for høye limtrebjelker kan i forenklede betraktninger settes tilnærmet lik 1 i hele frekvensområdet 100-3150 Hz. Dette gir litt for høye verdier - særlig i lavfrekvensområdet. Her synes imidlertid koblingen mellom de to bjelkesidene å gjøre seg mer gjeldende enn virkningen av koinsidens (grensefrekvens).
- Knutepunktsdempingen blir liten og kan forenklet settes lik null når limtrebjelken krysser en lettvegg.
- Enkle flanketransmisjonsbetraktninger synes med disse antakelsene å gi gode anslag for oppnåelig lydisolasjon.

DIAGRAM 1

Lydisolasjon mellom øvingsrom



——— Reduksjonstall mellom øvingsrom (110/131), Ia = 50 dB

- - - - Reduksjonstall øvingsrom/lager (110/121), Ia = 40 dB

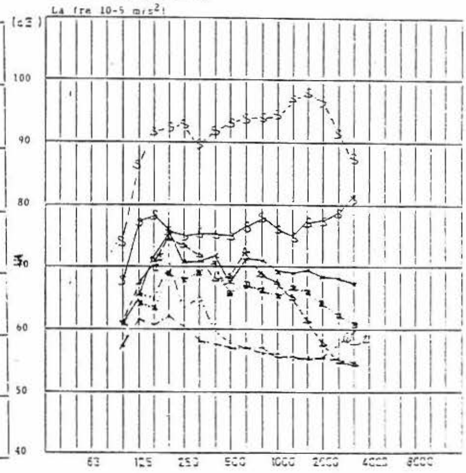
— · — Teoretisk lydisolasjon utfra flanketransmisjonsbetraktningen Ia = 47 dB

X——X Målt lydisolasjon etter utbedring: Ia = 58 dB

DIAGRAM 2

Målte akselerasjonsnivåer

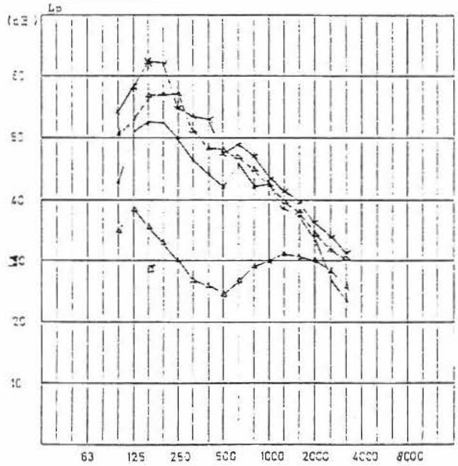
La fra 10-5 m/s^2)



Vegg		Mottakerrom
Bjelke		Mottakerrom
Takflate		Mottakerrom
Bjelke i flankerende vegg		Mottakerrom
Flankerende vegg under bjelken		Mottakerrom
Bjelke		Senderrom
Takflater		Senderrom

DIAGRAM 3

Lydnivå i mottakerrom



beregnet utfra:

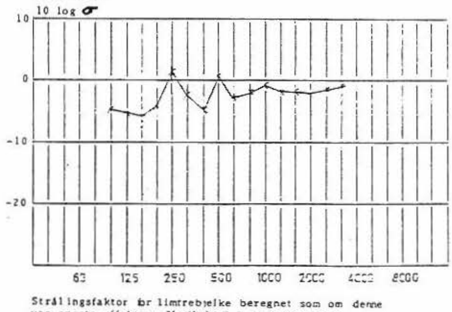
Vibrasjonsnivå på bjelke

Vibrasjonsnivå i takflaten

Vibrasjonsnivå på veggen

Lydnivå i rommet, MÅLT

DIAGRAM 4



Strålingsfaktor for limtrebjelke beregnet som om denne var eneste effektive "lydkilde" i mottakerrommet.

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EDB-PROGRAM TIL BEREGNING AF LYDISOLATION I BYGNINGER

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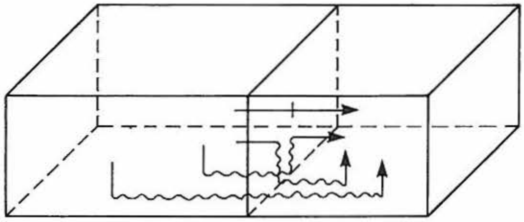
Computerteknologiens udvikling har gjort det muligt at anvende forholdsvis komplicerede beregningsmetoder til nem og hurtig beregning af lydisolation i bygninger. Der er på denne baggrund udviklet et regneprogram, som kan afvikles på et PC-anlæg og dermed benyttes af en bred kreds blandt fabrikanter af byggematerialer, rådgivende ingeniører og andre, som er beskæftiget med projektering af byggeri.

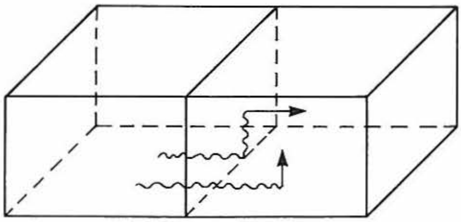
Formålet med programmet er i første række at kunne optimere bygningskonstruktioner i forbindelse med konkrete krav til lydisolationen. Dette sker dels ved en analyse af lydens vigtigste transmissionsveje og dels ved en beregning af luftlydisolation og trinlydniveau, der er mere præcis end ved p.t. udbredte metoder til forudsigelse af lydisolation.

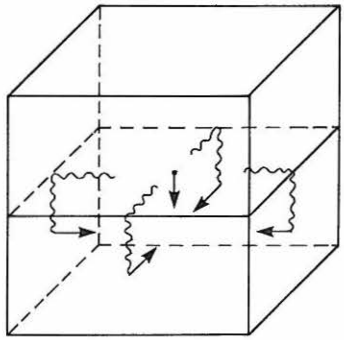
Teoretisk grundlag

Blandt de mere avancerede metoder til beregning af lydisolation mellem to rum er oftest anvendt statistisk energianalyse (SEA) eller en metode, som er udviklet af E. Gerretsen på TNO, Holland [1], [2]. Det udarbejdede edb-program er baseret på sidstnævnte metode, som forudsætter diffuse lydfelter og inddrager lydtransmission via skilleflade samt flankerende flader. Figur 1 viser de transmissionsveje, som indgår ved beregningerne. Ved beregning af luftlydisolation vandret og lodret tages hensyn til samme antal transmissionsveje.



1. Luftlydisolation vandret. Der er vist transmissionsveje via gulv og skillevæg. Idet der desuden medregnes lydtransmission via de øvrige flankerende flader, inddrages i alt 13 transmissionsveje i beregningerne.





2. Trinlydniveau vandret

3. Trinlydniveau lodret

Figur 1. —— luftlyd ∼∼ strukturlyd

Ved beregning af luftlydisolationen forudsættes kendskab til fladernes reduktionstal og knudepunktsdæmpningerne ved udbredelse af vibrationer gennem samlingerne. Disse størrelser kan bestemmes ud fra teoretiske formler eller fx resultater af laboratoriemålinger, hvorefter det tilsyneladende reduktionstal for lydtransmission via flade i og j beregnes af (1):

$$R'_{ij} = (R_i + R_j + D_{ij} + D_{ji})/2 + 10 \log \frac{S_0}{\sqrt{S_i S_j}} \qquad (1)$$

R_i : Reduktionstallet for flade i

R_j : Reduktionstallet for flade j

D_{ij}: Knudepunktsdæmpning ved udbredelse af vibrationer fra flade i til flade j

D_{ji}: Knudepunktsdæmpning ved udbredelse af vibrationer fra flade j til flade i

S_0 : Areal af skilleflade

S_i : Areal af flade i

S_j : Areal af flade j

Det tilsyneladende trinlydniveau - fra flade i - for udstråling fra flade j beregnes af (2):

$$L'_{n,ij} = L_F + 10 \log f + 10 \log \sqrt{\frac{\tau_i \, re(Y_i)}{m_i}} + 10 \log \sqrt{\frac{S_j}{S_i}} - (R_j + D_{ij} + D_{ji})/2 - 36 \qquad (2)$$

L_F : Kraftniveau for slag fra standardiseret bankemamaskine

f : Frekvens

τ_i : Efterklangstid for vibrationer i flade i

$re(Y_i)$: Realdelen af den mekaniske admittans for flade i

m_i : Masse pr. m^2 for flade i

(Betydning af øvrige symboler som i (1)).

Det udarbejdede edb-program

Programmet er primært udviklet med henblik på anvendelse i forbindelse med typiske danske bygningskonstruktioner. Reduktionstallet beregnes ofte teoretisk, men stammer dog i nogle tilfælde fra resultater af laboratoriemålinger. Knudepunktsdæmpninger bestemmes normalt ud fra teoretiske eller empiriske formler angivet af Gerretsen [2]. Mekanisk admittans for gulvkonstruktioner beregnes teoretisk, hvilket forudsætter massive dæk.

Der er mulighed for at indføre tynde gulvbelægninger, svømmende gulve og trægulve på strøer, idet programmet benytter laboratoriemålte trinlyddæmpninger og forbedringer af reduktionstal.

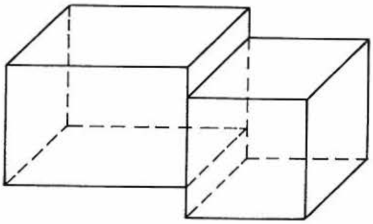
Der kan desuden indføres nedhængte lofter og forsatsvægge eller -beklædninger, idet forbedringerne af fladernes reduktionstal bestemmes teoretisk.

Skillevæggen mellem rummene kan være enkelt eller dobbelt.

I forbindelse med indlæsning af data kan vælges mellem en række almindeligt forekommende materialer og konstruktionstyper, hvorefter programmet bestemmer de størrelser, som indgår i (1) og (2). Alternativt kan disse størrelser indtastes af brugeren.

Indlæsningen foregår således, at der skal indtastes et minimum af data, fx såfremt lydisolationen skal beregnes mellem to ens rum. Rummenes indbyrdes beliggenhed og fladernes dimensioner kan dog varieres, og der kan fx udføres beregninger for situationen vist i figur 2.

Reduktionstal og trinlydniveau beregnes for 1/3 oktavbånd i frekvensområdet 100-3150 Hz. Desuden beregnes de vægtede størrelser (R_w, R'_w og $L'_{n,w}$) i henhold til DS 2186.



Figur 2.

Der kan udskrives alle indlæste data samt følgende beregningsresultater:

- Knudepunktsdæmpninger
- Reduktionstal for de enkelte flader
- Tilsyneladende reduktionstal for de enkelte transmissionsveje
- Tilsyneladende reduktionstal for lydtransmission via de enkelte flader
- Tilsyneladende reduktionstal for al lydtransmission
- Tilsyneladende trinlydniveau for udstråling fra de enkelte flader
- Tilsyneladende trinlydniveau for al lydudstråling

Regneprogrammet er skrevet i Pascal og udviklet på en Olivetti M24 Personal Computer (IBM-kompatibel). Indlæsning sker ved hjælp af tastatur, og udlæsning sker til skærm samt eventuel printer. Programmet kan afvikles på en computer med en hukommelse på 256 k (RAM).

Edb-programmets anvendelse

En analyse af de vigtige transmissionsveje ved lydudbredelse i bygningskonstruktioner foretages let ved hjælp af det udarbejdede program. Der er dermed skabt et grundlag for mere effektivt at kunne forbedre lydisolationen.

Nøjagtigheden af beregningsresultaterne afhænger af bygningskonstruktionerne. Lydisolationen beregnes mere præcist ved massive konstruktioner med stive samlinger end ved fx dobbelte skillevægge og komplicerede facadeløsninger. En generel anvendelse af programmet forudsætter en nærmere undersøgelse af beregningsresultaternes nøjagtighed samt indarbejdelse af yderligere empiriske data, ikke mindst vedrørende knudepunktsdæmpninger i typiske danske bygningskonstruktioner. Denne verifikation og udbygning af programmets database påregnes gennemført i løbet af 1-1½ år, hvorefter programmet vil blive tilgængeligt for projekterende teknikere via en abonnementsordning.

[1] Gerretsen, E. Calculation of the sound transmission between dwellings by partitions and flanking structures. Applied Acoustics Vol. 12 No. 6, 1979.

[2] Gerretsen, E. Berekening van luchtgeluid- en contactgeluidisolatie. TPD-rapport 007.101/2, Delft, 1983.

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FÖRBÄTTRING AV LJUDISOLERINGEN I OMBYGGNADSOBJEKT MED TRÄBJÄLKLAG

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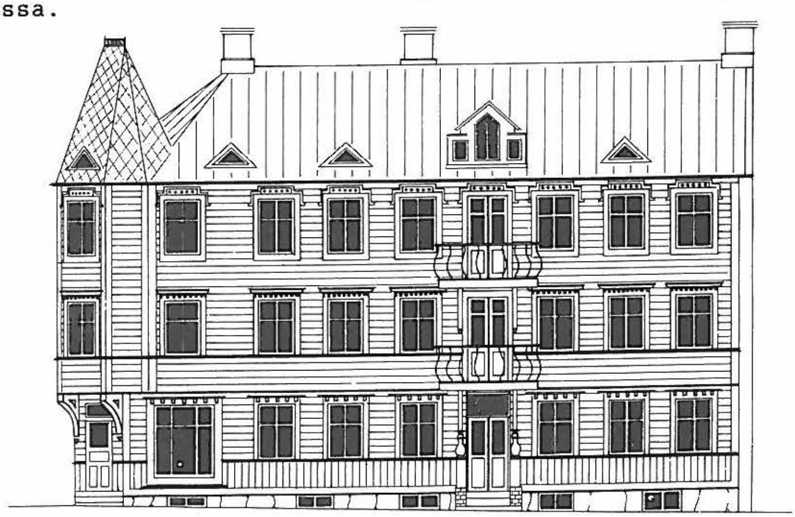
Bakgrund

Provningsanstalten har under perioden 8401-8607 drivit ett byggforskningsprojekt med titeln "Brand- och ljudfrågor rörande träbjälklag vid ombyggnad av flerfamiljshus. Projektet har bl a omfattat ljudisoleringsmätningar i sex flerfamiljshus före respektive efter ombyggnaden av dessa.



Fasadritning upprättad 1932

Effekten av ombyggnadsåtgärderna

Genom att studera ljudisoleringsresultaten för de sex objekten och genom att bestämma skillnaden mellan isoleringsvärdena efter respektive före ombyggnaden så har det varit möjligt att dra följande slutsatser om de tillämpade åtgärderna:

- Ett nedhängt undertak av 13 mm gips på stålreglar, pendlat med stålband till det gamla taket och utan mineralull i luftmellanrummet ger normalt en stor förbättring av ljudisoleringen. Förbättringen är normalt större än 10 dB om luftspalten är minst 100 mm. Störst effekt erhålles i trähus som från början har en mycket dålig ljudisolering.

- Ett mineralullsskikt mellan undertaket och det gamla taket tycks inte innebära någon garanterad vinst.

- Om man inte tillför ett nedhängt undertak eller någon annan verksam förbättring, så riskerar man oförändrade eller till och med försämrade ljudisoleringsvärden.

- Eftersom man hela tiden företagit olika kombinationer av åtgärder speciellt när det gäller golven i olika rum, är det svårt att uppskatta några generella förbättringsvärden för de vanligaste golvåtgärderna. Klart är emellertid att ljudisoleringen förbättras betydligt mindre av de aktuella golvåtgärderna än av undertaken.

Exempel

Som en illustration av vad som sagts ovan återges ett exempel i figur 1. Figuren presenterar den stegljudsisoleringsförbättring som erhållits med ett nedpendlat undertak. Observera att man med undertaket lever upp till nybyggnadskravet i Svensk Byggnorm.

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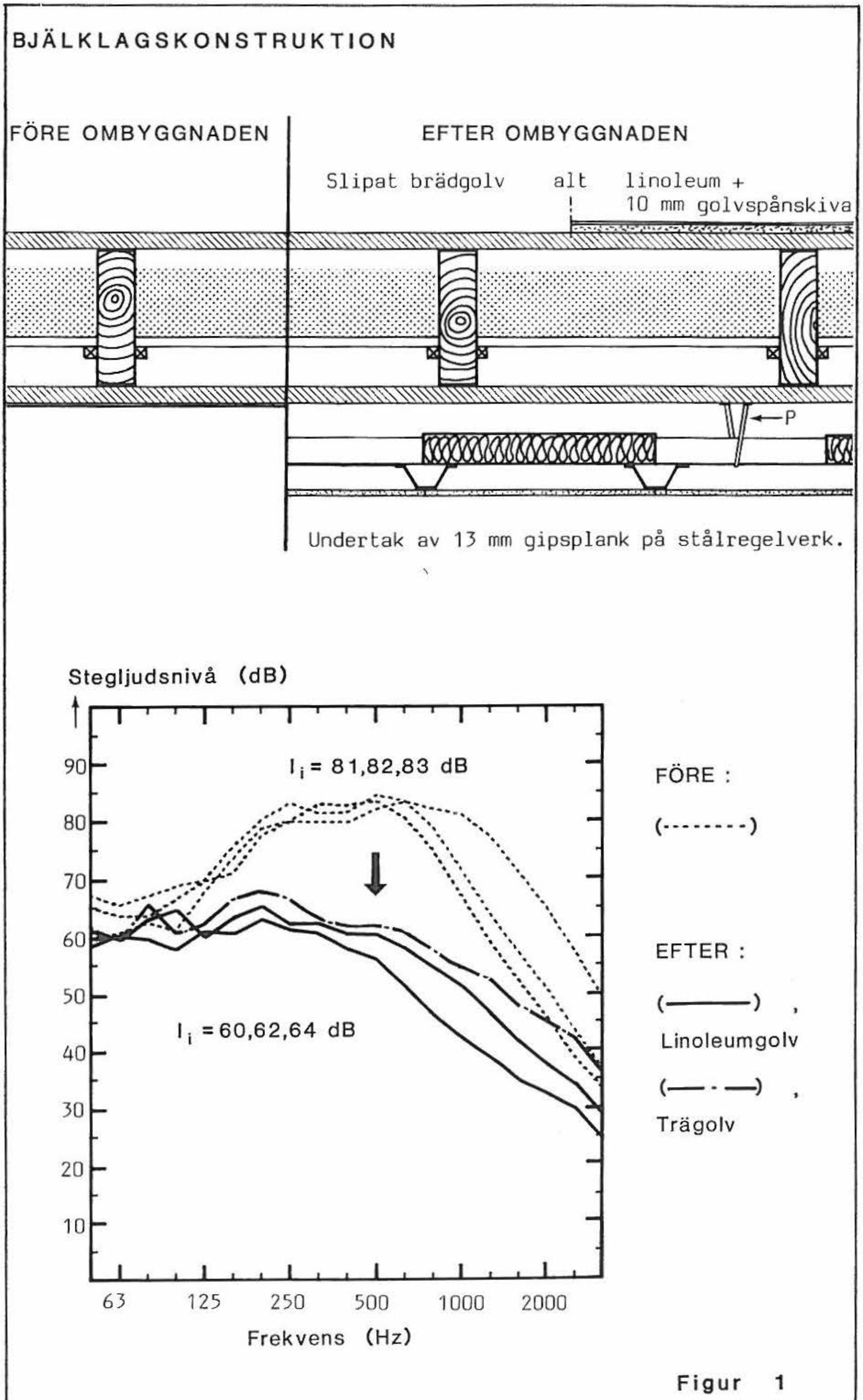
BJÄLKLAGSKONSTRUKTION
FÖRE OMBYGGNADEN
EFTER OMBYGGNADEN
Slipat brädgolv alt linoleum +
10 mm golvspånskiva
P
Undertak av 13 mm gipsplank på stålregelverk.
Stegljudsnivå (dB)
90
80
70
60
50
40
30
20
10
I_i = 81,82,83 dB
I_i = 60,62,64 dB
63
125
250
500
1000
2000
Frekvens (Hz)
FÖRE :
(--------)
EFTER :
(———) ,
Linoleumgolv
(—·—) ,
Trägolv

Figur 1

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HUDIKSVALLS FOLKETS HUS - ISOLATION OF RAILROAD VIBRATIONS

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In Hudiksvall a new Folkets Hus (Community Assembly Hall), is presently being build. The chosen site, near a railroad track, necessasitates a complicated building with certain parts isolated from ground vibrations.

To obtain a solid basis for our calculations, vibration measurements were made at an early stage in existing buildings. When the first parts of the structure was being built, we made complementary measurements on these constructions.

The calculations of the properties of the rubber bearings were made according to a mobility analogy shown in the figure below.

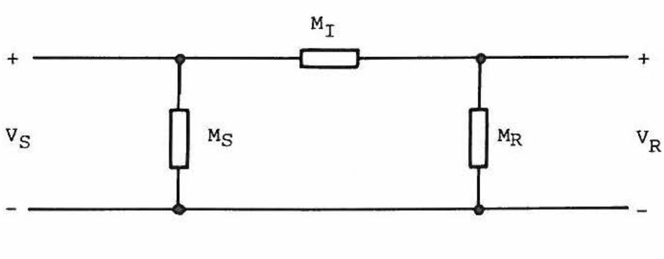


Figure 1.

M_S = point mobility of the foundation pillar

M_I = transmission mobility of the bearings

M_R = point mobility of the concrete slab
V_S = vibration velocity of the foundation pillar
V_R = vibration velocity of the concrete slab

This analogy leads to the following formula for the velocity transmission loss,

$$\Delta L_V = -10 \log (1+|M_I/M_R|^2) \qquad (1)$$

The mobility of a finite slab is higher than that of an infinite slab at certain resonance frequencies. This can be approximated, for the worst case, by the following formula:

$$M_R = M_\infty(1+ \sqrt{(f_g/4f)}) \qquad (2)$$

where f_g depends on the loss factor, area, thickness, density and bending stiffness of the slab. This means a slight rise in mobility at lower frequencies.

The transmission mobility of rubber bearings is described in the figure below, where levels and cut off frequencies depend on the dynamic stiffness, mass and loss factor of the bearing.

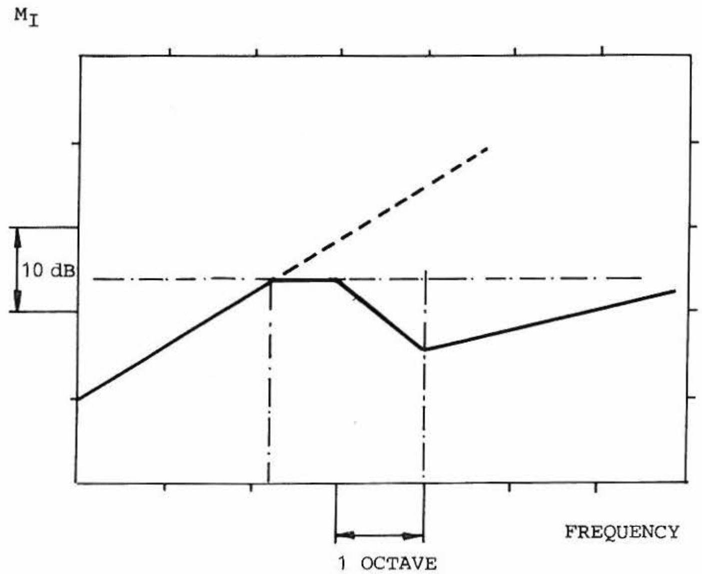
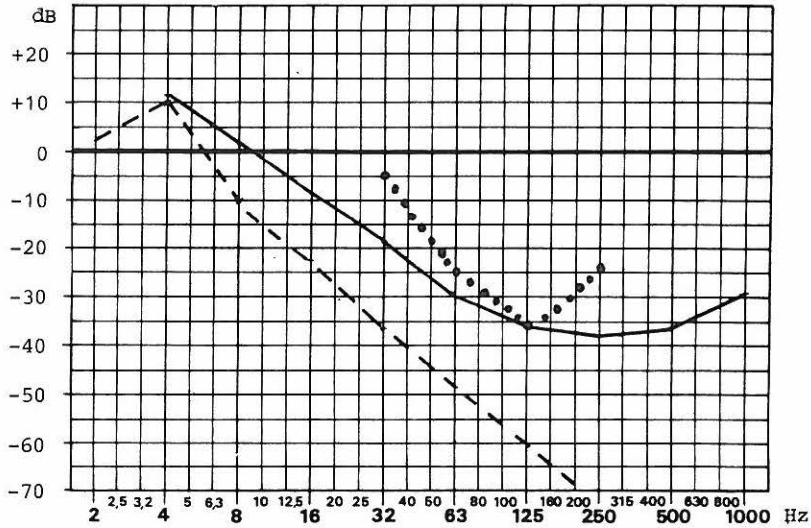


Figure 2.

In figure 3, below, the necessary transmission loss, to obtain the appropriate levels, is shown together with the calculated loss according to the mobility analogy. In the figure is also shown the transmission loss, according to a conventional calculation with a point mass on an ideal spring. At the most important frequency, 125 Hz, the mass-spring analogy will overestimate the loss by approximately 25 dB in comparision with the mobility analogy.



• • • • necessary transmission loss
——— calcualted t.l. according to mobility analogy
- - - - calculated t.l. according to mass-spring analogy

Figure 3. Velocity level transmission.

During the autumn of 1986 we hope to make measurements of the transmission loss at the building site.

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DÄMPNING AV STOMLJUD OCH VIBRATIONER FRÅN JÄRNVÄG - ETT PROJEKTERINGSFALL

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Sverige

INLEDNING

Vid projekteringen av ett ännu inte uppfört större kontors- och hotellbyggnadskomplex i Stockholm, kv Överkikaren, ställdes vi inför en avgörande problemställning.
Järnvägen från Stockholm mot södra Sverige skulle nämligen passera rakt igenom byggnadernas nedre våningsplan.

Följden var att framför allt de stomburna ljuden skulle bli avsevärt högre än vad som kunde anses tillfredsställande. Efter ljud- och vibrationsmätningar i intilliggande byggnader och i befintlig järnvägstunnel samt i mark beräknades ljudnivån till cirka 55 dB(A) i byggnadernas nedre våningar på grund av enbart stomljudstransmitterat ljud. Till detta kom även de luftburna trafikstörningarna via fasader och fönster.

LJUD OCH VIBRATIONSKRAV

Den ekvivalenta ljudnivån i hotellrummen bestämdes till maximalt 30 dB(A) medans den momentana ljudnivån inte får överstiga 40 dB(A). Det stomburna bullret kommer att stråla ut via väggar, golv och tak vilket ökar den psykiskt-subjektiva störningsupplevelsens i jämförelse med om ljudet enbart kommit från exempelvis ett fönster. En strävan att närma sig maximalt 35 dB(A) i momentan ljudnivå finns därför.

TVÅ SEPARATA LÖSNINGAR FICK TILLGRIPAS

Vibrationsdämpningen måste vara minst 15-20 dB inom det frekvensområde som ur stomljudssynpunkt är dominerande. Vid beräkningar och jämförelser med tidigare och liknande frågeställningar konstaterades att det var osannolikt att klara detta med endast en vibrationsdämpande åtgärd. Två separata åtgärder har därför projekterats.

A. Att i järnvägsbanvallen lägga ett fjädrande skikt med en dynamisk resonansfrekvens på ca 25 Hz med avsikten att dämpa det dB(A)-bestämmande frekvensområdet 100-250 Hz med minst 10 dB och högre frekvensen med minst lika mycket.

B. Att ställa delar av byggnadsstommen på fjädrande naturgummilager med en dynamisk resonansfrekvens på ca 30 Hz. Beräknad dämpning >10 dB över 100 Hz.
Se figur 1

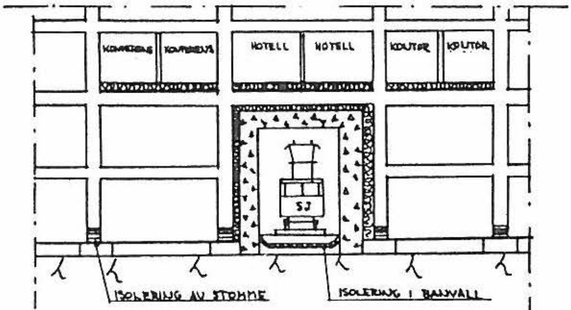

Fig.1

ÅTGÄRDER I BANVALLEN

Att finna fjädrande material som är lämpliga att använda i järnvägsbanvallar krävde ett ganska omfattande arbete. Många parametrar skall uppfyllas. Materialet skall ha rätt fjädringsstyvhet, tåla höga tryck, vara beständigt mot vatten, kemikalier, frost och åldring mm och ha en mycket definierad sammantryckning och krypning med hänsyn till de krav som SJ ställer. Dessutom skall materialet ha lägsta möjliga kostnad.

Mineralull avfördes pga otillräcklig hållfasthet samt pga osäkerheten beträffande åldringsbeständigheten just i denna krävande miljö.
Olika naturgummimaterial avfördes också pga felaktig fjädringsstyvhet.
De två materialen som studerades vidare var en skivprodukt av elastisk styrelcellplast (Ethafoam) och en skivprodukt av elastisk polyuretan (Sylomer).
I första steget koncentrerades intresset på Ethafoamen eftersom priset var endast ca 25% av Sylomermaterialet.
Ethafoamskivorna utsattes för dynamiska lastprov med upp till 1 milion lastväxlingar. Materialet var inte acceptabelt vad gällde krypning och hystereseseffekt i σ-ε försöken. Materialet komprimerades permanent efter ett större antal lastväxlingar.
Därefter kvarstog Sylomerprodukten som enda acceptabla material. En speciell skiva togs fram som var försedd med styvare och skyddande ytterskikt eftersom skivan skulle täckas av krossat stenmaterial (ballast) på både över- och undersida. Se figur 2.

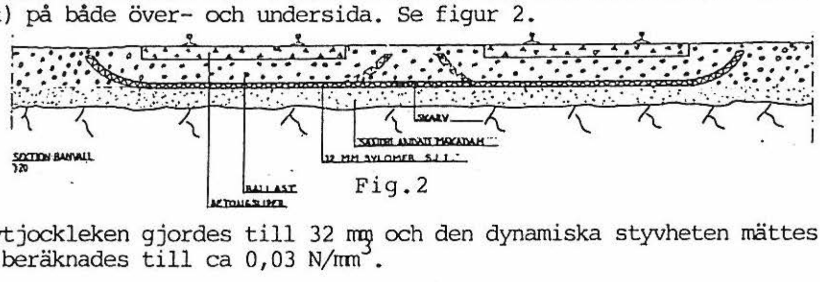

Fig.2

Skivtjockleken gjordes till 32 mm och den dynamiska styvheten mättes och beräknades till ca 0,03 N/mm^3.

Med aktuella lastförhållanden och med skivan lagd ca 300 mm under slipers blir den dynamiska resonansfrekvensen ca 25 Hz. Vibrationerna i den befintliga tunneln mättes både före och efter inläggningen av dämpskivorna. En avsevärd dämpning erhölls. Resultatet överensstämde väl med beräknad och förväntad dämpning. Se figur 3.

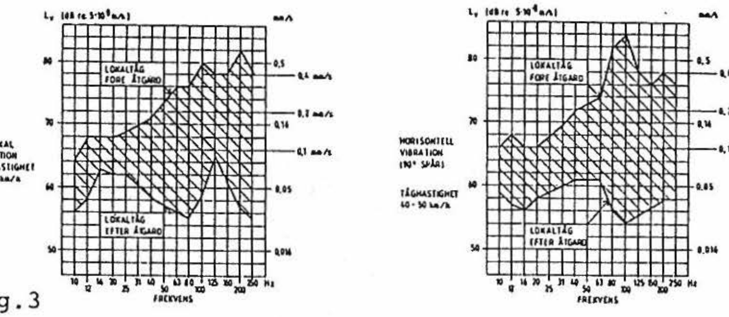

Fig.3

Som diagrammen visar erhålls 10-20 dB dämpning både för vertikalt och horisontellt riktade vibrationer i frekvensområdet 50-250 Hz.

Vid själva läggningsarbetet fanns flera praktiska svårigheter att lösa som var avgörande för isolermattans funktion. Exempelvis måste mattdelarna fogas och limmas mycket noga för att undvika akustisk kortslutning. Arbetet var mycket tidsreglerat eftersom järnvägstrafiken var i drift under hela projektet. Personalen som praktiskt skulle utföra arbetena var mycket ovana med denna problemställning och på de krav som ställdes på utförandet. Noggrann kontroll av akustiskt kunniga ingenjörer var därför nödvändigt.

DÄMPNING AV BYGGNADSSTOMMEN

Att bygga byggnadsstommen, helt eller delvis, på fjädrande gummilager är till dags dato mycket ovanligt i Sverige. Kunskapen bland byggnadskonstruktörer och entreprenörer är därför ringa. Den första reaktionen inför en sådan tanke mottogs därför med mycket stor tveksamhet. Övertalning och argumentering tog i detta fall en avsevärd tid. När denna del av arbetet var gjord, vidtog själva projekteringen.
Vad som först konstaterades var att hela byggnaden svårligen kunde byggas på naturgummilager, eftersom mark och grundförhållandena är komplicerade i området. Dessutom är denna typ av åtgärder kostsamma.
Det bestämdes att endast pelare och/eller väggskivor som var placerade inom 20 meters avstånd från järnvägen skulle ställas på naturgummilager. (Avståndsdämpningen i marken på 20 meters avstånd beräknades och mättes till 10-15 dB, dvs lika mycket dämpning som förväntades i bärlagren.)
Bärlagret bestämdes till en storlek som kan kombineras för varierande lastfall. Den statiska nedfjädringen bestämdes till 2,5 mm (f_o-stat = = 15Hz) vilket bedöms motsvara ca 30 Hz dynamisk resonansfrekvens.
Elementen valdes till 300x400x120 mm med 60 shore naturgummi. Varje element skall belastas så nära som möjligt med 10 MPa för att erhålla rätt statisk nedfjädring.
Naturgummi valdes för dess fördelar vad gäller bla åldring, krypning och dynamiska styvhet. Lagren är uppbyggda av tolv skikt gummi som separeras och vulkas i stålplåtar för att få rätt formfaktor (S) för given last och nedfjädring. Gummilagren måste skyddas för bla kemikalier och brand och kläs därför in med stenull.

Vidare måste lagren kunna inspekteras och bytas ifall skador skulle uppstå. Väggdelarna mellan lagren utformas så att huset endast kan sjunka 20 mm vid en eventuell katastrofsituation. Se fig 4 och 5. De horisontella krafterna från bla vind, krypning och exentriciteter måste tas upp i speciella bärlager eller i kloroprenanlägg och kloroprendubbar. I vissa punkter fås omväxlande horisontella tryck eller drag och här måste vertikalt sittande bärlager ta kraften i båda riktningar

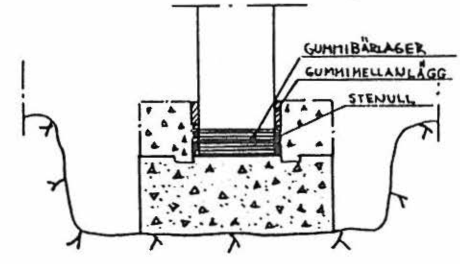

Fig.4

Arbetets praktiska genomförande måste i många delar lösas under projekteringen.
Exempelvis måste väggarna gjutas i flera etapper så att den nedre väggdelen hinner få erforderlig böjhållfasthet innan ytterligare pågjutning kan göras. Detta för att inte väggskivorna skall trycka sönder formmaterialen som skiljer husstommen från marken.

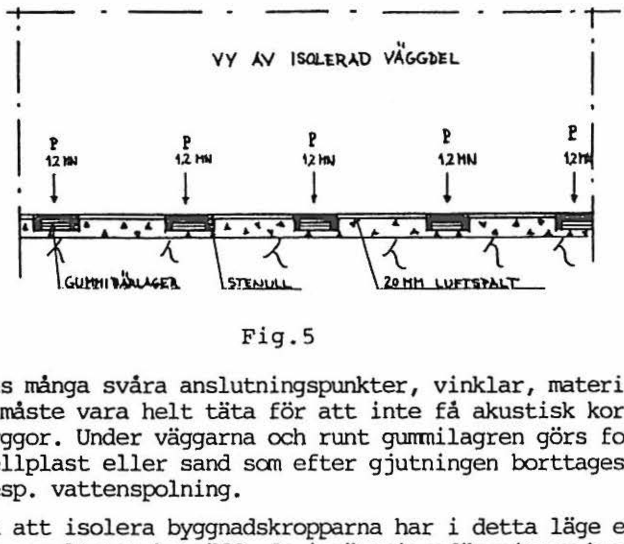

Fig.5

Vidare finns många svåra anslutningspunkter, vinklar, materialövergångar som måste vara helt täta för att inte få akustisk kortslutning via gjutbryggor. Under väggarna och runt gummilagren görs formar av antingen cellplast eller sand som efter gjutningen borttages med kemikalier resp. vattenspolning.

Arbetet med att isolera byggnadskropparna har i detta läge endast påbörjats och resultatet beträffande insättningsdämpningen kan därför inte redovisas.

Det skall betonas att vi har upplevt projekteringen som komplicerad - inte vad gäller själva vibrationsdelen - utan de praktiska lösningarna och främst hur man skall ta ned horisontella krafter till grunden utan att isoleringen försämras.

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DYNAMIC STIFFNESS AND MOBILITY FOR VIBRATION ISOLATORS

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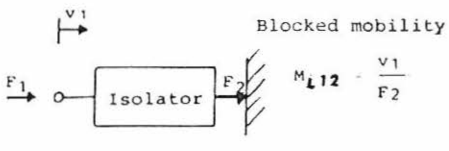
1. Introduction

A test arrangement has been developed in which dynamic properties of vibration isolators may be measured. Measured results and discussions of material properties are presented in the paper.

In calculations of vibration isolation using the simple mass/stiffnes one degree of freedom-modell often static stiffnes for the isolators are known. However the relationship between dynamic and static stiffnes have to bee estimated, and our measurements shows that this factor may vary a lot.

In mobility calculations of vibration isolation the complete description of the isolators is the four pole modell. However in most cases the blocked mobility for the isolators is a sufficient description, see for instance (2). In our work we have concentrated on blocked mobility data in normal direction for the isolators when expoced to static loads.



The principle of the test arrangement is shown in (1). There is a lower frequency limitation from which the dynamic properties may be measured.

2. Damping mechanisms in rubberbased materials

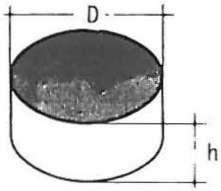
Although many textbooks in vibration isolation still introduces damping as a viscous dashpot, the usual assumption now is that rubberbased materials behaves hysteretic. A modell which combines viscous and hysteretic behaviour is experimentally verified in (3).

On the basis of theoretical studies and measurement, we conclude from our work that for all practical purposes the damping mechanism is hysteretic.

3. Dynamic stiffness and blocked mobility for rubberbased isolators

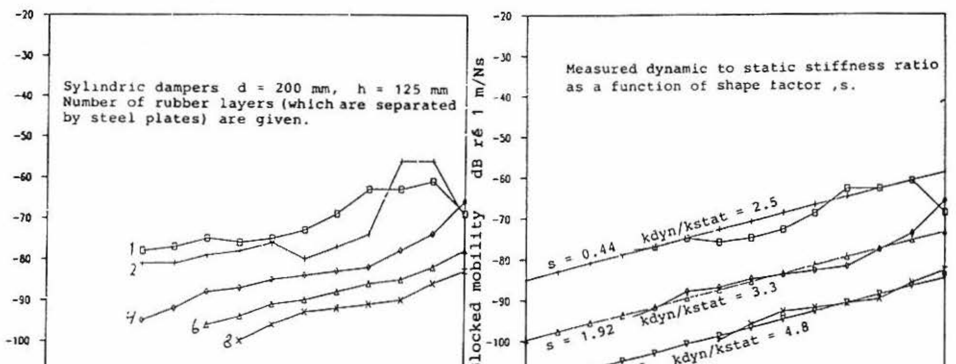
The principle that the static stiffness depends on the loading is well known, and data is easy to find. However, the relationship between dynamic and static stiffness have to be estimated. The manufacturers usually states a value around 1.7, however, our measurements shows that the factor may be considerable higher.

The factor which determines the degree of curvature on the load/deflection curve for rubberbased isolators is the shape factor, which is defined as the relationship between the loaded area and the free area. However, the dynamic to static stiffness-ratio also depends highly on the shape factor, the ratio is increasing when the shape factor is increasing. An example is shown below. For the calculated



Shape factor , cylinder

$s = D/4h$



curves it is seen that the dynamic to static stiffness ratio increases with increasing shape factor. In this example was used 60 shore-rubber. Measurements indicates that the dynamic to static stiffness ratio also increases with increasing shore value. However, further measurements should be done to confirm this, and to quantify the values.

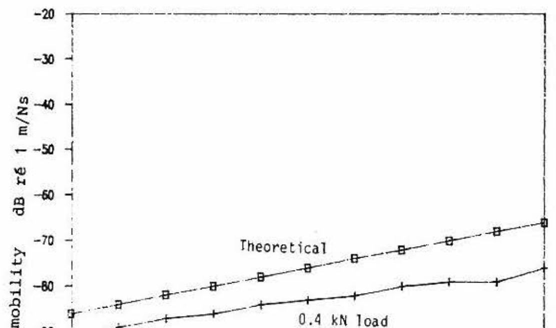
From the example is seen that the values for the damper with the lowest shape factor is most fluctuating in the high frequency region. The lowest standing wave resonance frequency in the rubber is:

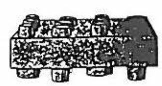
$$f_1 = (1/2\pi) \cdot 0,5 \sqrt{k_d/m} \qquad (1)$$

k_d is dynamic stiffness and m rubber mass

Our measurements confirm that the curves are relatively linear and stiffness controlled up to about this frequency, but for higher frequencies the curves fluctuates. Therefore we may calculate the mobility for frequencies lower than f_1, but for higher frequencies measurements should be done.

Another example which shows that is is important to be carefull in calculations of dynamic stiffness is the test of a 90 shore rubber mat which consists of many small knobs with different areas. Measurements of dynamics stiffness when the specimen was pressed to maximum static load gave 31 dB higher dynamic stiffness than calculated value from the data given from the manufacture.



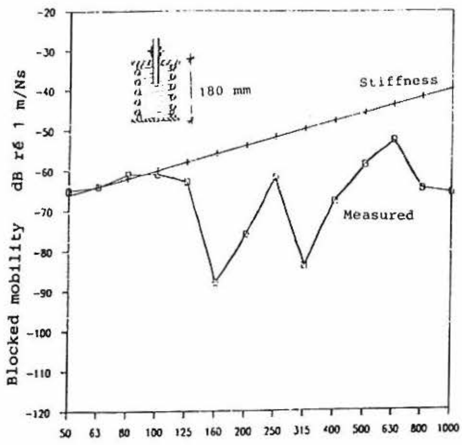


Rubloc rubber mat

maximum load 1 MPa, deflection 3mm

Test specimen area 125 x 150 mm^2

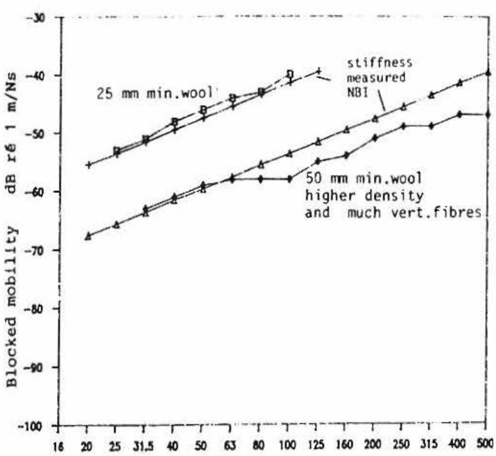
4. Measurements on other materials



Steelspring

The dynamic stiffness is equal to the static stiffness for steel springs. However, internal resonances are very dominant and may reduce the mobility even at low frequencies as shown in the example.

Mineralwool



Four cylindric mineralwool specimens that had been tested using the resonance method on the Norwegian Building Research Institute (NBI) was measured. The results are almost identical in the low frequency stiffness region. However, one of the specimen had higher degree of vertically fibres, and the measured mobility deviates at higher frequencies, probably because of wave transmission phenomena. For test of mineralwool for the purpose of floating floors such deviation from stiffness behavior may be important, and may explain deviation between measured and theoretical improvement.

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PREREQUISITES OF A COMPUTERIZED VIBRATIONAL PROGRAM BASED ON MULTIPOLE MODELS

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1. Introduction

In the study of structural vibration transmission, structure borne sound transmission etc., one primarily meets the problem to calculate the generalized forces and velocities at different points in the structure. Typical mechanical constructions are often divided into parts (substructures) interconnected in more or less "point"-like regions. If the structure is linear one can use methods from the linear multipole theory to calculate the forces and the velocities in the connection points.

The multipole theory provides a flexible way to store and to couple data needed for vibration studies in coupled systems. One important feature of multipole methods is that the input data, at least in principle, can be obtained experimentally as well as theoretically. The major drawback is the vast amount of input data. In practice it is often impossible to measure all data needed to calculate the vibration transmission in a real structure. Thus, one is often compelled to reduce the amount of input data in advance.

When a program package based on multipole methods is implemented on a computer, it is important to consider:

i. the storage capacity needed,
ii. the numerical facilities and algorithms needed and finally but not least important,
iii. the accuracy of the input data needed to produce calculated results with a given accurace.

The following sections are meant to shed some light on these matters.

2. Representing structures by multipoles

Generally, a linear mechanical structure can be divided into a source structure, a number of cascade coupled passive transmission structures and a passive receiver structure, see figure 1.

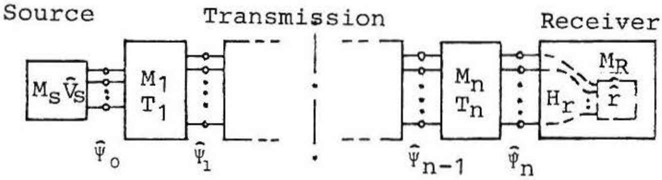

Figure 1 General cascade coupled linear structure

Referring to figure 1; $\hat{v}$, $\hat{F}$, $\hat{\Psi}$ and $\hat{r}$ are velocity, force, state ($\Psi=[F,v]^T$) and response vectors; M, T and H are mobility, transfer and force-response transfer matrices; indices are s-source, R-receiver, i-intersection i; $\wedge$ denotes fourier component.

Figure 1 also shows the quantities needed to calculate the state vector Ψ in each intersection and some physical response quantity r in the receiver.

With the following relations, all state vectors Ψ and physical quantities r can be calculated.

$$\hat{v}_o=\hat{v}_S-M_S\hat{F}_o \quad , \quad \hat{\Psi}_{i+1}=T_{i+1}\hat{\Psi}_i \quad , \quad i=0,1,\ldots n-1 \qquad (1,2)$$

$$\hat{v}_R=M_R\hat{F}_R \quad , \quad \hat{r}=H_r\hat{F}_R \qquad (3,4)$$

Thus, the elements of $\hat{v}_S$, M_S, $T_1 \ldots$, T_n, M_R and H_r defines the structure in terms of multipoles.

3. Required storage capacity of the computer

Before the implementation of a multipole program in a medium sized computer, it is important to consider the amount of storage needed. The major part of the storage capacity needed is required to store the input data, i.e. the matrices and vectors that defines the multipoles of the structure. To get an example of the required storage capacity, consider a structure with a source, a transmission and a receiver. Suppose there are n_e degrees of freedom coupled in the intersection between the source and the transmitting structure, n_R dof coupling the transmitting structure and the receiver and that the response vector has p components. Table 1 shows the number of independent input matrix elements and the required storage for each frequency line.

Note that apart from the input data, storage capacity must also be reserved for the program, the intermediate and the final results.

Substructure	Number of elements	Storage [bytes]
Source: $\hat{v}_S$	n_e	$8n_e$
M_S	$n_e(n_e+1)/2$	$4n_e(n_e+1)$
Transfer: M_T	$(n_e+n_R)(n_e+n_R+1)/2$	$4(n_e+n_R)(n_e+n_R+1)$
Receiver: M_R	$n_R(n_R+1)/2$	$4n_R(n_R+1)$
H_{Fr}	$n_R p$	$8n_R p$
Total: Σ	$n_e(n_e+n_R+2)+n_R(n_R+p+1)$	$8[n_e(n_e+n_R+2)+n_R(n_R+p+1)]$
Example: $n_e=30$ $n_R=30$, $p=10$	3090	24720

Table 1 Number of independent elements and required storage (6-7 significant digits) for a typical structure

4. Computational scheme and numerical facilities

The computations are organized in the following scheme.

i. Use equations (1)-(3) to formulate an equation system with an unknown force vector $\hat{F}_O$ at the source or $\hat{F}_R$ at the receiver. The choice of unknown force vector depends on where in the structure the result is wanted.

ii. Solve the system in a way that minimizes potential errors.

iii. Multiply the solution of ii. with appropriate sequencies of rearward or forward transfer matrices to compute the wanted results.

The scheme uses transmission structure data in the transfer matrix form. However, for two reasons the structural input data is preferably provided in mobility matrices. First, structural mobilities are normally simpler to obtain experimentally. Secondly, mobility matrices provide a very simple way to extend or reduce the number of dof of the multipole model. The model can be extended or reduced simply by adding or taking away appropriate rows and columns, without changing the other mobility matrix elements. This necessitates numerical routines that transform a mobility matrix to the corresponding rearward and/or forward transfer matrices. Rubin [1] has given useful interrelations between different matrix representations.

In general, finding the solution of the equation system requires rather sophisticated numerical routines. Firstly, the coefficient matrix of the system generally is rectangular. Secondly, it may well happen that the coefficient matrix is badly conditioned, i.e. small perturbations in the input data or round-off errors cause great fluctuations in the solution. Problems with badly conditioned systems are likely to appear when the structural wavelength is large enough compared with a distance between two coupling points in the same intersection. These problems demand some advanced numerical facilities such as for example "singular value decomposition".

In summary the considerations above necessitate the following numerical facilities.

i. Matrix multiplication, addition and subtraction.
ii. Right and left pseudoinverses of rectangular matrices (Solution of rectangular system and computation of transfer matrices).
iii. Routines that provide an estimate of the errors in the results.

5. Accuracy of the computed results

The computed results are always more or less perturbed by errors. If the mathematical model of the mechanical problem at hand is assumed to be correct, the errors essentially are due to inaccurate input data and round-off errors during the numerical computations. It is of great importance to obtain a measure of the accuracy of the results and to identify the dominant error source.

The propagation of errors from the sources to the result is characterized by the condition number. A low condition number implies that no catastrophic growth of relative errors occurs during the computations. In general, there are two independent causes for error growth, the numerical algorithms can be unstable (badly conditioned) or the mathematical model itself can be unstable. Unstable algorithms cause large round-off errors, while unstable models cause large errors due to inaccurate input data. An upper bound for the relative error can be computed with the formula $e_r = C_M(e_{in} + C_A u)$ where C_M and C_A are the condition numbers for the model and the algorithm. e_{in} is the relative error in the input data and u is the machine-unit, a measure of the round-off error in a single arithmetic operation.

If the relative accuracy of the result is poor because of a badly conditioned algorithm, i.e. the term $C_A u$ is dominant, there are two possible measures. First, try to reduce C_A by reformulating the algorithms. Secondly, reduce u by using higher-precision arithmetic. If, on the other hand, the model is poorly conditioned the usual measure is to reformulate the mathematical model.

Thus, the accuracy problem requires estimates of the condition numbers C_A and C_M. Given C_A and C_M, an upper bound for the relative error in the result can be computed. Given C_A, one can decide whether or not it is worthwhile to increase the accuracy of the input data. The problem is that in many cases it is very cumbersome to estimate C_A and C_M. Possible methods are forward error analysis, backward error analysis, interval analysis and systematic experimental perturbation analysis.

To conclude, it is of great importance to work out a fairly general procedure to estimate C_A and C_M.

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ON VIBRATIONAL POWER TRANSMISSION BETWEEN STRUCTURES
An application to road vehicle problems

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Introduction

It is often so that noise problems originate from structure-borne sound sources. Vibrational energy is propagated in structures, is transmitted from one structure to another, resulting in noise and vibration problems at locations remote from the excitation. The reduction of vibrational power transmission from a structure-borne sound source to a receiving structure is therefore an important stage in a vibration control procedure. The basic theory concerning vibration control can be found in e.g. [1]. The vibrational power transmitted is a function of the internal vibration of the source and the degree of coupling between the structures at their interface see e.g. [2]. On the basis of power transmission considerations, see e.g. [2], [3], [4], [5] a practical industrial problem has been investigated in cooperation with SAAB-SCANIA car division in Trollhättan (Sweden). The problem studied concerned the input of vibrational power from the contact wheel-road to the chassis of a road vehicle. Vibrational energy has to take a 'structural path' from excitation -engine or road- into the vehicle compartment.

Model

A road vehicle can be represented as shown in Figure 1. It contains 5 structure-borne sound sources, 4 wheel units and the engine, connected to a receiving structure, the chassis. The various transmission paths are also shown in the figure. The engine and the road provide two different types of vibration excitation. The engine delivers a tonal type excitation whereas the road excitation provides a noise type one or a transient one if there are obstacles on its surface. In order to consider only the excitation from the road, the engine is switched off and is considered as part of the chassis. The transmission path, engine to wheels, is also removed (drive axle). For the prediction of the power transmitted it is necessary to investigate the degree of coupling between the contact points [2]. With coupled points the total power transmitted is generally less than for uncoupled points but the prediction model is more complicated. The four wheels can be considered as independent sources (no structural coupling through the road). For the chassis, it has to be experimentally investigated.

Results

Transfer mobility measurements can be used to estimate the degree of coupling between points. It was of prime interest to find out if the points on the chassis corresponding to the connections with the various wheel units were coupled. Vibrational energy can be fed into the chassis through all components of motion. In a first step, the direction perpendicular to the structure surface was considered to be the preponderant direction of excitation. This choice was made since bending waves are easily excited. Nevertheless, with such a complicated structure as a car chassis, an in-plane excitation can, due to wave conversion, provide a strong bending wave excitation in other parts of the structure and therefore should be taken into account in a complete analysis. It was generally found that most of the points were strongly coupled only in a very low frequency domain, up to 40 Hz for the two front wheel units. At higher frequencies, some points may still be strongly coupled as shown in Figure 2. In the region 100-200 Hz the two points are strongly coupled in the y direction, similar transfer mobility and point mobility in magnitude. It shows that an input of energy at these points from the road excitation, can result in a high response in other parts of the system. Apart from this example the vibrational energy transfer between the two front wheel is very low above 40 Hz. This implies that they can be considered as independent sources. Thus, one can study the transmission of power through the multi-point interface associated to one wheel unit at a time. The main transmission paths in the wheel unit are the shock absorber-spring system, the wishbone and the path associated to the steering system. An example of chassis input mobility is given in Figure 3. Below 40 Hz, some of the complete chassis modes are excited even if they are rather damped at the present measuring point. The real part of the input mobility is rather high resulting in a possible high transmission of power. Above this frequency range, the bending wave length becomes less than the typical dimensions of important parts of the chassis, resulting in the possible excitation of sub-structure modes. Further up in frequency the bending wavelength becomes very small and the mobility is very locally determined. In the range 30 to 100 Hz the mobility is stiffeness governed and no power is transmitted. Also it was found previously that most of the points were coupled below 40 Hz. This is in agreement with the fact that few modes of the whole chassis are excited.

Conclusions

The data obtained provides valuable information for both qualitative and quantitative analysis of vibrations in car structures. The ratio, wave length to typical dimension of the structure under study, is shown to be one of the most important parameter for vibration analysis. The few measurements presented here show that the mobility theory can be advantageously used in the prediction, of the power transmission between structures and of the identification of transmission paths. The range of applicability of this approach is in no way limited to the car industry.

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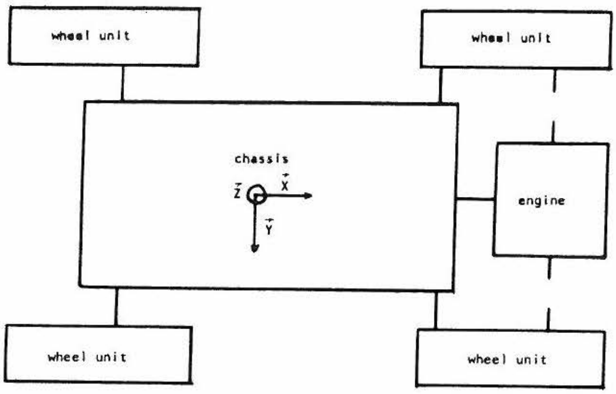

Figure 1. Model for a road vehicle

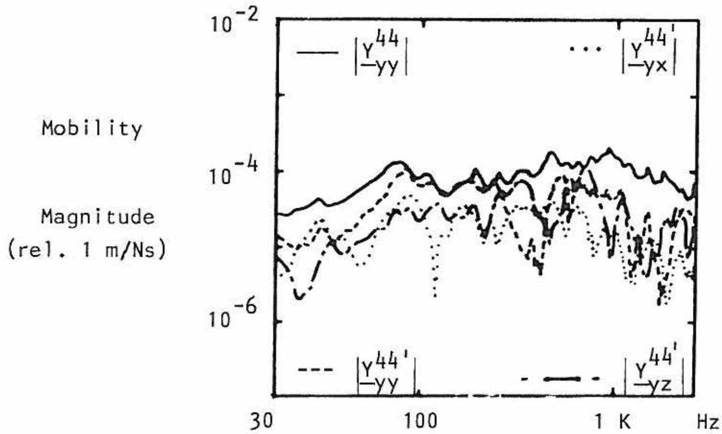

Figure 2. Coupling between points

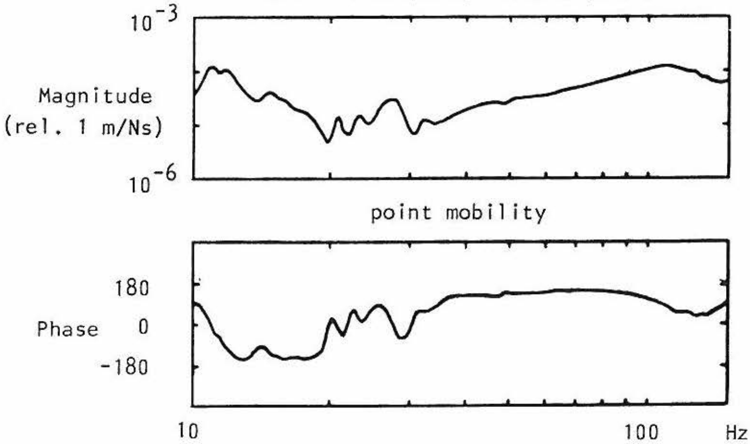

Figure 3. Example of chassis point mobility

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STRIP MOBILITY

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Introduction

In many engineering applications two subsystems, e.g. source and receiver are coupled over strip-like areas or in a number of discrete points. In order to predict the sound and vibration transmission concerning point coupled subsystems the ordinary mobility theory offers a straight-forward way. It would be most valuable to extend this theory to comprise also coupling over strip-like areas. Hereby, it would be possible to optimize an installation with respect to the power supplied to the receiving structure. In this paper an analysis of some features of strip-coupling is presented. The strip is defined as a long and narrow surface which can be treated one-dimensionally.

Analysis

The strip mobility, Y_{Ω}, is based upon the power concept, [1]. This is due to the fact that power often is the quantity sought in order to handle problems concerning structure-borne sound and vibration transmission.

It must be emphasized that such a definition of the strip-mobility implies a quantity which is not entirely governed by the structural characteristics, but is also influenced by the spatial distribution of the field variables at the contact area. Hereby, to get a survey of the problem, it must be split up into a number of categories [1], see Table 1.

In this paper we will concentrate on the firm contact condition and the two idealized cases i.e., when $\lambda_{strip} > \lambda_{object}$ and $\lambda_{strip} < \lambda_{object}$, henceforth denoted, case 1 and case 2 respectively.

Table 1. Categories of strip-coupling.

Contact condition \ Stiffness relation	Stiff strip ($\lambda_{strip} > \lambda_{obj}$)	Intermediate ($\lambda_{strip} \approx \lambda_{obj}$)	Soft strip ($\lambda_{strip} < \lambda_{obj}$)
Gravity	Random point excitation along the strip	Surface excitation with force distribution governed by the common wavenumber	Random point excitation along the strip
Discontinuous firm	Multi-point excitation with uniform displacement prescribed at all points	-"-	Multi-point excitation with uniform force distribution prescribed at all points
Firm	Surface excitation with displacement governed by the wavenumber of the strip	-"-	Surface excitation with force distribution governed by the wavenumber of the receiver

Case 1 implies that the object is excited by a uniform, conphase velocity distribution along the finite strip and the latter one, case 2, that the object is excited by a uniform, conphase pressure distribution. Figure 1 illustrates a continuous arbitrary pressure distribution $\sigma(y_0)$ along a finite strip, exciting an infinite, plate-like structure.

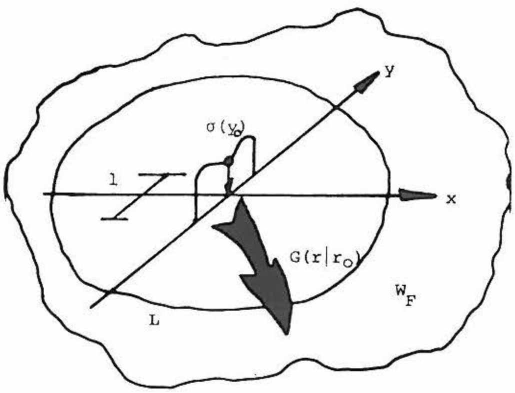

Figure 1. An infinite plate-like structure excited by a pressure distribution $\sigma(y_0)$.

From $\sigma(y)$ and $G(\vec{r}|\vec{r}_0)$ the power flow, W_F, into the farfield can be determined formally from

$$W_F = \text{constant} \cdot \int_L \left| \int_l G(\vec{r}|\vec{r}_0)\, \sigma(y_0) dy_0 \right|^2 dL$$

W_F can be interpreted as the real part of the comlex power input which, in turn, can be shown to be proportional to the real part of the strip mobility, $Re[Y_Q]$. With respect to case 2 the procedure described above is fairly straightforward whereas, in case 1, the mixed boundary value problem is circumvented by approximating the actual pressure distribution, due to a uniform velocity along the excitation line, with that of a rigid strip-indenter, found from a static analysis. This distribution is valid strictly merely up to $k_B l \leq \pi$.

The approximate, theoretical results for the real part of the strip mobility, $Re[Y_Q]$, are presented in Figure 2. Moreover, $Re[Y_Q]$ is normalized with respect to $Re[Y_p] = 1/(8\sqrt{MB})$. Hereby, some interesting observations can be made.

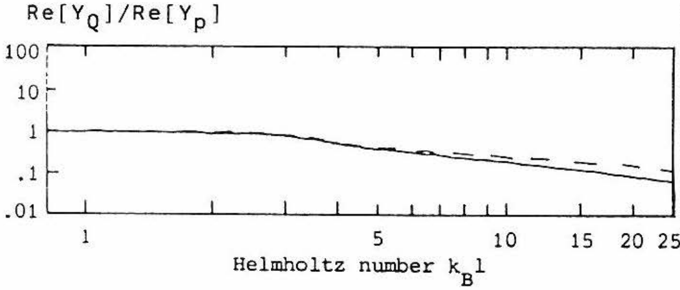

Figure 2. Normalized strip mobility versus the governing wave number, k_B, and the length, l, of the strip (---) case 1 and (——) case 2.

The results in Figure 2 indicates a quite similar behaviour in the two excitation cases. It is also seen that for $k_B l \leq 2.5$, the strip mobility is almost equal to the ordinary point mobility. Moreover, as kl increases one obtains a decrease of the nomalized strip mobility $Re[Y_Q] / Re[Y_p]$.

Experiment

In Figure 3 the arrangements used in the experiments in the farfield power flow is shown for cases 1 and 2 respectively.

The excited object is a 1 mm thick plate of aluminium, 1100 x 2000 mm^2, embedded in sand at the edges. The indenter used is a 10 mm thick, triangular disc with a length of 100 mm and height 90 mm. In case 1 the indenter was fastened to the aluminium plate by means of a cyano-acrylate type glue reinforced with aluminium powder. In case 2 a soft rubber strip was placed between the indenter and the aluminium plate. The measurement results of $Re[Y_Q]$ is shown in Figure 4. $Re[Y_p]$, i.e. the real part of the ordinary point mobility, was measured in order to normalize $Re[Y_Q]$ on $Re[Y_p]$.

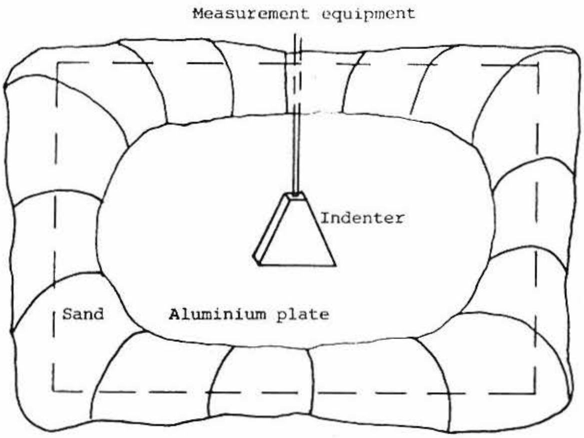

Figure 3. The arrangement in order to measure the farfield power flow.

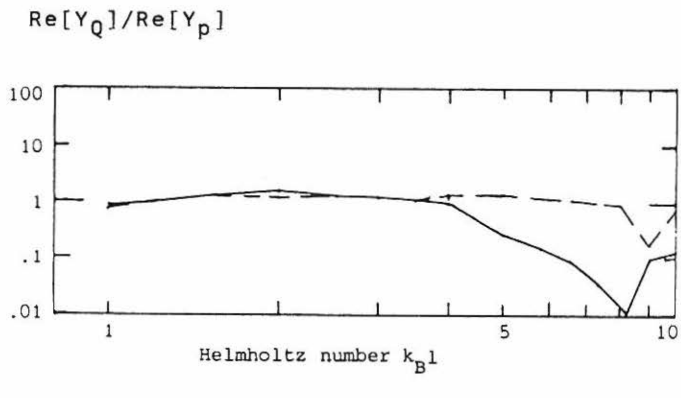

Figure 4. Measurement results showning the real part of the strip mobility with respect to Helmholtz number $k_B l$. (---) case 1 and (——) case 2

Concluding remarks

The results for case 2 indicates that the strip mobility becomes more reactive than the ordinary point mobility for high frequencies. This means that a reduction in the power transmission can be gained by using strip coupling. From the work with respect to case 1 it may be concluded that the power transmission will not exceed the case of point coupling. However, the pressure distribution applied in case 1 seems to be too a coarse simplification to gain sufficient insight into the corresponding transmission process, and a refined distribution is presently investigated.

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EXPERIMENTAL DETERMINATION OF STRUCTURAL DAMPING

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1. INTRODUCTION

In the prediction of the vibrational response of mechanical systems, knowledge of the damping characteristics of the structural components is essential. In general these properties must be determined experimentally.

Owing to the diversity of structural components encountered, a variety of experimental techniques for determining structural damping have been developed; see e.g. Section 12.1 in reference [1]. The purpose of this paper is to discuss the advantages and disadvantages of some of these experimental procedures; a new method is also presented.

2. MEASUREMENT OF DAMPING

The power input method

The 'power input method' is, in effect, based on inversion of the equation of power balance governing a structure subjected to vibrational excitation. Thus the loss factor is determined from the relation $\eta = P/\omega E$, where P is the power input and E is the total vibrational energy of the structure [2,3]. The input power flow can be estimated from the time-averaged product of force and velocity at the driving point, and the energy is determined as twice the kinetic energy, which is estimated by integrating the product of density and squared velocity over the structure. This method has the advantage of being closely associated with the definition of structural losses [1]. An additional advantage, at least from a theoretical point of view, is that it is possible, in principle, to determine material damping even in a frequency band without any structural resonances [4].

Conceptually satisfactory though it is, the power input method has several disadvantages which severely limit its applicability: (i) the mass distribution of the structure under test must be uniform; and (ii) the method depends critically on measurements of input power: on lightly damped structures such measurements are not very accurate; see reference [4].

Modal identification techniques

Estimation of damping by means of curve-fitting procedures (either in the frequency or in the time domain), as used, for example, in experimental modal analysis, gives excellent results if the modes of the structure under investigation are well separated. Refined curve-fitting procedures can also cope with moderately coupled modes, but the methods are ill-suited to structures with a considerable modal overlap.

The traditional decay method

In the standard reverberation decay technique, the loss factor is deduced from the slope of the logarithmic average energy decay curve. Some of the limitations of this technique have recently been examined [5]; the results are summarized in the following.

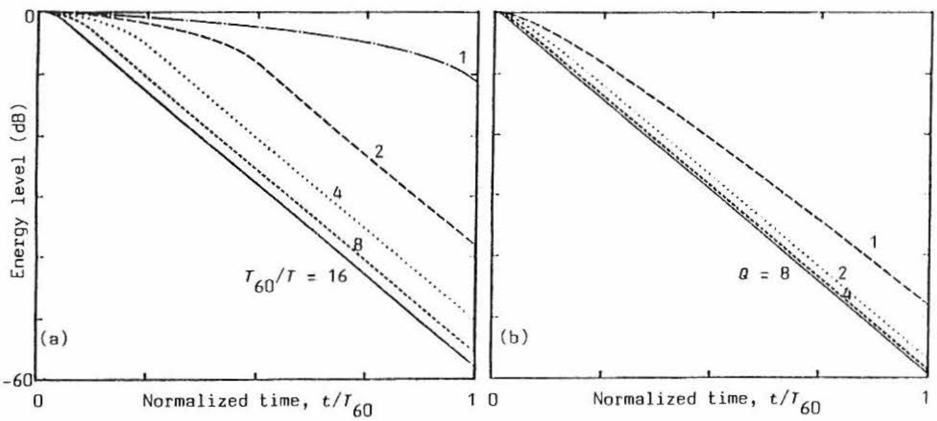

Figure 1 *Distortion of decays produced by (a) linear and (b) exponential averaging.*

To reduce the considerable statistical fluctuations of random noise decay curves, the squared, filtered vibrational signal is normally smoothed by some averaging device. The influence of this smoothing is demonstrated in figure 1. It is apparent that acceptable decay curves are obtained only if $T_{60}/T > 4$ or if $Q > 2$, where T_{60} is the reverberation time of the structure under test, T is the averaging time of the 'linear' averaging device, and Q is the ratio between T_{60} and the reverberation time which is observed if a signal applied to the 'exponential' device is suddenly interrupted. These requirements lead to the inequalities

$$\eta < 1/2f_c T \quad \text{and} \quad \eta < 1/f_c\, T_{60}(\text{device}) ,$$

where f_c is the centre frequency of the bandpass filter.

Another limitation is due to 'ringing' of the filter: figure 2 illustrates this effect. It can be seen that, with fairly selective filters

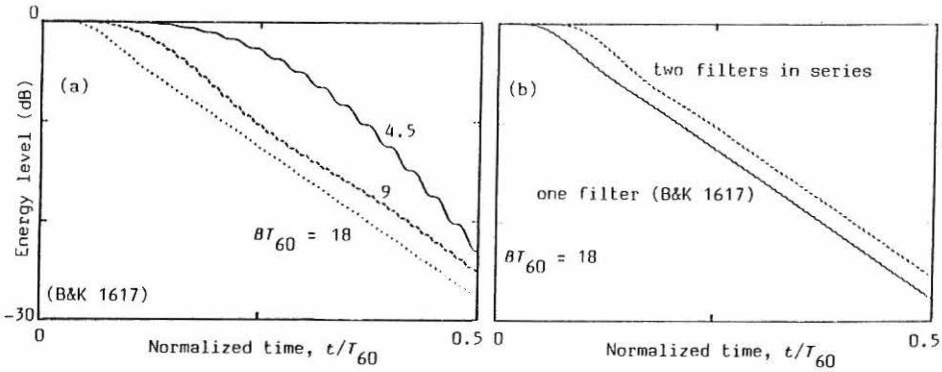

Figure 2 *Influence on decays of (a) bandwidth and (b) selectivity of bandpass filter.*

as, for example, the third-octave band filters in B & K 1617, the inequality $BT_{60} > 16$ should be observed (B is the 3-dB bandwidth of the filter). For third-octave bands this corresponds to the requirement $\eta < 0.03$.

Still another limitation is due to the interaction between the structure under test and the shaker used in exciting the structure: clearly, damping due to the shaker should be negligible. Otherwise expressed, in the steady state, the force signal applied to the structure should be independent of the fluctuations with frequency of the driving-point mobility. This requirement cannot always be met, in particular if the structure is very light or very lightly damped.

Other decay techniques

Decay curves, and thus in turn loss factors, can also be determined from impulse response functions [6]. An interesting method of estimating such functions is based on excitation with periodic, pseudo-random 'maximum length sequences' [7]; the impulse response can be determined by means of a very efficient algorithm known as 'the fast Hadamard transform' [8]. However, this method relies on the porportionality between the impulse response and the cross-correlation function between excitation signal and response, that is, the force signal is also here assumed to be independent of the test object.

Yet another decay method is described in reference [9]. In this method a frequency response is measured with a digital dual channel frequency analyzer; in order to determine the losses in frequency bands this response function is multiplied by a series of weighting functions; in turn an impulse response is computed from each resulting frequency response; decays are computed from the impulse response functions; and, finally, frequency band loss factors are deduced from these decays.

One of the limitations of this method is set by the size of the FFT transform of the frequency analyzer. If excitation with pseudo-random noise is used, the impulse response is, in effect, merely truncated. The resulting decays are shown in figure 3(a), from which it can be concluded that the errors are tolerable if $T_{60}/T < 2.5$, where T is the record length of the analyzer.

Another restriction is due to the bandpass 'filters', i.e. weighting functions; see figure 3(b). With a 50-percent cosine-tapered window, the distortion will be moderate if $BT_{60} > 16$.

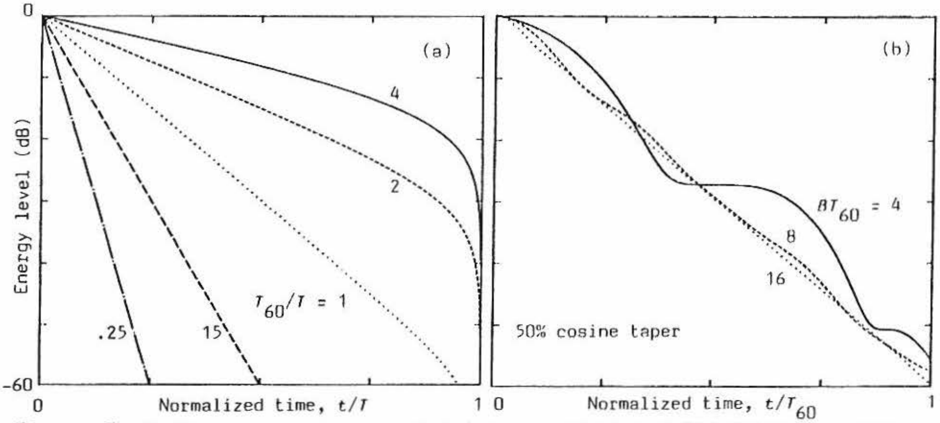

Figure 3 *Influence on decays of (a) truncating and (b) filtering the impulse response.*

If the spectral analyzer has an FFT block size of 2048 corresponding to about 800 usable spectral lines, the two requirements can, for third-octave bands, be combined in the inequalities $0.03 > \eta > 0.0003$. Using another frequency band has the effect of shifting this range.

3. CONCLUSIONS

The considerations of the previous section can be summarized as follows: The power input method is theoretically satisfactory, but the practical use of the method is limited to uniform structures and the method is unfit for examining very light damping. Methods based on curve-fitting require a low or moderate modal overlap. The restrictions imposed by the traditional decay method are largely due to the detector and filter, that is, they are, in principle, controllable, but the influence of the exciter may impede examining very light, and very lightly damped, structures; this applies also to the decay method described in reference [7]. The limitations of the decay method described in reference [9] are, to some extent, under the control of the experimenter.

ACKNOWLEDGEMENTS

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VÄTSKEDROPPEN SOM STRUKTURAKUSTISK KÄLLA

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I många tekniska sammanhang spelar idag vätskedroppar en stor roll. Som exempel kan nämnas olika rengöringsmetoder såsom diskning och högtrycksspolning samt vid slipning. Vid sidan om den önskade mekaniska effekten uppkommer också ofta en oönskad ljudalstring.

Mycket omfattande studier har utförts rörande de mekaniska effekter som uppträder då en vätskedroppe träffar en hård yta, speciellt vid höga hastigheter. Däremot har de akustiska konsekvenserna inte uppmärksammats i samma grad.

Detta arbete visar resultat från en experimentell studie av vätskedroppar som träffar en homogen struktur vid ganska låga hastigheter. Dessutom jämförs de experimentella resultaten med en enkel predikteringsmodell.

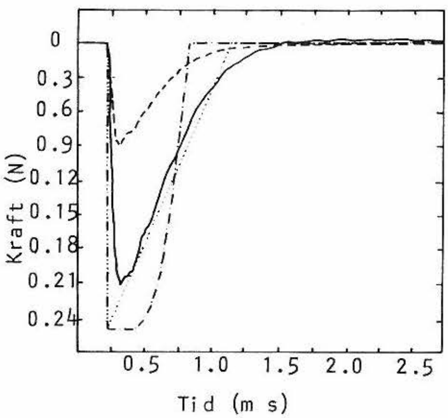
En droppe som faller fritt i en gas antar en geometri som i inledningsskedet är sfärisk. Därefter plattas undersidan till något medan översidan, på grund av undertrycket ovanför droppen, dras ut [1]. Detta betyder att då droppen, efter att ha fallit en tid, träffar en yta så är formen inte längre sfärisk utan kan närmast karakteriseras såsom ellipsoidal.

Kontakten mellan en vätskedroppe och en yta kan lämpligen indelas i två faser, stöt- och flödesfas. I kontaktögonblicket sprängs inte droppen omedelbart utan små jetstrålar slungas utåt sidorna strax ovanför kontaktpunkten. De flesta studier har koncentrerats till stötfasen eftersom avsikten har varit att klarlägga de mekaniska effekterna av stötkraften från en droppe mot en hård yta, varvid jetstrålarna tycks ha en avgörande betydelse [2]. Dessutom kan jetstrålningen vara mer signifikant då de gäller höga hastigheter, dvs för hastigheter högre än de slutliga hastigheter som uppnås vid fritt fall. Akustiskt sett kan dock stötfasen väntas ha mindre betydelse varför denna undersökning mer ägnats den andra fasen - flödesfasen.

Experiment

Vätskedropparna släpptes från en pipett på varierande höjder ovanför en kraftgivare. Kraftgivaren var monterad på en kraftfördelningskon som var limmad på en lågmobil betongplatta i laboratoriet. På kraftgivaren var en tunn titanskiva fastskruvad.

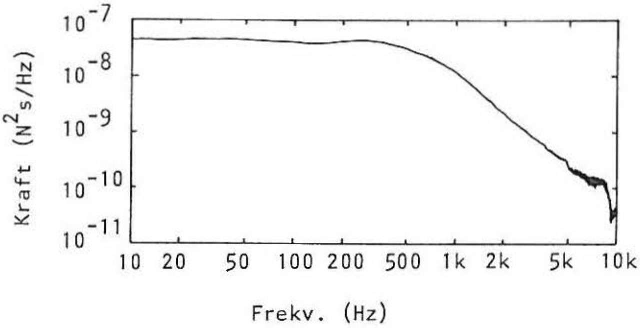
Repeterbarheten i kraftpulsen befanns vara utomordentligt god med praktiskt taget ingen skillnad mellan dropparna. Som nästa åtgärd undersöktes lineariteten i impulserna. Höjden varierades och integralen av kraftpulsen bestämdes. Inom området för dropphastigheter mellan 2 - 7 m/s visade sig impulsen vara linjär. Vid de ovan nämnda experimenten användes vattendroppar och för att undersöka en eventuell inverkan av ytspänningen hos vätskan utfördes också experiment med etanol. Ytspänningen hos etanol är ungefär 1/3 av den hos vatten.



Figur 1. Jämförelse mellan mätt kraftpuls från fritt fallande droppe av (——) vatten och (---) etanol från 1 m höjd. Predikterade resultat för (–·–) cylindrisk-halvsfärisk och (···) paraboloidal droppform.

I figur 1 finns kraftpulserna för de båda vätskorna exemplifierade. Pulsformerna är mycket lika och med en skalning av massorna, etanol till vatten, enligt m_e/m_w = 1/3 uppnås en mycket god överensstämmelse.

I figur 2 visas kraftspektrum för en vattendroppe från 2 m höjd.



Figur 2. Kraftspektrum för en vattendroppe från 2 m höjd.

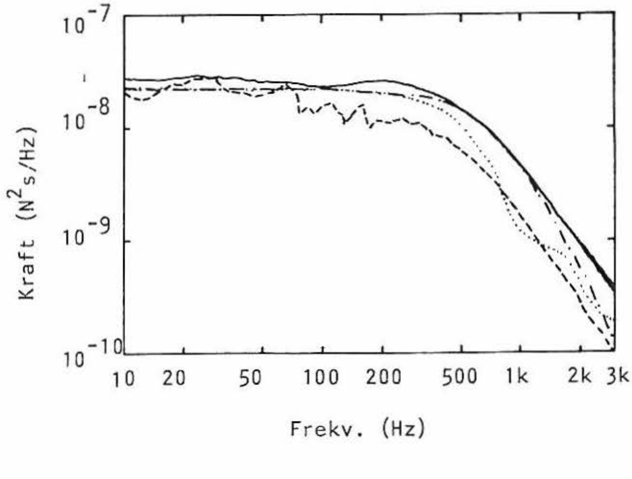
Av figuren framgår klart att den utvecklade kraften på en stel struktur är lågfrekvensdominerad. Brytfrekvensen kan tecknas geometriskt såsom

$$f_b = v_0/2R \tag{1}$$

där v_0 är droppens hastighet i kontaktögonblicket och R radien av den initiellt, sfäriska droppen. För en flexibel mottagarstruktur blir pulsen förlängd. Detta innebär att brytfrekvensen blir lägre jämfört med den för en stel struktur. För att kunna avgöra om den mottagande strukturen är att betrakta som stel eller flexibel måste punktimpedansen för strukturen jämföras med flödesimpedansen hos droppen och inte, som intuitivt skulle kunna tänkas, med dess massimpedans. Med hjälp av en lineariserad modell för flödesfasen fås flödesimpedansen som

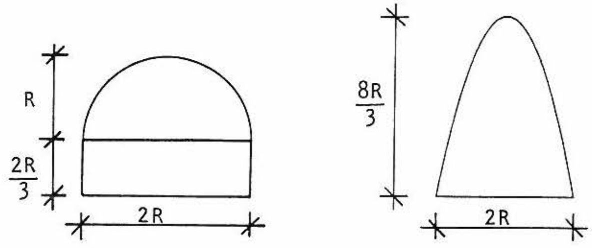
$$Z_{flow} = \rho\, S \cdot v \tag{2}$$

där ρ är vätskans densitet, S tvärsnittsaren hos droppen och v droppens hastighet. Inverkan av flexibilitet hos mottagarstruktur presenteras i figur 3.



Figur 3. Kraftspektra av vattendroppar på (——) 400 mm betongplatta och (---) 2 mm plexiglas. Predikterande resultat för (–·–) cylindrisk-sfärisk och (···) paraboloidal droppform.

En mycket enkel modell [3] kan baseras på förutsättningen att droppens hastighet i kontaktögonblicket bibehålles konstant under hela flödesfasen. I detta fall bestämmes pulsformen av droppens utseende. Två enkla geometriska former på droppen har undersökts, se figur 4.



Figur 4. Två förenklade droppformer.

I figur 1 visas som jämförelse de beräknade kraftpulserna för de cylindrisk-halvsfäriska och paraboloidala dropparna. Motsvarande beräknade kraftspektra visas i figur 3 och är där jämföra med ett spektrum mätt på en styv mottagarstruktur.

Det syns här att för båda droppformerna överskattas pulsens maximum och att, i det cylindrisk-halvfäriska fallet pulstiden är för kort medan pulstiden i fallet med paraboloiden är acceptabel.

Sammanfattning

För låga hastigheter kan konstateras att den resulterande impulsen från en droppe på en styv struktur är praktiskt taget linjär med hastigheten. Kraftpulsen beror på droppens form och hastighet under kontakten med strukturen.

Styvheten hos strukturen kan i sin tur bedömas genom jämförelse mellan strukturens punktimpedans och flödesimpedansen hos droppen.

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ESTIMATION OF SOUND POWER RADIATED FROM A PLATE WITH EDGE EXCITATION

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1. Introduction

In reference [1] a well-known formula for estimating the sound power emitted from a finite, baffled and point-force excited plate, well below the critical frequency, is derived [eq (9) p 498]. It is easy to realize that the steps used in this derivation can be made the basis of a general estimation procedure. A procedure which, at least in theory, can be used to estimate the sound power emitted from finite, baffled thin plates for any type of excitation. The aim of this paper is to present such an estimation procedure and illustrate its application to a special case. The general ideas involved in the procedure given below are in no way novel, however, to the authors knowledge the here presented formulation is new.

2. Estimation procedure

The basic assumption behind this procedure is that we are in the multi-modal (MM) region of the plate.

The estimation procedure applies to a plate where bending waves are generated by prescribed point-like or extended excitations on the plate or at its edges.

In the MM-region the input impedance for some specified type of excitation of the plate can, according to Skudrzyk [2], be taken as the input impedance for the corresponding infinite system.

The steps of the estimation procedure will be demonstrated below through an application example, which is of engineering interest for the design of a closely fitting radiation shield. The type of radiation shield in question, consists of a plate tightly mounted to a flat machine surface where the mounting is through a "rigid" connecting strip around the edges of the plate (Figure 1).

To be efficient a radiation shield should be mounted on a machine surface that has a radiation efficiency equal to 1 in the frequency region of interest. The main principle behind the radiation shield is that it consists of a plate with a radiation efficiency much smaller than 1 in the same frequency region.

In reference [3] it is argued that the following simplified model could be a possible description of the excitation in the case depicted in Figure 1: *The excitation consists of a number of uncorrelated line-excitations, each with a length in the order of the structural wavelength on the machine surface which acts around the edges of the plate.*

The estimation procedure can now be divided into three steps

Step 1: The mechanical power input to the plate is obtained from the power input to the corresponding infinite system, i.e., for the case in question a semi-infinite plate with a line excitation at its edge.

For this case the power input to the plate becomes [3]

$$W_{in} = P\bar{v}_e^2 \, \mathrm{Re}[Z_e'] \qquad (1)$$

where $\bar{v}_e^2$ is the spatial average of the squared RMS-velocity at the edges of the plate, P is the plate perimeter and Z_e' is the input impedance/unit length. From reference [1] the following result is obtained for Z_e'

$$Z_e' = \kappa m c_b (1+j)/2 \qquad (2)$$

where κ=1 for "simply supported" edges and 2 for "clamped" edges, c_b is the phase velocity of bending waves in the plate, m is the plate mass/unit area and j is the imaginary unit.

Step 2: The mechanical power fed into the plate will generate a bending wave field. In the MM-region it is appropriate to divide this field into two parts.

The first part can be taken as the vibration field which would be obtained if the excitation acted on the corresponding infinite system. The second part corresponds to the modes excited on the plate, this part we refer to as the reverberant field. A simple relation exists between the power input to the plate and the vibration level in the reverberant field [1]

$$W_{in} = \eta\omega m S\langle v^2\rangle \qquad (3)$$

where η is the plate loss factor, ω is the angular frequency, S is the surface area and $\langle v^2\rangle$ denotes the spatial mean of the squared RMS-value of the reverberant normal surface velocity on the plate.

Using eqs (3) and (2) we obtain

$$\langle v^2 \rangle = \frac{P\bar{v}_e^2 \, \mathrm{Re}[Z_e']}{\eta \omega m S} \tag{4}$$

Step 3: The sound power emitted from the plate is assumed to be the direct sum of two parts, i.e., interference effects are neglected. The first part is that emitted from the excitation regions (W_∞). The second part is that emitted from the reverberant field (W_{rev}). The first part should be estimated as the sound power emitted from the corresponding infinite case. The reverberant part is estimated from the equation

$$W_{rev} = \sigma \rho c S \langle v^2 \rangle \tag{5}$$

where σ is the radiation efficiency, ρ and c are the density and speed of sound in the surrounding medium. The radiation efficiency can be estimated from results given in reference [4].

Above the critical frequency (f_c) only the sound power emitted from the reverberant field needs normally to be taken into account. Below the critical frequency both contributions to the emitted sound power are normally of importance. To estimate W_∞ in that frequency range we regard the corresponding infinite case, which is a baffled semi-infinite plate excited at its edge along a strip with the width w. From reference [3] we obtain the following expression for W_∞

$$W_\infty = r \rho c w' P \bar{v}_e^2 \tag{6}$$

where r is the radiation efficiency for an infinite strip with the width w' moving as a rigid piston [5], $w' = w + \kappa/\sqrt{2}k_b$ and k_b is the bending wave number on the plate.

3. Comparison with experiments

Measurements have been made for a number of different radiation shields, all with the dimensions 0.7x0.9 m^2. More details about the experimental procedure can be found in reference [3].

Figure 2 shows a comparison between the measured sound power levels and the theoretical sound power levels calculated from the above described procedure for two typical cases. It can be seen from figure 2a that there is a very good agreement between measured and theoretical curves for this case, a plexiglass plate with $f_c \approx 16$ kHz. Here the emitted sound power is strongly dominated by the part coming from the excitation region (W_∞).

For the 1 mm thick steel plate shown in Figure 2b, the reverberant part of the sound power is also of importance below the critical frequency ($f_c \approx 12$ kHz). The agreement between the measured and the theoretical curves is not as good as in the first case. However, it is still more than acceptable for engineering purposes.

4. Conclusions

A estimation procedure for the sound power emitted from plates with any type of excitation has been proposed in this paper.Measurements have been made on a number of plates to verify the estimation procedure. The comparison between theory and experiment indicates that the estimation procedure works well, the exception being cases where W_{rev} dominates the sound radiation. Then theory and measurement can differ with up to 5 dB in certain frequency bands. However, despite this the proposed method can probably be a very useful tool in engineering applications.

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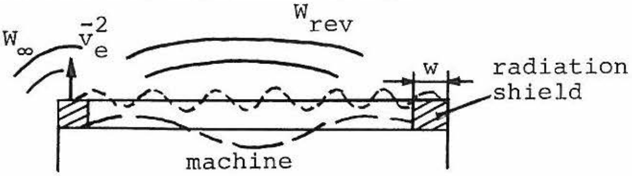
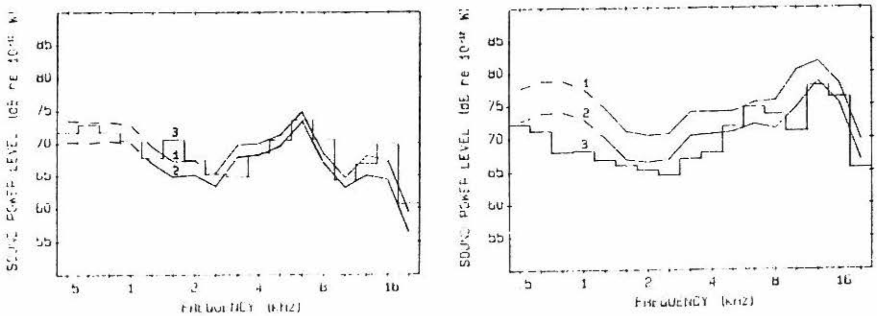
Figures

1. Configuration for radiation shield.



2. Emitted sound power level from plates with line exitation around the edges, 1 theoretical (clamped boundaries), 2 theoretical (simply supported boundaries), 3 measured.



a) 2 mm thick plexiglass plate ($\eta \approx 5 \cdot 10^{-2}$).

b) 1 mm thick steel plate ($\eta \approx 4 \cdot 10^{-3}$).

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AIR-BORNE SOUND EXCITATION OF A HOMOGENEOUS PLATE

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Introduction

In the field of building acoustics, the coupling between incident air-borne sound and the response of a plate is generally described by a bending wave equation of the Bernoulli-Euler type. The range of applicability of this equation is often expressed as a frequency region, where the upper limit is given by the condition that the wavelength of the free bending wave equals six times the thickness of the plate.

However, this limit is applicable to the case of free vibrations only. If the plate is excited by air-borne sound the state of things is more complicated. In this case, the range of applicability cannot be defined as a frequency region only; a wave number region must also be specified.

This arguing leads to an interesting question. If a wave number region is specified for the bending wave equation, what is then to be found beyond the upper limit of this region?

The plate equations for low frequencies

The plate is here assumed to be of uniform thickness and infinite extent. It is excited on one side by a plane wave, which is propagating in the x-direction,

$$p = \hat{p}\exp(-jk_x x + j\omega t). \qquad (1)$$

There are two types of solutions for the displacement: one is symmetric (S) with respect to the neutral plane of the plate and the other is antisymmetric (A). In the case of a "thin" plate, the symmetric solution describes the coupling to the quasi-longitudinal wave and the antisymmetric solution the coupling to the bending wave. These solutions can formally be written as

$$w = (\hat{w}_A + \hat{w}_S)\exp(-jk_x x + j\omega t), \tag{2}$$

$$\hat{w}_A = \hat{p}\,[12(1-\nu^2)]\,[E\delta^3(k_x^4-k_B^4)]^{-1} = \hat{w}_B, \tag{3}$$

$$\hat{w}_S = \pm\hat{p}\,[\delta(1-\nu^2)(k_x^2-h^2)][4E(k_x^2-k_L^2)]^{-1}, \tag{4}$$

where the plus sign in eq (4) is valid for the excited side of the plate and the minus sign for the free side and where k_x = wave number of forcing pressure, ω = radian frequency, ν = Poisson's ratio, E = Young's modulus, δ = thickness of plate, k_B = wave number of the free bending wave, h = wave number of the pure longitudinal wave, k_L = wave number of the free quasi-longitudinal wave.

These solutions are valid only if the plate is thin compared with wavelength of the forcing pressure. If this is not the case, i.e., the wave number k_x is comparatively large, another pair of solutions must be used [1],

$$\hat{w}_A = \hat{p}\,[(1-\nu^2)/E][2\cosh^2(k_x\gamma)/(k_x(\sinh(2k_x\gamma)-2k_x\gamma))], \tag{5}$$

$$\hat{w}_S = \pm\hat{p}\,[(1-\nu^2)/E][2\sinh^2(k_x\gamma)/(k_x(\sinh(2k_x\gamma)+2k_x\gamma))], \tag{6}$$

where $\gamma = \delta/2$.

The new plate equation, described by equations (5) and (6), is valid for arbitrarily large values of k_x, provided that $k_x \gg k_B$ and $k_x \gg k_L$. For small values of $k_x\gamma$, equation (5) turns into

$$\hat{w}_A = \hat{p}[3(1-\nu^2)][2E\gamma^3 k_x^4]^{-1}. \tag{7}$$

This result is also obtained from the bending wave equation provided that $k_x \gg k_B$. In the same way, the two symmetric solutions turn into

$$\hat{w}_S = \pm\hat{p}\delta(1-\nu^2)/4E. \tag{8}$$

It can be shown that equation (7) deviates with less than 1 dB from equation (5) if $k_x\delta < 0.77$. In the symmetric case, the corresponding condition is $k_x\delta < 3.5$.

For large values of $k_x\gamma$, the new plate equation describes a case of forced excitation of Rayleigh waves.

Influence of the quasi-longitudinal wave

It can be seen from equations (3) and (4) that the coupling to the quasi-longitudinal wave is most important when the plate is excited with a single wave number k_L. As the propagation speed of sound in air is lower than that of the quasi-longitudinal wave in most materials used in plates, it is clear that excitation with a wave number k_L can be realized by a plane wave in air that is incident with a certain angle Θ,

$$\Theta = \sin^{-1}(k_L/k), \tag{9}$$

k = wave number of incident sound, Θ = angle from the normal of the plate. If this coincidence condition is fulfilled, the out-of-plane displacement may be of the same order of magnitude as the displacement of the bending wave [1].

Thus, the influence of the quasi-longitudinal wave on the sound transmitted through the plate may be observed but cannot be expected to be very dramatic. However, a quasi-longitudinal wave has a comparatively large amplitude in the plane of the plate and also a comparatively large characteristic impedance. It is then evident that the quasi-longitudinal wave can be very important for the energy flow to a connected plate with this type of excitation.

Excitation with large wave numbers

Of particular interest is here the limit of $k_x\delta = 0.77$ for the use of the bending wave equation. In the case of a 150 mm thick wall of concrete, excited by sound in air with grazing incidence, this limit corresponds to a frequency of only 280 Hz.

The bending wave equation does not predict the displacement of the excited side very well at frequencies above the limit given by $k_x = 0.77$. For instance, with the same wall but at a frequency of 1180 Hz (this frequency corresponds to $\lambda_B = 6\delta$), the bending wave equation gives an underestimate of the amplitude with 16 dB. However, the bending wave equation gives a better result for the free side of the plate [2].

The new plate equations have the character of a quasi-static solution. It can, nevertheless, be used for fairly high frequencies in this case, at least up to 17 kHz, see Figure 1.

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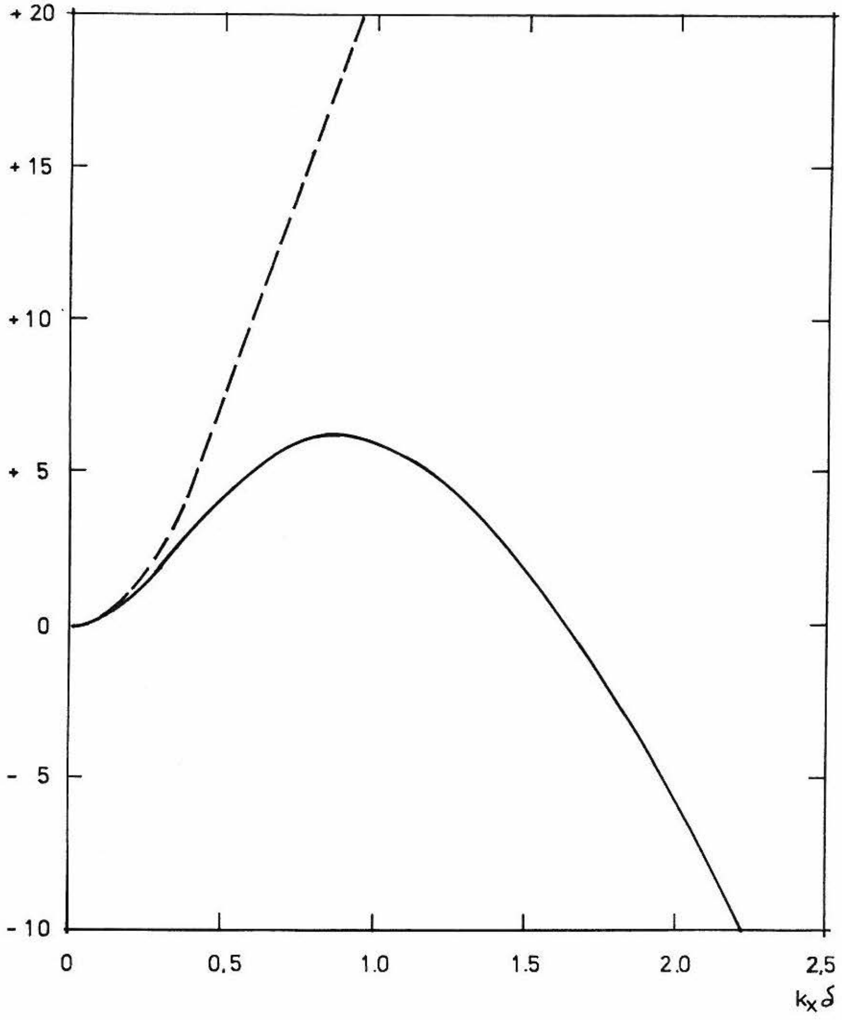

Figure 1. 20 lg $[(\hat{w}_A+\hat{w}_S)/\hat{w}_B]$ with $\hat{w}_A$, $\hat{w}_S$ and $\hat{w}_B$ from equations (5), (6) and (3) respectively. ———, the free side of the plate: -----, the excited side of the plate.

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Modalanalyse og dens anvendelse til produktudvikling.

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1. INDLEDNING

At foretage modalanalyse på en konstruktion vil sige, at analysere dennes dynamiske egenskaber ud fra et kendskab til konstruktionens egensvingningsformer, samt de dertil knyttede egenfrekvenser og dæmpninger.

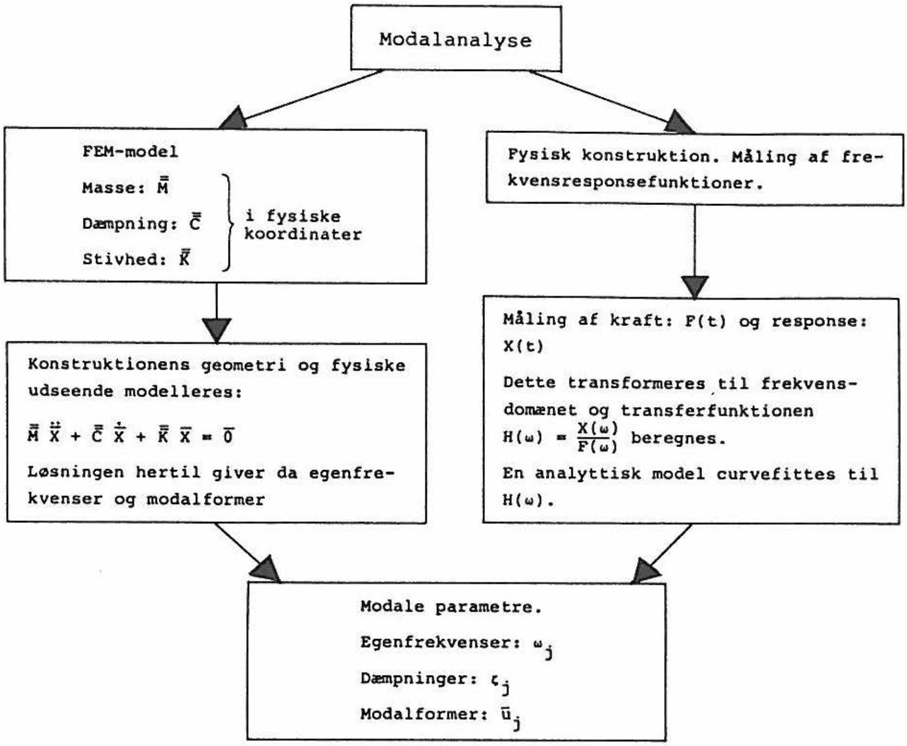
Den stadige udvikling inden for PC- og FFT-områderne har nu muliggjort opbygningen af et effektivt og mobilt modalanalyseudstyr. Hermed er der skabt en basis for gennemførelse af modalanalyser in-situ.

2. PRINCIPPET I MODALANALYSE

Indenfor produktudvikling og -konstruktion er man ofte henvist til forsøg-fejl-metoden, når man vil forbedre en konstruktions dynamiske egenskaber. Er man i en udviklings-/konstruktionsfase for et nyt produkt, er man således ofte nødt til at fremstille dyre prototyper, før man får kendskab til konstruktionens dynamiske egenskaber.

Bestemmelse af de modale parametre foregår enten ud fra kendskab til stivheds-, dæmpnings- og massefordelingen af en konstruktion, eller ud fra målinger på en eksisterende konstruktion.

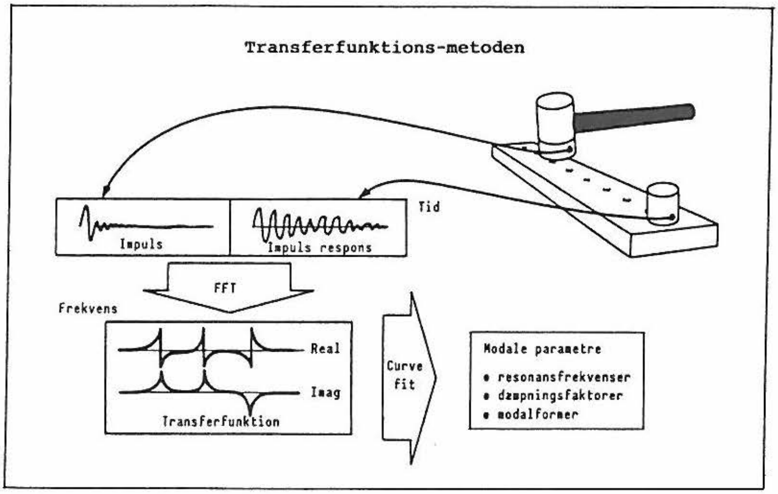
Førstnævnte metode anvendes især i de indledende designstadier, hvor prototypen endnu ikke er fremstillet. Den måletekniske fremgangsmåde anvendes især til verifikation af eksisterende konstruktionsegenskaber, samt som grundlag for videreudvikling. Der fokuseres her på den måletekniske modalanalyse.



Figur 1
Beregningsmæssig hhv. måleteknisk opnåelse af de modale parametre.

3. MÅLE- OG ANALYSETEKNIK

Den her anvendte teknik baserer sig på måling af transferfunktioner bestående af accelerationssignalet (eller hastighed eller udbøjning) ét sted på konstruktionen, divideret med kraftsignalet målt et andet (evt. det samme) sted på konstruktionen. For en konstruktion med N frihedsgrader skal der bestemmes N transferfunktioner, fx responset i én frihedsgrad forårsaget af en kraft i hver af de N frihedsgrader. For lineære konstruktioner med viscos og proportional dæmpning er disse N transferfunktioner tilstrækkelige til beskrivelse af konstruktionens modale parametre.



Figur 2
Principiel fremgangsmåde ved måling af transferfunktioner.

3. MÅLE- OG ANALYSETEKNIK

Den nødvendige måleteknik består således af en krafttransducer, et accelerometer, nogle ladningsforstærkere samt en frekvensanalysator (FFT).

Til de målte transferfunktioner curve-fittes nogle matematisk udtrykte funktioner, ud fra hvilke de modale parametre fremgår. Dette kan danne grundlag for en "tegnefilm" på en PC-skærm, visende de enkelte modale udbøjningsformer.

Når de modale parametre er bestemt, kan de anvendes til følsomheds- og konsekvensberegninger, dvs. det kan beregnes hvor på konstruktionen en evt. afstivning vil give størst effekt på de modale størrelser. Endvidere kan konsekvenserne af konkrete ændringer beregnes.

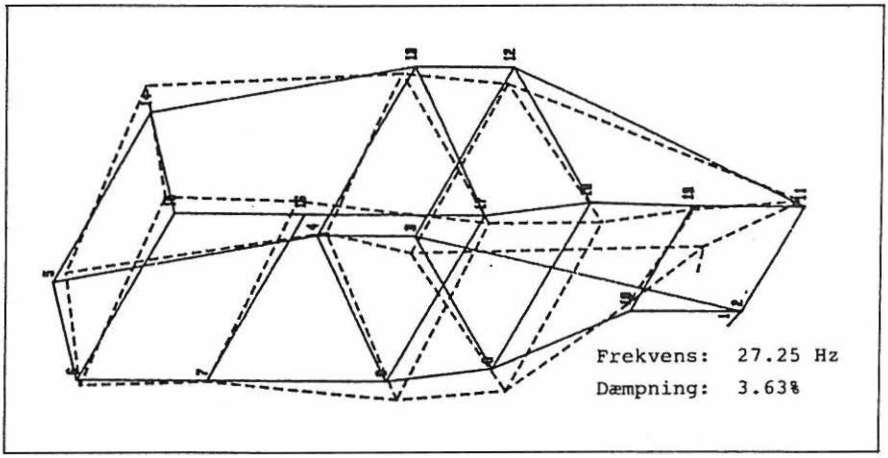
Beregning af konstruktionens svingningstilstand ud fra et givet kraftinput kan ligeledes foretages. Beregning i tidsdomænet og i frekvensdomænet er muligt.

4. ANVENDELSESEKSEMPEL

En bilimportør ønskede at ombygge en station-car model til en pick-up model. Ombygningen krævede indgreb i de bærende dele af karrosseriet, og der ønskedes klarhed over konsekvenserne på de dynamiske egenskaber. Denne klarhed blev skabt ved hjælp af en modalanalyse af den uombyggede hhv. den ombyggede model. Resultaterne ses af tabel 1, og på figur 3 er station-car'ens torsionssvingning ved 27.3 Hz vist.

Model	Frekvens Hz	Dæmpning %	Bemærkninger
Pick-up	25,7	3,25	Overvejende torsion (~179 km/h)
	27,5	2,39	Overvejende torsion (~192 km/h)
	29,3	1,79	Overvejende bøjning (~205 km/h)
Station-car	23,3	1,95	Bøjning (~163 km/h)
	27,3	3,63	Torsion (~191 km/h)

Tabel 1
Oversigt over analyseresultater.



Figur 3
Torsionssvingning, station-car.

Den ombyggede vogn udviste nogenlunde samme dynamiske egenskaber som det oprindelige køretøj, og alle egenfrekvenser var over det kritiske område svarende til hjulfrekvensen ved kørehastighed.

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BYGNINGSVIBRATIONER FRA PÆLERAMNING

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INDLEDNING

I forbindelse med ramning af jernbetonpæle og jernspuns kan der i nærliggende bygninger opstå vibrationer, der er meget kraftigere end de, der normalt forekommer. Ofte skal pæle og spuns rammes mindre end 1 meter fra de nærmeste bygninger. Er disse dårligt vedligeholdte eller svagt funderede, kan det give anledning til spørgsmålet om bygningerne vil tage skade af vibrationerne.

Indtil videre findes der ikke ensartede retningslinier i forskellige lande for måling og vurdering af bygningsvibrationer i disse tilfælde. Nogle af de mest udførlige retningslinier findes i et forslag til tysk norm "ENTWURF DIN 4150" (1). Efter disse skal der måles maksimalværdier af vibrationshastigheden på bygningernes fundamenter og på bygningens øverste etagedæk. De grænseværdier, der skal anvendes, er afhængige af frekvensen og bygningens stand. Grænseværdien varierer mellem 3 mm/s og 50 mm/s. I det følgende beskrives erfaringer med måling og vurdering med udgangspunkt i dette normforslag.

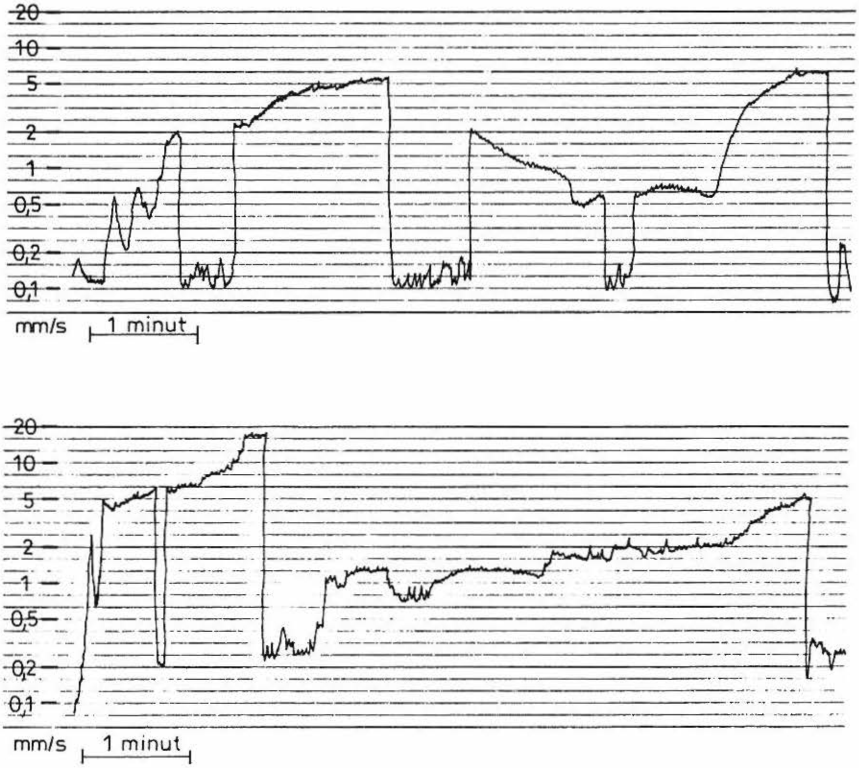
RAMNING

Moderne rammeudstyr er hydraulisk drevet og har automatisk styring af faldhøjde og slagfrekvens. Faldhøjden er normalt mellem 10 cm og 60 cm, og slagfrekvensen kan varieres fra enkeltslag til knap 1 slag pr. sekund. Selve hammeren vej-er 4 - 5 tons. Jernbetonpæle har normalt et tværsnit på 25 cm x 25 cm eller 30 cm x 30 cm og en længde mellem 6 m og 18 m. Spunsjern er normalt ca. 1/2 m brede og rammes ned to og to. Længden varierer som pælene.

MÅLINGER I PRAKSIS

Den grænseværdi, der skal anvendes i hvert enkelt tilfælde, fastlægges på grundlag af en erfaringsmæssig vurdering af bygningens tilstand og fundering og eventuelt geotekniske undersøgelser. Der er selvfølgelig ikke tale om skarpe grænser.

Det er vanskeligt på forhånd at forudse, hvorledes vibrationshastigheden ændres under ramningen. På figur 1 er vist eksempler på forskellige rammeforløb.



Figur 1. Eksempler på vibrationshastighedens ændring under ramning.

Ved de målinger, Skandinavisk Lydteknik har udført, er der ikke blevet foretaget en konstant overvågning af vibrationssignalets frekvensmæssige sammensætning, fordi det har vist sig, at den under normale forhold er nogenlunde konstant. I stedet registreres maksimalværdien af vibrationshastigheden i frekvensområdet 1-100 Hz. Normalt etableres der faste målekæder på de dele af fundamenterne, der er nærmest rammestedet. Desuden bliver det med bærbart måleudstyr undersøgt, om der forekommer for kraftige vibrationer andre steder. Dette er meget sjældent tilfældet.

Signalerne overvåges på et oscilloskop, og der foretages en registrering på en papirstrimmel som dokumentation. På strimlen er det muligt at aflæse den maksimale vibrationshastighed ved hvert enkelt slag. Overvågningen sikrer ligeledes, at bygningen er faldet til ro inden det næste slag, således at resonansfænomener undgås. Der er v.h.a. radio forbindelse med rambukføreren, så ramningen kan stoppes umiddelbart, hvis en overskridelse af grænseværdien er blevet konstateret. Som regel er det muligt at fortsætte rammearbejde med nedsat faldhøjde. Omvendt er det nogle gange muligt at tillade, at faldhøjden sættes op, så rammearbejdet kan forløbe hurtigere. Hvis der i undergrunden er rester af ældre bygningsverker, kan det give anledning til en hurtig og kraftig stigning i vibrationsniveauet, hvis pæle eller spunsjern rammer disse. Dette forekommer temmelig ofte, og så er det vigtigt at stoppe ramningen med det samme.

Det kan blive nødvendigt helt at opgive at gennemføre det oprindeligt planlagte rammearbejde. Dette var tilfældet ved en opgave, hvor et nedrevet hus i en tætbebygget bykerne skulle erstattes med et nyt. Selv med den mindst mulige faldhøjde var det ikke muligt at holde vibrationerne under den fastlagte grænseværdi. Det blev først muligt, da projektet blev omarbejdet, så det blev baseret på 25 cm x 25 cm pæle i stedet for de oprindelige 30 cm x 30 cm pæle. I et andet tilfælde blev ramning af spunsjern standset på grund af for kraftige vibrationer. Ved at ramme jernene ét ad gangen i stedet for som normalt to og to, og så standse ramningen når grænseværdien blev nået, var det alligevel muligt at etablere en brugbar spunsvæg.

SKADER

Af hensyn til erstatningsspørgsmålet bliver de omkringliggende bygninger grundigt fotograferet, så det er muligt bagefter at konstatere om eventuelle skader skyldes rammearbejde, eller om de fandtes i forvejen. Der er ikke konstateret alvorlige skader som følge af de rammearbejder, hvor der er foretaget overvågning af bygningsvibrationer. I enkelte tilfælde er der opstået mindre skader af kosmetisk art, f.eks. nedfald af puds, dannelse af små revner og udvidelse af bestående revner.

MENNESKERS REAKTIONER

Vibrationer fremkaldt af rammearbejde bliver mennesker ikke normalt udsat for, så når de opleves, giver de ofte anledning til en kraftig reaktion. Reaktionerne spænder lige fra irritation til frygt for bygningssammenstyrtning, og ofte bliver normal udførelse af arbejde besværliggjort eller umuliggjort. Bygninger kan som regel tåle kraftigere vibrationer end personer i bygningerne umiddelbart forestiller sig. Det har vist sig, at måling af bygningsvibrationer har en ekstra fordel ved at virke beroligende på de personer, der udsættes for vibrationer, hvis de er bekendt med at der udføres målinger.

VURDERING

Erfaringerne fra de rammearbejder, hvor Skandinavisk Lydteknik har overvåget bygningsvibrationer, viser, at det ved at lave målinger på fundamenterne af vibrationshastigheden i frekvensområdet 1-100 Hz, og derefter styre rammearbejdet så maksimalværdien ikke overstiger 4 mm/s i de aller svageste bygninger og ikke overstiger 8 mm/s i bygninger i lidt bedre stand, er muligt at undgå stort set alle skader. Der er i nogle tilfælde blevet konstateret begyndende skader, når disse værdier er blevet overskredet.

Selvom vibrationsmålinger umiddelbart virker fordyrende på rammearbejdet, indebærer det store fordele:

- Risikoen for alvorlige skader bliver uhyre ringe.
- Risikoen for mindre skader bliver væsentlig mindre
- Det bliver eventuelt muligt at gennemføre et projekt, der ellers måtte opgives.
- Det muliggør ofte forsikring på bedre vilkår.
- Det virker beroligende på personer i omgivelserne.

Vibrationsmålinger anvendes mest fordelagtigt, hvor:

- Der rammes tæt ved bygninger.
- Disse bygninger er gamle, dårligt vedligeholdt, dårligt funderede eller specielt bevaringsværdige.
- Undergrundens beskaffenhed er mere eller mindre ukendt eller fyldt med rester af ældre bebyggelser og anlæg.

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DETERMINATION OF THE SOURCE CHARACTERISTICS OF FLUID MACHINES

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Introduction

Fluid machines, such as engines, pumps, fans and compressors generates flow and pressure fluctuations in ducts or pipes connected to the machines. These fluctuations can cause mechanical damage to the system, and can radiate as noise to the surrounding medium through the pipe walls of through openings. When designing duct- and piping-systems to minimize these unwanted effects it is essential to know the characteristics of the machines as sources of fluid borne noise. A review of different source characterization methods can be found in [1].

Linear models

The most widely used way to analyse transmission of plane acoustic waves in ducts or pipes is linear acoustic filter theory. It is natural to extend this analogy with electrical transmission line theory to the source and describe it in the frequency domain by a source strength and an internal source impedanse (fig 1). This model have been used, for combustion engines [2], for hydraulic pumps [3], and for fans [4].

The source is here described by a linear differential equation with time invariant coefficients, i.e. it is a linear time invariant model. The source characteristics will normally have to be determined experimentally. When nothing is known about the source impedance one of the values, 0 (i.e. constant pressure source), ∞ (i.e. constant volume velocity source) or $\rho c/s$ (i.e. reflection free source) is normally assumed.

When analyzing the source impedance at, for instance, the outlet port of a single cylinder combustion engine it is seen that the impedance is infinite when the exhaust valve is closed and finite but depending on the position of the piston when the valve is open. This means that the source impedance is inherently time dependent. The adequate formulation of the problem is then a linear differential equation with time varying coefficients.

$$A(t)\,\ddot{Q}(t) + B(t)\,\dot{Q}(t) + C(t)\,Q(t) = P_s(t) - P(t) \qquad (1)$$

where Q is the volume velocity, P is the pressure and P_s is the source pressure. We then have a linear time variant model. If it is possible to derive a theoretical model for the source the problem can be solved in the time domain. A lumped parameter version of this approach have been used for refrigeration compressors [5] and a two-stroke engine [6].

Since all the time varying quantities in eq (1) are periodic it is possible to obtain a frequency domain formulation of eq (1) by expressing them in complex Fourier series.

$$A(t) = \sum_{n=-\infty}^{\infty} A_n e^{jn\omega_0 t}, \quad B(t) = \sum_{n=-\infty}^{\infty} B_n e^{jn\omega_0 t}, \quad C(t) = \sum_{n=-\infty}^{\infty} C_n e^{jn\omega_0 t}$$

$$Q(t) = \sum_{n=-\infty}^{\infty} Q_n e^{jn\omega_0 t}, \quad P(t) = \sum_{n=-\infty}^{\infty} P_n e^{jn\omega_0 t}, \quad P_s(t) = \sum_{n=-\infty}^{\infty} P_{sn} e^{jn\omega_0 t} \qquad (2)$$

The terms in eq (1) where two time varying quantities are multiplied result in a double sum which can be rearranged in matrix form resulting in the following expression

$$[Z_s](Q) = (P_s) - (P) \qquad (3)$$

where $[Z_s]$ is the source impedance matrix and (P_s) is the source pressure vector. The diagonal elements of this matrix corresponds to the time-invariant model and the elements outside the diagonal represent coupling between different frequency components at the source. This model have been used for a combustion engine air induction system [7].

Non-linear models

There excist a number of numerical methods for solving the full unsteady gas-dynamic equations for one dimensional fluid flow in pipes as for instance the method of characteristics and finite difference methods. If a theoretical non-linear model for the source can be derived the unsteady fluid flow in the pipe can be calculated by these methods. This approach is obviously much more complicated than the linear acoustic approach, but has been used for simple engines and exhaust systems [8][9]. The largest pressure in the piping-system pulsation normally occurs at the source so it should be the most non-linear element in the system. It might therefore be interesting to combine a non-linear source model with a linear model for the rest of the system.

Esperimental determination of source characteristics

For the linear time invariant model two methods have been used to measure the source characteristics. The two-microphone method [10] requires a high intensity secondary sound source to send sound towards the machine while it is active. If the secondary source can dominate over the sound from the machine itself the source impedance can be determined from pressure measurements at two positions in the pipe. To determine the source strength an additional pressure measurement over a known load impedance is required. The main problem with this method is to find a secondary source that can give sufficiently high sound levels to dominate over the machine while at the same time not violating the assumption of linearity. Another problem might be that sending these high intensity signals towards the machine could alter its source characteristics. The two-microphone method have been used to measure the source impedance of combustion engines [2].

The two-load method requires complex pressure measurements to be made over two known load impedances. From these measurements the two unknowns, source impedance and source strength, can be calculated by solving a simple equation system. The input impedances of the loads have to be known for the actual conditions occuring in the piping-system coupled to the machine. This can be achieved by measuring the impedances with the two-microphone method using the machine as the source. The two load method have been used for source characteristics measurements on hydraulic pumps [3].

A comparison of source impedance measurements by the two methods is shown in fig 2 from [11] where also information on how to minimize errors in the two-load method is included. Information about errors in the two-microphone method can be found in [10].

For the linear time variant model modified forms of the above described measurement methods can be used. For the two microphone method broad band excitation can no longer be used. The secondary source will have to be able to send high intensity pure tones, so that the reflected sound at other frequencies than the excitation frequency can be measured. The secondary source will have to dominate over the fluid borne noise from the machine at all frequencies i.e. not only at the excitation frequency. This will be very difficult if not impossible to implement. The modified form of the two-load method, the multiple load method, seems to be a much more realistic alternative in this case. Since we in the model of eq (3) have more than two unknowns we simply aquire more equations by using more loads. If we are interested in studying n tones in the tone spectrum of the machine we will need 2n+1 loads to determine the source impedance matrix and the source strength vector. One of the conclusions of [11] was that in order to get good results from the two-load method some overdetermination was required. This means that more than 2n+1 loads will probably have to be used for the multiple load method.

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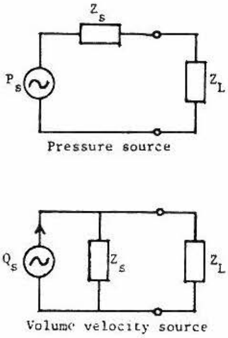

Figure 1
Equivalent acoustic sources

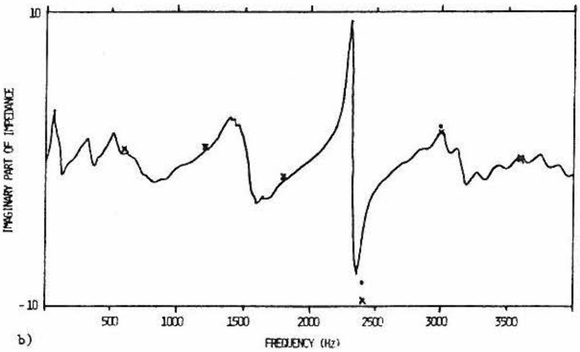

Figure 2 Measured imaginary part of source impedance. ——— two-microphone method, ·x two-load method, over-determined problem analyzed by two different methods [11]

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DESCRIPTION OF A FLOW NOISE TEST FACILITY

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Introduction

At the department of Technical Acoustics (TA) at the Royal Institute of Technology in Stockholm a test facility for flow acoustic research and development works has been built. This new facility offers a quiet air flow for studies of aero-acoustical phenomenons.

The test facility is primarily intended for TA:s own research and development activities but will also be available for external users. The facility can be used for research and development aids and for product testing in many fields where airflow and noise in combination needs to be studied.

Some examples of ongoing or coming research projects are:

- Development of measurement methods for studying sound transmission in duct elements, for example, measurement methods for determining the transmission loss for a silencer and the four-pole parameters of a duct element, see ref 1 & 2.
- Studies on reactive silencers based on multiple scattering (Bragg-scattering), see ref 3.
- Studies on simple two-pole source-models for flow-generated noise from control valves, orifice plates, sudden area expansions, etc.
- Studies on methods for measuring the acoustic intensity in systems with flow.

Description of the test facility

Figure 1 shows a section of the test facility, which consists of 3 chambers. The centrifugal fan (1) sucks the air from the fan-room (A) through an inlet sound attenuator with baffle elements (2) and further through an air cooler and droplet eliminator (3). The fan drives the air through an outlet silencer (4) to the room (B) under the fan room where a pressure is built up. This pressure drives the air through the duct configuration mounted in the heavy air-tight doors (5) which separates the room (B) from a reverberation room (C)(with a volume of 200 m^3). The duct configuration can for example consist of an (inlet) anechoic termination (6), a loudspeaker unit (7) for generating test signals, test object (8) and an (outlet) anechoic termination (9). From the reverberation room the air stream flows back to the fan-room through a big sound attenuator (12) with baffle elements.

One advantage with this design is that the fan is not directly connected to the test duct but drives the test duct from the pressure reservoir in room (B). This design blocks effectively the structure-borne sound paths from the fan unit to the test duct. It also eliminates the need for transition elements for different test duct dimensions. Finally with the above design concept sound power - and sound intensity measurements can be made in the duct or in the reverberation room.

This test facility allows a continuously adjustable air flow due to the frequency converter that infinitely controls the standard squirrel-cage motor which drives the fan. Max air flow is 2.6 m^3/s in combination with a max total pressure increase of 3600 Pa, which gives a maximum air speed of 75 m/s in the test duct. The air flow can be reversed and the temperature can be controled due to the air cooler. The test facility allows test ducts up to ø630/ø645 mm, but with open doors (5) even bigger objects can be tested.

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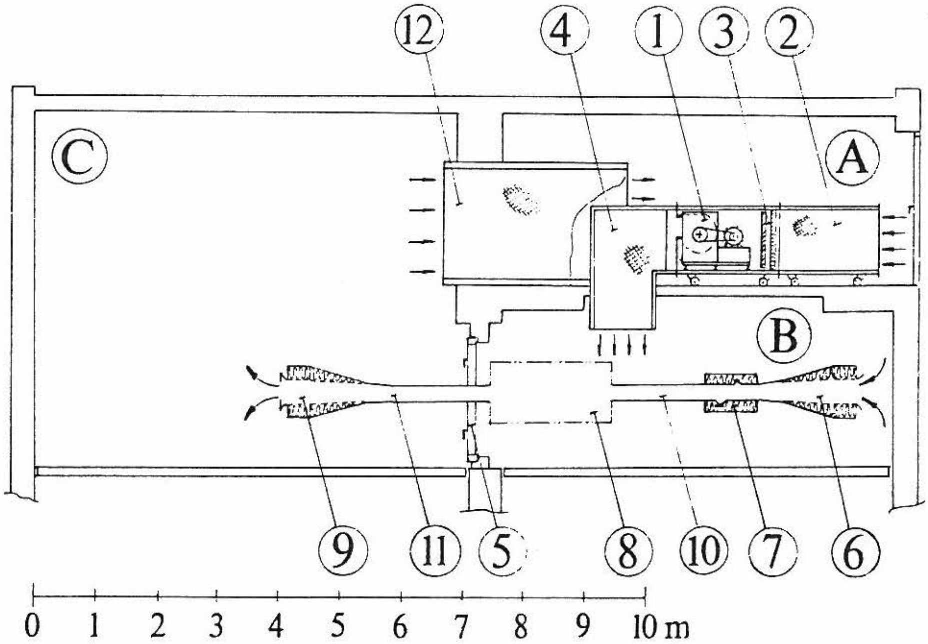

Figure 1 Flow noise test facility

A Fan room
B Pressure room
C Reverberation room
1 Fan unit
2 Sound attenuator
3 Air cooler
4 Sound attenuator
5 Heavy air-tight doors
6 Anechoic termination
7 Loudspeaker unit
8 Test object
9 Anechoic termination
10 Test duct
11 Test duct
12 Sound attenuator

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DEVELOPMENT OF A SOUND REDUCING EXHAUST FIRED BOILER AQ-16

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INTRODUCTION

In connection with the development of a new type of Exhaust Fired Boiler applied for diesel engines, Aalborg Boilers requested an investigation of the acoustical properties of the boiler.

The purpose of the investigation was to elucidate whether it is possible to apply the boiler as the only silencer in the exhaust system and also whether it is possible to improve the intake and outlet chambers with special emphasis on obtaining a substantial insertion loss in a wide frequency range. Furthermore, the impact of the airflow through the boiler relating to the insertion loss was investigated.

The boiler is a combined type of exhaust fired boiler with an oil fired boiler as supplement. The boiler is a vertical, cylindrical type with the exhaust gas being conducted through an intake chamber and into a number of water-cooled pipes. From the pipes the exhaust gas is led through the outlet chamber. See Figure 1.

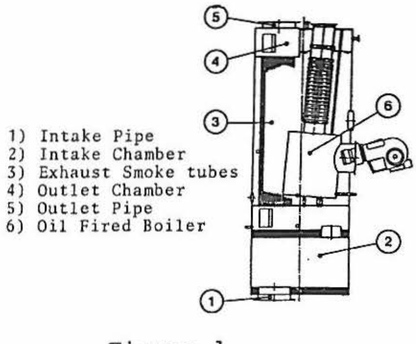

Figure 1.
Construction of an exhaust fired boiler.

MEASUREMENT METHODS

The insertion loss of the boiler has been evaluated by means of scale model experiments. The scale model was made of steel in a geometric scale 1:13.8 of the boiler shown in Figure 1. The geometric scale factor 1:13.8 has been chosen with the objective of obtaining a scale factor 1:10 for the frequency at the temperature 280°C of the exhaust gas.

The scale model was connected with an intake pipe and outlet pipe with dimensions corresponding to a typical exhaust installation in a ship. The total length of the pipe system in the scale model was 2430 mm corresponding to 33.4 metres in full scale. This length was constant during all measurements.

The measurement set-up applied is shown in Figure 2. The loudspeaker was supplied with band-pass filtered white noise in the frequency range 100 Hz to 10 kHz.

The measuring microphone was placed 250 mm from the opening of the outlet pipe in the axial direction of the pipe.

The insertion loss was determined as the difference of the sound pressure level in the measuring point with and without the boiler being mounted in the pipe system. Thus, the insertion loss expressed the obtainable reduction of the sound pressure level by installing the boiler in an already existing exhaust installation.

The measured noise signals were analysed both in one-third octave frequency bands and in narrow band frequency bands.

In order to evaluate the impact of the airflow through the boiler on the insertion loss, the pipe system was connected to a fan equipped with silencers. The airflow in the outlet pipe varied during the measurements between 13 m/s and 20 m/s. With the airflow through the scale model a high level of flow-induced noise was generated. Even when applying special loudspeakers it was not possible to obtain an acceptable signal/noise ratio between the loudspeaker signal and the flow-induced noise.

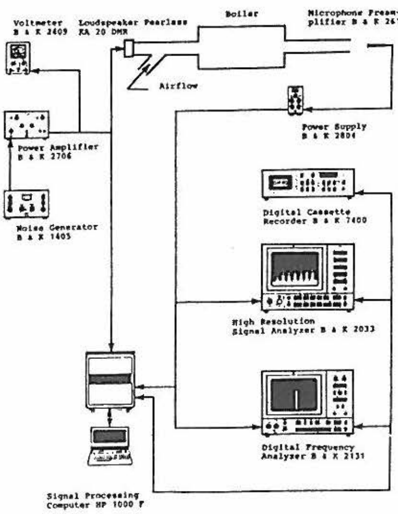

Figure 2.
Measurement set-up for scale modelling test.

Consequently, at measurements carried out with airflow the insertion loss was determined applying a 2-channel correlation measuring method. The input noise signal measured at the loudspeaker terminals were connected to one of the channels and the output signal measured in the microphone positions to the other channel. The cross correlation function $R_{xy}(t)$ between the two noise signals were determined. Furthermore, the autocorrelation function $R_{xx}(t)$ was determined for the input signal. The transfer function $H_{xy}(f)$ was determined as,

$$H_{xy}(f) = \frac{F_{xy}(f)}{F_{xx}(f)}$$

The cross spectrum $F_{xy}(f)$ and the power spectrum $F_{xx}(f)$ is determined by a forward Fourier transformation of the respective correlation functions. The transfer function $H_{xy}(f)$ is independent of the flow-induced noise, as $F_{xy}(f)$ and $F_{xx}(f)$ only contain frequency components correlated with the input signal.

The insertion loss is determined as the difference between the measured transfer functions with and without the boiler mounted in the pipe system.

For a few of the investigated scale models, the insertion loss has been calculated by applying a computer programme. Only axisymmetric model variants have been calculated. The insertion loss is determined by a combination of finite element calculations and calculations using a one-dimensional Four-pole model. This latter model is applied for acoustical calculations in duct systems with special reference to exhaust silencer design purposes (3).

The Four-pole model is used in sections of the exhaust system, where the theory of plane wave propagation is valid and the finite element model is used for the sound reducing element, e.g. the intake chamber.

The individual elements in the exhaust system are each described by means of a matrix with the relation between the pressure and the volume velocity before and after the element, being given as,

$$\begin{bmatrix} P_1 \\ V_1 \end{bmatrix} = \begin{bmatrix} A & B \\ C & D \end{bmatrix} \cdot \begin{bmatrix} P_2 \\ P_2 \end{bmatrix}$$

For each element the Four-pole parameters A, B, C and D are determined according to (3). The matrix for the total system is calculated as the matrix product of all matrices.

Four-node isoparametric elements are applied in the finite element model for which local reacting boundary conditions (2) are assumed.

RESULTS OF MEASUREMENT

During the investigation a total of approximately 50 different scale models has been examined.

In Figure 3 is shown the insertion loss measured with and without airflow through the scale model. The measurements are performed using cross correlation technique.

There was no significant difference between the measured insertion loss with and without airflow of the investigated scale models.

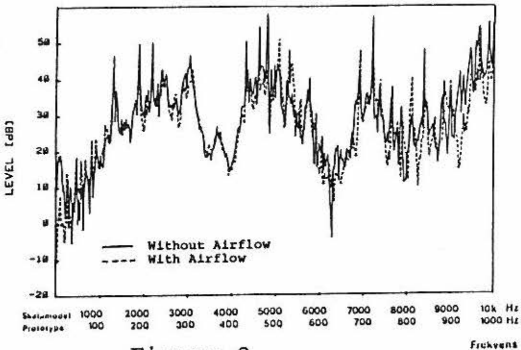

Figure 3.
Measured insertion loss with and without airflow through one of the scale models.

In Figure 4 is shown an example of a typical insertion loss curve for one of the scale models. Furthermore, the calculated insertion loss is shown by applying Finite Element calculations.

On the basis of the investigation performed by scale modelling tests, a boiler has been constructed in full scale. The boiler has been installed in a ferry. The insertion loss of the boiler, including the exhaust pipe system, has been determined using cross correlation technique. The results of the measurements are in fine accordance with the scale model measurements.

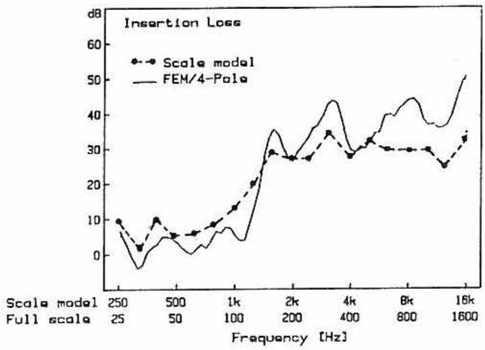

Figure 4.
Measured and calculated insertion loss for one of the boiler variants without airflow.

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A FINITE ELEMENT MODEL FOR INFINITE SPACE RADIATION.

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1. Introduction

A model has been developed for harmonic acoustic radiation from axisymmetric sources of complex shape embedded in an infinite fluid. The source is inscribed in an hypotethical sphere and the field distribution within the sphere calculated by the use of finite elements. The field exterior to the sphere is described by an expansion in analytic functions, figure 1.

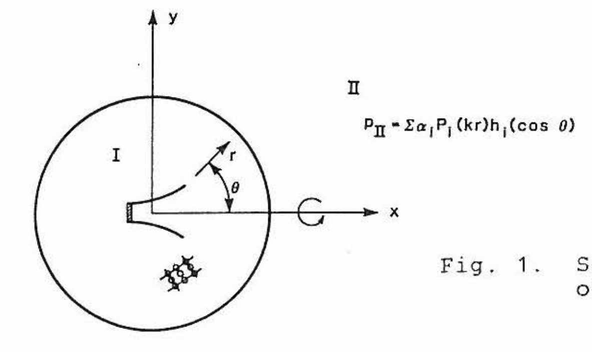


Fig. 1. Schematic drawing of the geometry.

The variational method used for this system is based on the usual conservative functional yielding the Helmholtz equation. Complex acoustic admittances are then introduced by the method of La Grange multipliers. The resulting functional is considered complex throughout the necessary variational procedure.

The presentation will consider examples of such sound radiation. The accuracy of the model is investigated to a certain extent, and the problem of non-unique solutions at special frequencies (the hard walled modes of the spherical volume) is described.

2. Mathematical Model

The functional has the form

$$\pi = \int_{\Omega} \frac{1}{\rho\omega^2} [\nabla p \nabla p - k^2 p^2] d\Omega - \sum_i \frac{1}{j\omega} \int_{\Gamma_i} p^2 \; A_i \; d\Gamma_i$$

The terms are recognized as the kinetic and potential acoustic energies of Ω and the energies crossing the boundaries Γ_i. The Euler equations are $\nabla^2 p + k^2 p = 0$ with $\partial p/\partial r = -j\omega\rho A p$ at the boundaries.

The acoustic pressure of region I is approximated by $p_I = \lfloor N \rfloor \{p\}$, the normal finite element formulation; and p_{II} by an expansion in Legendre and spherical Hankel functions

$$p_{II} = \lfloor \psi \rfloor \{\alpha\}$$

with $\psi_m = P_m(\cos\theta) h_m(kr)$

A constant velocity source, and the field in II are coupled to the finite element volume by the admittances A_1 and A_2 respectively.

$$A_1 = \frac{u(\theta,r)}{\lfloor N \rfloor \{p\}} \quad , \quad A_2 = \frac{\frac{\partial}{\partial r} \lfloor \psi \rfloor \{\alpha\}}{\lfloor N \rfloor \{p\}} \cdot \frac{1}{j\omega\rho} \frac{\lfloor \varphi \rfloor \{\alpha\}}{\lfloor N \rfloor \{p\}}$$

Minimizing the functional yields:

$$\left[\lfloor M_1 \rfloor - k^2 \lfloor M_2 \rfloor \right] \{p\} - \lfloor M_3 \rfloor \{\alpha\} = \lfloor M_5 \rfloor$$

with $\lfloor M_1 \rfloor$ and $\lfloor M_2 \rfloor$ acoustic mass and stiffness matrixes,

$$\lfloor M_3 \rfloor = \int_{\Gamma_2} \lfloor N \rfloor^T \lfloor \varphi \rfloor d\Gamma_2 \quad , \text{ and } \lfloor M_5 \rfloor = j\omega\rho \int_{\Gamma_1} \lfloor N \rfloor^T u(r,\varphi) d\Gamma_1$$

Because of the α coefficients an additional equation is needed. This is obtained by considering pressure continuity over the Γ_2 boundary by a weighted residual method with $\lfloor \varphi \rfloor$ as weighting functions.

$$\int_{\Gamma_2} (p_I - p_{II}) \lfloor \varphi \rfloor d\Gamma_2 = 0 \quad , \text{ gives us}$$

$$\lfloor M_3 \rfloor^T \{p\} - \lfloor M_4 \rfloor \{\alpha\} = 0 \quad , \text{ with}$$

$$\lfloor M_4 \rfloor = \int_{\Gamma_2} |\psi||\varphi| d\Gamma_2$$

The final matrix equation to be solved is therefore:

$$\begin{bmatrix} \lfloor M_1 \rfloor - k^2 \lfloor M_2 \rfloor & \lfloor M_3 \rfloor \\ \lfloor M_3 \rfloor^T & \lfloor M_4 \rfloor \end{bmatrix} \begin{Bmatrix} p \\ \alpha \end{Bmatrix} = \begin{Bmatrix} \lfloor M_5 \rfloor \\ 0 \end{Bmatrix}$$

3. Results

Figure 2a shows the directivity of a horn loudspeaker radiating into a free field. 9-point LaGrange elements were used. The model incorporates 638 complex modal values and 12 outer field coefficients. The CPU-time spent in solving the matrix equation was 56.6 sec. Figure 2b shows the axial response within the finite element volume at the same frequency.

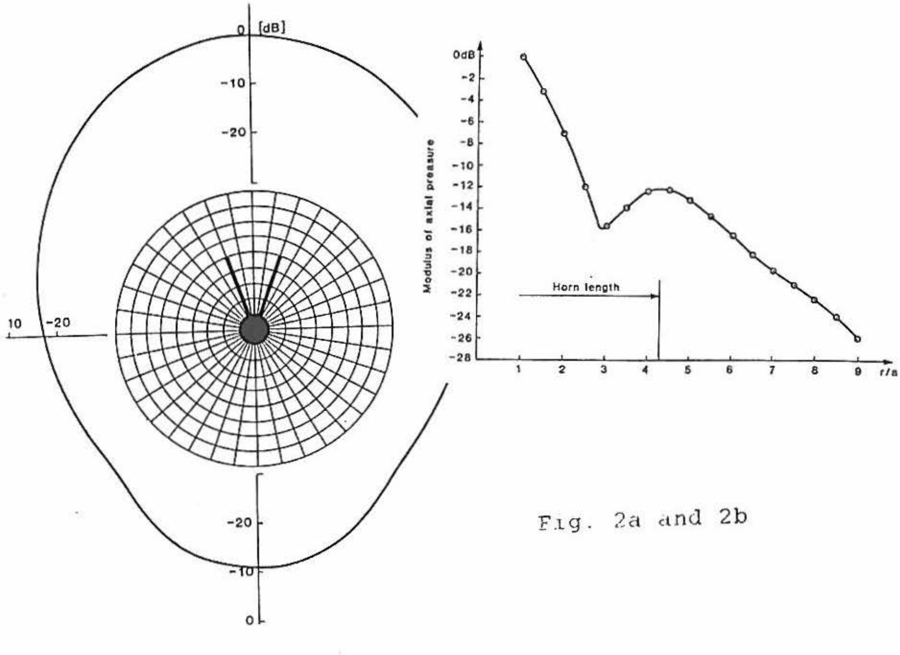
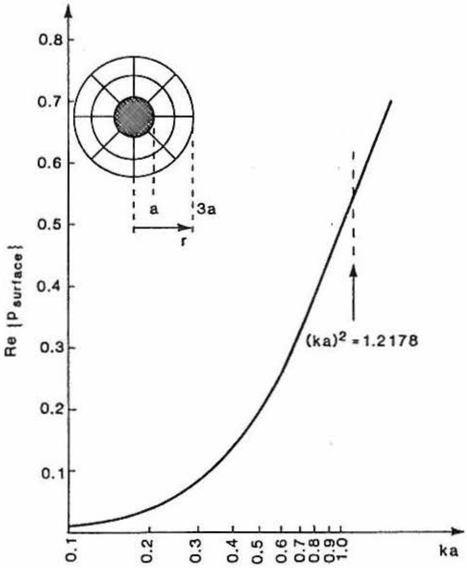


Fig. 2a and 2b

The model will have non-unique solutions at the eigenfrequencies of the hard walled spherical volume.

This effect was investigated by calculating the surface pressure on a monopole. Figure 3 shows the situation for a finite element volume having its first resonance at $(ka)^2 = 1.2178$. The effect on the surface pressure is observed to be a very narrow band effect indeed.



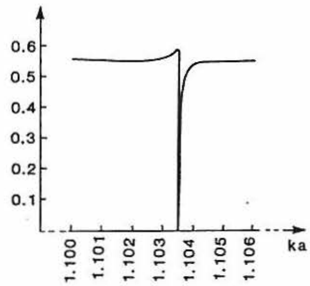


Fig. 3.

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THE FINITE ELEMENT METHOD (FEM) APPLIED TO SOUND FIELDS IN POROUS MATERIALS

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INTRODUCTION

The differential equations for sound-fields in porous materials are derived in this paper. An energy functional which is an integral form of the differential equations, is introduced. The FEM discretization is based on this functional. Axi-symmetric 8-node isoparametric square elements are used for calculation on a reflection muffler with two chambers. The calculations are compared with measurements. The long inlet and outlet pipes are modelled with a plane wave-theory (a four-pole-theory) which is coupled to the FEM-model. This hybrid method is computationally efficient.

The procedure is applicable in the most common situation where the diameter of the pipes is small in comparison with the diameter of the muffler itself. This often ensures that the waves are plane in the interesting frequency range.

EQUATIONS FOR A POROUS MEDIUM

Equation of continuity: $$-\rho\cdot\mathrm{div}(\underline{u}) = \Omega\cdot\frac{\partial\rho}{\partial t} \qquad (1)$$

ρ is the density, $\underline{u}$ is the mean velocity component, Ω is the porosity and t is time. The relation between ρ and the acoustic pressure p:

$$\delta\rho = \rho-\rho_s = \kappa\rho p \qquad (2)$$

ρ_s is the density in the undisturbed medium and κ is the compressibility.

$$\kappa = \frac{1}{p_s \gamma} \tag{3}$$

p_s is the static pressure and $\gamma=1$ in the case of isothermal compressibility and $\gamma=(C_v+R)/C_v$ for adiabatic compressibility. C_v is the specific heat capacity for a constant volume and R is the gas constant.

Equation of motion: $$-\nabla p = \rho_p \cdot \frac{\partial u}{\partial t} + \phi \underline{u} \tag{4}$$

ϕ is the flow resistance of the porous medium. ρ_p is the effective density of the ideal gas in the porous medium.

$$\rho_p = K_s \cdot \rho \tag{5}$$

$K_s>1$ is the structure factor which has a value between 3 and 7. Reference (1) shows that $K_s = 3$, when the medium is isotropic porous with the pores randomly distributed and infinitely rigid. If we introduce the speed of sound $C_p = (\Omega\kappa\rho_p)^{-\frac{1}{2}}$ we can reduce the previous equations to the wave equation (6).

$$\Delta p = \frac{\phi}{\rho_p \cdot C_p^2} \cdot \frac{\partial p}{\partial t} + \frac{1}{C_p^2} \cdot \frac{\partial^2 p}{\partial t^2} \tag{6}$$

The harmonic varying pressure $p \cdot e^{j\omega t}$ can be substituted into (6) if the sound field is stationary. ω is the angular velocity.

$$\Delta p + \left(\frac{\omega}{C_e}\right)^2 p = 0 \qquad \text{on V} \tag{7}$$

Here $C_e = C_p\ (1 - j\phi/(\rho_p\omega))^{-1/2}$ is equivalent to the speed of sound in a non-porous medium. The boundary conditions to (7) are,

$$\frac{\partial p}{\partial n} + (\phi + j\omega\rho_p)V_n = \frac{\partial p}{\partial n} + (\phi + j\omega\rho_p)A\ p = 0 \text{ on S}$$

or

$$p = p_o \quad \text{on S} \tag{8}$$

$\frac{\partial p}{\partial n}$ is the pressure derived with respect to a normal drawn outward from the surface, V_n is the particle velocity in the same direction, A is the admittance and p_o is a known pressure. Either V_n, A or p_o must be known along the whole surface S of the acoustic volume V if (7) is to be solved. Because of the energy-loss in the system expressed in the terms containing ϕ and A, we need to introduce the adjoint system which gains the same amount of energy, in order to use the Lagrange Hamilton formalism. The adjoint system then has a negative flow resistance $(-\phi)$, a negative complex conjugated admittance $(-A^*)$ and a complex conjugated wave number $(\omega/C_e)^*$. Let us introduce the total energy functional:

$$4\rho_p\omega^2\cdot\chi = \int_V (\nabla p\cdot\nabla q^* + \nabla p^*\cdot\nabla q)\,dV - \int_V (K^2 pq^* + (K^2)^* p^* q)\,dV$$
$$+ \int_S (\alpha V_n q^* + \alpha^* V_n^* q - \alpha^* V_n p^* - \alpha V_n^* p)\,dS + \int_S (\alpha A p q^* + \alpha^* A^* p^* q)\,dS \qquad (9)$$

q is the acoustic pressure in the adjoint system. The asterisk means complex conjugate. $K=\omega/C_e$ is the complex wave number and $\alpha=\phi+j\omega\rho_p$ makes the notation easier. A virtual variation of χ with respect to q* leads to (7) and (8) as conditions for χ being stationary. (Green's theorem is used). The acoustic pressure p is interpolated within each element according to,

$$p = \underline{N}^T\cdot\underline{P} \qquad (10)$$

$\underline{N}$ is the shape functions in a column matrix, $\underline{P}$ is the pressures in the nodes. Substituting the expression $q^* = \underline{N}^T\cdot\underline{Q}^*$ and (10) into (9) and making the virtual variation of χ with respect to $\underline{Q}$* and applying the condition of χ being stationary, we get the general linear FEM equations.

$$\left\{\int_V \nabla\underline{N}\cdot\nabla\underline{N}^T dV - \int_V K^2\underline{N}\cdot\underline{N}^T dV + \int_S \alpha A\underline{N}\cdot\underline{N}^T dS\right\}\cdot\underline{P} = -\int_S \alpha V_n \underline{N}\,dS \qquad (11)$$

FEM-APPLICATION

Figure 1 shows a reflection muffler with two chambers.

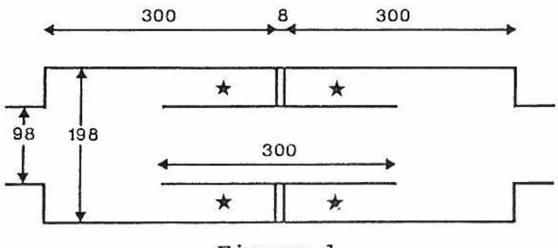

Figure 1.
Reflection muffler with dimensions in mm.
The inlet and outlet pipes are not shown.

The inlet and outlet pipes are modelled with a plane wave theory. The radiation impedance at the end of the outlet pipe is given by expressions in ref. (2). The source impedance is estimated to 5 times the characteristic impedance. Figure 2 shows the measured (full line) and the calculated (dotted line) insertion loss for that particular muffler.

The discrepancy beneath 50 Hz is due to the loudspeaker used in the measurements. It has a lower limiting frequency of about 50 Hz. The calculated peaks at approximately 540 Hz and 1620 Hz are due to the cavities marked with an asterisk. Each cavity acts like a quarter-wave-resonator and as a consequence good damping is obtained. These peaks are not measured due to uncertainties in the geometry and the signal

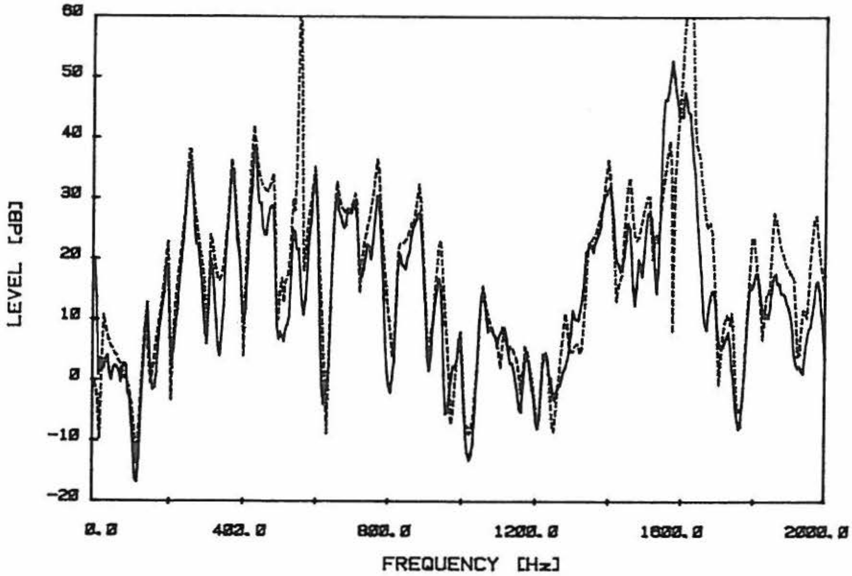

Figure 2.
Insertion loss.
——— measured ------- calculated.

to noise ratio. The limit frequency beneath which higher order waves are stable in the pipes is about 2 kHz. Higher order waves become less unstable as the frequency approaches 2 kHz. This explains the tendency towards increased discrepancy near 2 kHz.

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ACOUSTIC FINITE ELEMENT METHODS (AFEM) APPLIED TO PROBLEMS WITH POROUS ABSORBERS

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1. Introduction

In the design of a passenger vehicle such as aircraft, cars, etc, it is important to consider the low frequency noise and vibration. An interesting method to deal with these problems is the finite element method (FEM). Originally having been a structural mechanics tool, FEM has become more and more used for other areas in engineering analysis such as acoustics, heat transfer, fluid dynamics etc. Due to the relatively long wave lengths at low frequencies good results are obtained with rather coarse discretisation and thus the computational effort is reasonable both in time and cost.

Since the low frequency noise is rather difficult to handle, the effects of absorptive materials must be well understood to obtain an optimum use of the available space and the allowed weight budget in e.g. an aircraft. The high flight altitude requires thermal insulation and there might be beneficial effects from choosing a porous material wich has a good acoustic performance at low freequencies.

A finite element describing the behaviour of a porous material under certain assumtions has been developed at FFA.

2. Theoretical background

The effect upon a sound wave incident on a porous material is modelled in a macroscopic theory as friction loss when the fluid flows through the pores. Under the assumtion of a random distribution of the fibres in space and orientation, an isotropic material, giving the friction force $\underline{F}$ on the fluid,

$$\underline{F} = C \cdot \underline{u} \tag{1}$$

where C is a material constant
$\underline{u}$ is the fluid particle velocity

is obtained. If the effects of compressibility are neglected then the propagation of sound through the porous media is described by Helmholz equation with a complex factor taking the introduced absorption into account. It is important to points out that this kind of approach models a non-locally reacting material.

Through a Galerkin procedure a symmetric functional, which might be discretised in the usual FEM manner, is obtained. From this discretisation a finite element is developed for both plane and 3D applications.

An alternative way of describing porous media is through the specification of a normal surface impedance. This model which is of the locally reacting type does not take into account the fact that separate points in the absorbent affect each other due to the propagation of sound in the material. For plane waves which are normally incident this will be a reasonable assumption while for other cases it will be erraneous. The normal impedance formulation is built from the same basis as the finite element discussed above and hence depends upon the same material constant C.

The constant C in equation (1) is dependent on the type of material used and is frequency independent for a rigid absorbent. However, the assumption of a rigid material is not valid at low frequencies since in this regime the inertia of the material is important. If instead it is assumed that the material is allowed to move as if it was limp then the friction force $\underline{F}$ is no longer in phase with the motion of the air and thus C will be a complex constant

$$C^* = C^* \, (f_2/f) \tag{2}$$

where f_2 is a characteristic frequency of the material.

The characteristic frequency f_2 is a measure of the inertia effects caused by the motion of the limp material. For high frequencies C* will take the rigid value C while in the low frequency regime it will be altered considerably.

3. Application example

As a test of the bahaviour of the developed elements a model of a reverberant room was built, see figure 1. The room was excited in the upper right hand corner and in the lower right hand part a sample of a porous material with 0.1 m thickness and typical material parameters was placed. The frequency of the excitation was 200 Hz. The values choosen for f_2 and the corresponding mass densities are shown in table 1.

Case:	1	2	3	4
f_2:	0	52	260	520
M:	∞	61	12.2	6.1

Table 1 The characteristic frequency f_2 in Hz, an the mass density M in kg/m^3

4. Comparison between non-locally and locally reacting model of a rigid porous material

A rigid porous material model corresponds to a characteristic frequency f_2 of 0 Hz, i.e., inertia effects are neglected. This corresponds to case 1 i Table 1. Figure 2a and 2b show the results from calculations performed with the finite element and the normal impedance model respectively. The impedance formulation seems to overpredict the absorption compared to the finite element.

5. Effects of correcting for flexibility in the finite element

If the material is allowed to be limp then the results shown in figures 2c to 2e are obtained. The characteristic frequency f_2 corresponds to the values shown in Table 1, case 2 to 4. The result for a rigid material is shown in figure 2. For an increasing characteristic frequency, i.e. a decreasing mass density for the porous material, the acoustic energy in the room is partly moved to the interior of the absorbent. The sound pressure levels are decreasing as f_2 is increased up to 260 Hz. Increasing f_2 further gives higher levels both in the absorbent and in the room. It seems as if there is an optimum material for a specific frequency giving a high absorption.

6. Conclusions

A finite element calculation, modelling the non-locally reacting nature of a flexible porous absorbent, shows that there is in theory a best choice of the mass density for a given frequency. The behaviour of the flexible material is much like a dynamic vibration absorber tuned to a certain frequency.

The differences between the normal impedance formulation and this element indicate that for some problems errors are introduced by the locally reacting surface assumption.

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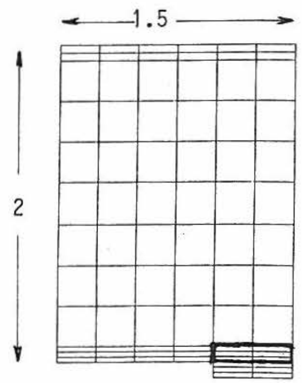

Figure 1 FE model of reverberant room. Dimensions in [m]

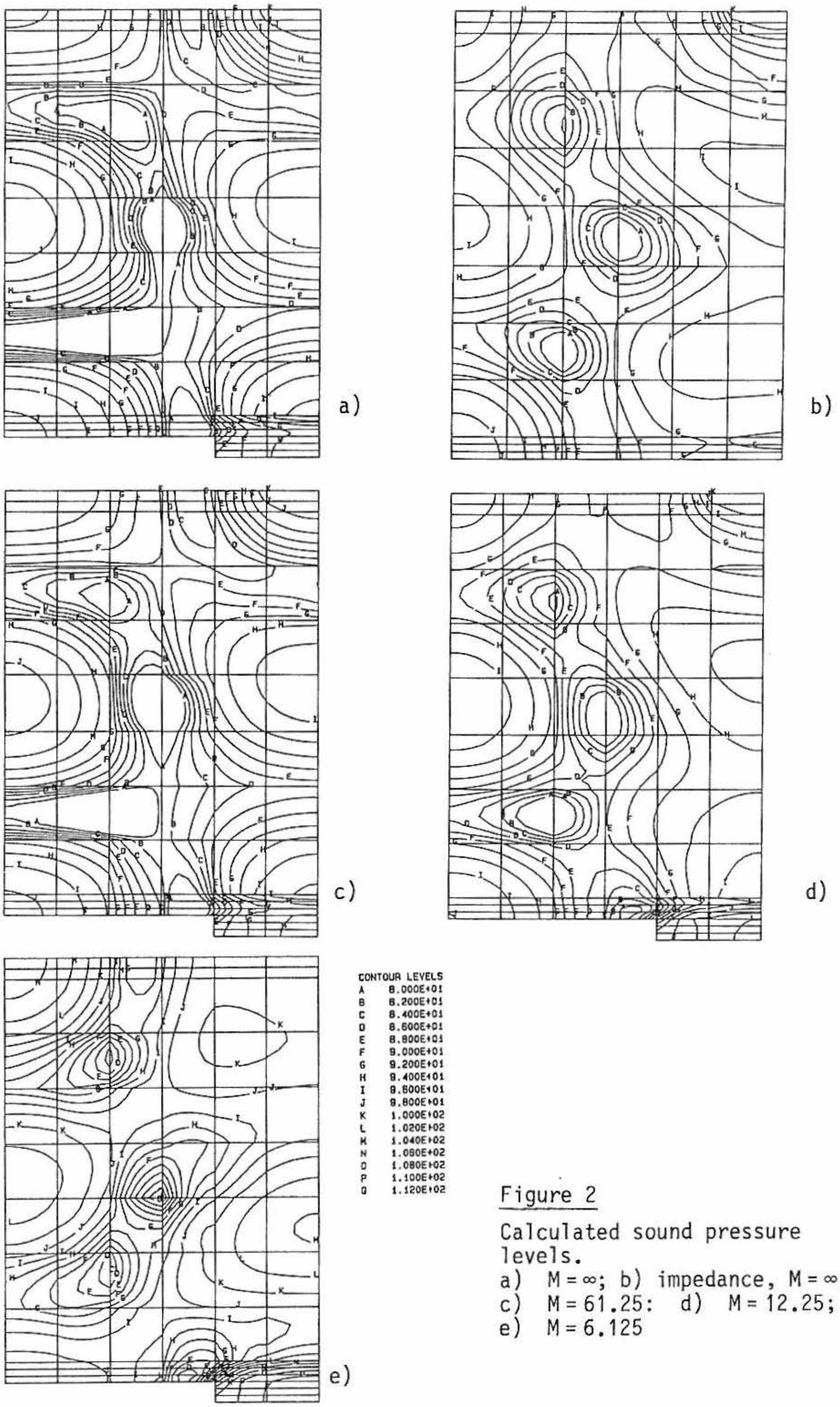

Figure 2

Calculated sound pressure levels.
a) $M=\infty$; b) impedance, $M=\infty$;
c) $M=61.25$: d) $M=12.25$;
e) $M=6.125$

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THE INTERACTION OF INCOHERENT SOURCES

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Introduction

The development of the two-microphone technique has made it possible to make sound power measurements in such circumstances where they have formerly been difficult to be performed. By using a measuring surface enclosing the source to be measured the effect of other sources to the result of sound power determination can be eliminated. One fundamental condition for the ability of the method is that losses are not enclosed by the measuring surface.

The sound power is generally considered as a property of the source and to be independent of the surroundings. So can often be assumed. However, it is possible that the total power of coherent sources differs from the sum of the individual sources radiating alone. The acoustic sinks are one of the most distinct expressions of the phenomen, exploited e.g. in the active attenuation of noise sources.

The source itself has internal losses which is against the fundamental conditions for the ability of the method. Because of the losses the radiated power of the source depends also on other incoherent sources. Even the incoherent sources can influence on the source to such a degree that it works as a sink in some frequency bands. For the sake of coherent and incoherent interaction the power of sound source is not a property of the source alone.

Examination based on the statistical energy analysis

Let us define as subsystem one the source volume the power of which we want to measure and as subsystem two all the surroundings of the volume. The net power flowing from subsystem one to subsystem two is according to the balance equations of the statistical energy analysis

$$P = \frac{a_{12}}{a_{11}a_{22} + a_{12}a_{22} + a_{11}a_{12}} \left(a_{22}P_1 - a_{11}\, P_2 \right) , \quad (1)$$

where P_i is the acoustic power fed to subsystem i and

$$a_{ij} = n_i\eta_{ij} = n_j\eta_{ji} = a_{ji} , \quad (2)$$

where n_i is the modal density of subsystem i and η_{ij} is the coupling loss factor of subsystems i and j (η_{ii} is the internal loss factor of subsystem i). It can be thought that the boundary of the subsystems contributes to the balance equations. Its contributions can, however, be included to the parameters of subsystems 1 and 2 involving that no primary powers are affecting on it /1/.

The net power coming out from subsystem 1 is the same as the power fed to subsystem 1 only if subsystem 1 is lossless. This is also true in those special cases where subsystem 2 is free space ($n_2 = \infty$) or no power is fed to subsystem 2 ($P_2 = 0$). As an extreme case when subsystem 2 is lossless the power fed to subsystem 1 doesn't contribute to the net power flow at all. The power fed to subsystem 2 always diminishes the net power flow from subsystem 1 to subsystem 2. The net power flow is negative (flow direction is towards subsystem 1) if the power fed to subsystem 2 is great enough.

In the statistical energy analysis the sources feeding the powers to the subsystems are assumed to be mutually incoherent. So the equation (1) can be interpreted as follows: The power of the source we are measuring depends on the powers of the incoherent sources situating in the surroundings if the source has internal losses. The dependence is the stronger the smaller the modal density of the surroundings is compared with the modal density of the source and the smaller the losses of the surroundings are compared with the internal losses of the source (the power flows towards losses). The contribution of the surroundings always diminishes the net power coming from the source. If the power originating from the surroundings is great enough, the net power flows towards the source.

Examination based on field equations

The (complex) acoustic power radiated from the volume V of a fluid is

$$P = \oint_A \vec{I}\cdot\vec{n}\; dA = \int_V \nabla\cdot\vec{I}\; dV = \int_V \nabla p\cdot\vec{u}^{*}\; dV + \int_V p\nabla\cdot\vec{u}^{*}\; dV \ , (3)$$

where A is the surface enclosing the volume V, $\vec{n}$ its unit normal vector (outwards) and $\vec{I}$ the complex intensity. Let us suppose that the medium is a flowless and homogenous Newtonian fluid. From the linearized equations of the acoustic fields /2/ one can derive the formulae for ∇p and $\nabla \cdot \vec{u}$. Assuming the fluid to have small losses, i.e. k_0, μ, μ_v and μ' are small, where μ is the coefficient of viscosity of the fluid, μ_v is the expansion coefficient of viscosity, k_0 is the thermal conductivity of the fluid and

$$\mu' = \mu \left(1 + \frac{3}{4} \frac{\mu_v}{\mu} \right) , \tag{4}$$

the formulae for ∇p and $\nabla \cdot \vec{u}^*$ can be manipulated to yield according to equation (3)

$$P = P_1 + P_2 + P_3 + P_4 + P_5 , \tag{5}$$

where

$$P_1 = \int_V \left(\rho_0 \vec{f}_1 \cdot \vec{u}_1^* + p q^* \right) dV$$

$$P_2 = j\omega \int_V \left(\frac{1}{\rho_0 c_0^2} |p|^2 - \rho_0 |\vec{u}_1|^2 \right) dV$$

$$P_3 = - \left(\frac{\omega}{c_0}\right)^2 \int_V \left(\frac{4}{3} \mu' |\vec{u}_1|^2 + k_0 \frac{\gamma - 1}{\rho_0^2 c_0^2 c_p} |p|^2 \right) dV$$

$$P_4 = \int_V \left(k_0 \frac{\gamma - 1}{\rho_0 c_0^2 c_p} p \left(\nabla \cdot \vec{f}_1^* + j\omega q^* \right) + \frac{4}{3} \mu' \left(\nabla q - j \frac{\omega}{c_0^2} \vec{f}_1 \right) \cdot \vec{u}_1^* \right) dV \tag{6}$$

$$P_5 = \int_V \left(\left(\vec{f}_1 - j\omega \vec{u}_1 \right) \left(\rho_0 - j \frac{4}{3} \mu' \frac{\omega}{c_0^2} \right) \cdot \vec{u}_2^* + \frac{4}{3} \mu' \nabla q \cdot \vec{u}_2^* \right) dV .$$

where ρ_0 is the density, c_0 the speed of sound, c_p the specific heat at constant pressure, γ the adiabatic constant, q the monopole source distribution (1/s, contains the mass sources) and $\vec{f}$ the dipole source distribution (N/kg, contains the body force sources). The effect of thermal and stress sources is not included. The particle velocity and the dipole source distribution has been divided into irrotational (subscript 1) and solenoidal (subscript 2) component. The solenoidal fieds do not contribute to the sound pressure but they affect, however, to the acoustic intensity.

The term P_2 is purely imaginary, so it does not affect to the resistive acoustic power. The net resistive power radiated by the source volume is got from the real parts of the other terms. In the terms P_1, P_4 and P_5 there are affecting besides the source distributions also the sound pressure and the particle velocity in the source volume. Those fields are affected, besides by the source distributions inside the volume, also by the sources everywhere else outside the volume. It can be seen that only the other sources coherent with the sources inside the volume can contribute to the terms. The interaction of coherent sources, the action of acoustic sinks and active attenuation are based on these terms. The term P_4 tells us about the effects of internal losses of the source volume and the term P_5 of the solenoidal fields to the power of the source.

The term P_3 is purely real and negative. It is non-zero only if the source volume has internal losses. So it tells us about the diminishing effect of the losses of the source to the power radiation. It can be seen that besides the coherent other sources also the incoherent sources outside the source volume contribute to it (inside the integral the absolute values of the field quantities as factors). The interaction of the incoherent sources diminishes the power of the source we are concerning. The existence of that kind of incoherent interaction has been observed in preliminary measurements. The effects of the phenomen can be seen in the dependence of the "in situ" power measurements on surroundings and other sources and also in the reliability of the mutual comparison of the powers of sources measured in different conditions.

The properties of the surroundings (reflecting surfaces e.g.) contribute to all of the terms $P_1 \ldots P_5$ through the field quantities. The rotationality of intensity fields is attached to the coherent interaction of sources and parts of sources /3/, so it affects in the terms P_1, P_2, P_4 and P_5. The interaction of incoherent sources that can be seen in the term P_3 can also produce rotationality in the intensity fields because the sources inside the volume can act as passive sinks for the other incoherent sources.

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ACOUSTIC INTENSITY AND SPATIAL TRANSFORMATION USED TO DESCRIBE THE SOUND FIELD AROUND A SEISMIC VIBRATOR.

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1. INTRODUCTION

During onshore seismic surveys using the vibroseis system it is often recognized that the airborne sound radiated from the vibrator is creating undesired signals in the measuring system. However, before any successful noise reduction can be performed, it is necessary to determine the contribution from the various parts of the vibrator as related to the total airborne noise radiation.

This noise source identification can be carried out in different ways. An earlier but still widely used method is the so-called lead wrapping technique. According to this method the noise source, in this case the vibrator, is acoustically enclosed. Then the various partial noise sources are exposed successively and the changes in radiated noise is measured. This method, however, is quite tedious. Further, it requires an effective acoustic enclosure which can be difficult to establish, especially for low-frequency noise sources as the seismic vibrator.

Consequently the recently developed acoustic intensity technique was employed for the noise source identification. Furthermore, another two-channel measuring technique called Spatial Transformation of Sound Fields (STSF) was used.

2. DESCRIPTION OF VIBRATOR

In the vibroseis system a very powerful vibrator is used to create ground vibrations. The vibrations are then detected far away from the vibrator by means of an array of geophones. By means of a rather complicated signal processing technique, it is then possible to estimate the structure of the underground with respect to oil and gas deposits.

A typical seismic vibrator is mounted on a heavy truck. A diesel engine is driving a hydraulic pump which, through a controlled servo valve, is operating a hydraulic piston system. The piston is connected to the baseplate which transfers the vibration energy into the ground. During operation the baseplate is lowered to the ground, carrying the weight of the truck. This investigation has only considered vibrators with vertical displacements. The vibration signal is sinusoidal with varying frequency (sweep), in the frequency range 6 Hz - 250 Hz. The sweep time used was 30 sec.

3. MEASUREMENT TECHNIQUE

As already mentioned two recently developed measuring methods were employed, i.e. acoustic intensity and spatial transformation, both methods using two-channel measurements. The principle of the acoustic intensity method is based on the use of two microphones placed with narrow space, connected to a two-channel analyzer. The spatial transformation method is employing the measurement of the cross-correlation between a microphone and a reference signal.

3.1 Acoustic Intensity Measurements

The acoustic intensity measurements were performed by means of a two-microphone intensity probe employing a microphone distance of 200 mm, giving a useful frequency range of 25 Hz - 500 Hz. The probe was connected to a B&K type 3360 intensity analyzer. During the measurements the probe was situated in 24 points, located in a plane 1 metre to the left of the truck. In each point the probe was orientated in three orthogonal directions, thus making it possible to determine the acoustic intensity vector in each point. Each measurement consisted of a 32 sec. linear integration, covering one sweep.

3.2 Spatial Transformation Measurements

One of the problems associated with the acoustic intensity measurements was extraeneous noise originating from the diesel engine and the hydraulic pump as these sources radiated significant noise within the frequency range of the sweep.

As the extraeneous noise was uncorrelated with the sweep noise, it was decided to make use of correlation technique in order to suppress the extraeneous noise in the measurement results. The technique used is known as Spatial Transformation of Sound Fields. This method is a combination of Nearfield Acoustic Holography and Helmholtz' integral equation and is further described in ref. 1. In principle the cross spectrum between a microphone signal and the electrical sweep signal (pilot signal) was measured. The microphone was successively situated in 55 points located in a plane 1 metre to the left of the truck. The measurements were performed with a modified version of a B&K type 3360 intensity analyzer, and a 32 sec. linear sweep was carried out in each measurement point.

4. MEASUREMENT RESULTS

4.1 Acoustic Intensity

The noise radiation from the vibrator is represented by the projection of the acoustic intensity vector in the x-y plane, i.e. by an arrow which originates from the measurement point. The length of the arrow indicates the numerical value of the projection of the vector and the direction of the arrow shows the direction of the acoustic intensity flow.

An example is seen in Figure 1, showing the vector projected in the x-y plane for the one-third octave with centre frequency 80 Hz. Figure 1 clearly indicates that the noise is primarily radiated from the vibrating baseplate. Further, some radiation from the intake and exhaust openings of the diesel engine can be seen. This noise contribution is, however, uncorrelated with the vibrator sweep.

4.2 Spatial Transformation

Using nearfield acoustic holography it is possible to calculate the acoustic field in a plane closer to the vibrator than the measuring plane. One result of the calculated sound field 1.5 m closer to the vibrator than the measuring plane is shown in Figure 2, and is given as equal contour diagrams of the acoustic intensity in the z direction, i.e. perpendicular to the measuring plane. Figure 2 shows in accordance with Figure 1 that the major airborne noise is radiated from the baseplate in the 80 Hz one-third octave frequency band.

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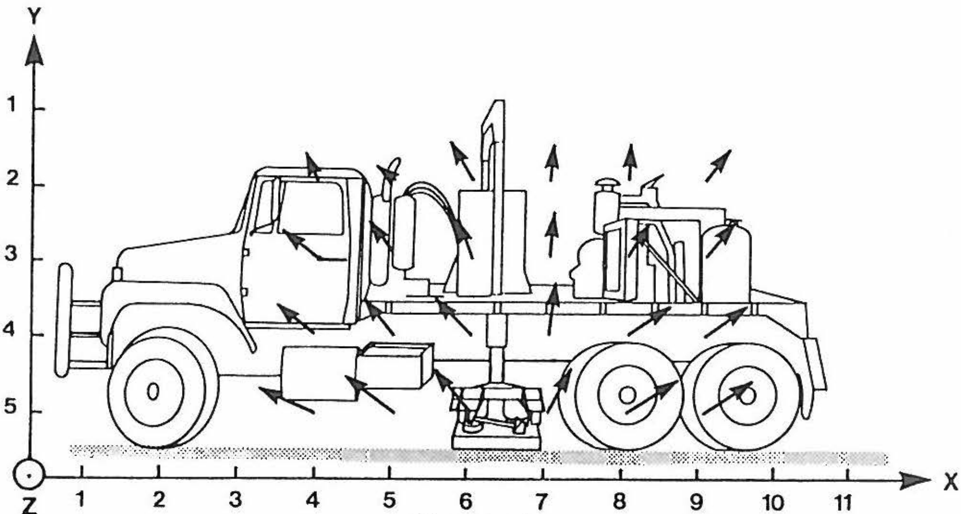

Figure 1.
Acoustic intensity vector in the x-y plane.
One-third octave band centre frequency: 80 Hz.
Reference level: 80 dB re. 1 pW/m^2.
Scale: ⟼ = 20 dB.

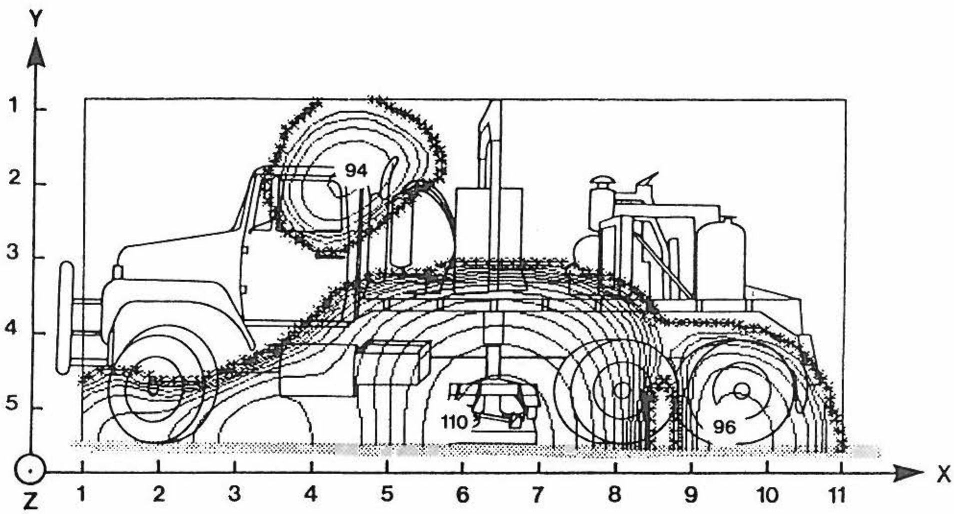

Figure 2.
Acoustic intensity radiated in the z direction.
Equal contour diagram for a plane 1.5 m closer to the vibrator than the measuring plane.
One-third octave centre frequency: 80 Hz.
Reference level: 84 dB re. 1 pW/m^2.
Distance between contour lines: 2 dB.

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UNDERWATER NOISE FROM SEISMIC VESSELS
DETERMINATION OF SOURCE STRENGTH OF MACHINERY

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INTRODUCTION

In connection with marine seismic prospecting, the quality of the reflected signal strongly depends on the level of the background noise on the hydrophone array. Often the level of the background noise on a hydrophone section is mainly determined by noise contributions radiated from the machinery and the propeller of the vessel towing the streamer, see Figure 1. Usually a seismic contractor has to meet certain criteria for the maximum noise level acceptable on the hydrophone sections, e.g. 2-5 μbar.

It is observed that the level of the ship noise varies very much from one location to another, depending on the water depth and the sea bottom properties. The highest levels are generally observed at shallow waters with a strong reflecting sea bottom. A seismic hydrophone section consists of several parallel coupled hydrophones. Therefore also the frequency of the noise and the length of the hydrophone sections are important for the observed level of the noise originating from the ship. Especially short sections with very little directivity at low frequencies e.g. 6-10 metres length, are sensitive to noise radiated from the ship.

MEASUREMENT OF NOISE LEVELS

The procedure that at first seems most appropriate for determination of the noise level of a seismic ship is to actually measure the noise by means of the ship's own towed hydrophone streamer.

However, as mentioned above the level of the noise will depend heavily on the measuring location, e.g. the water depth, the sound velocity profile, and the sea bottom properties. The result is that the data obtained by such a measurement cannot be applied directly to describe the noise properties of a ship. Another problem is that a normal seismic section consists of an array of hydrophones which has a strong directivity depending on the frequency.

A method that is better suited for measuring the underwater noise from a seismic ship is to use omnidirectional calibrated hydrophones for recording of the received ship noise as function of distance to the ship. The hydrophones may be submerged into the water from an assisting boat. Simultaneously with the noise recording, the distance to the seismic ship is recorded.

The properties of the sound transmissions in the measurement area are measured together with the noise. This can be done by detonating small calibrated explosive charges at various distances to the measuring hydrophones. By measuring the received sound pressure on the hydrophones, the sound transmission loss TL at the measuring site may be determined as function of distance and frequency.

MONOPOLE NOISE SOURCE STRENGTH

The noise sources in a ship are placed very close to the sea surface. Noise from the ship may be considered as originating partly from the ship itself, partly from an image source radiating a 180 degree out of phase ghost signal, see Figure 2. This means that the noise radiation from a source on the ship may be considered as being radiation from a dipole source. As is well-known, a dipole source has a strong directivity, which depends on the frequency and the distance between the real source and the image source, i.e. in this case the depth location of the noise source. Very little noise radiation occurs from shallow sources close to the surface and in directions with small angles to the horizontal plane. Thus, most of the radiation occurs in the directions towards the sea bottom. This is why bottom reflected or refracted noise is much more important than direct transmitted noise.

Although a dipole source strength is range independent it is not very practical for characterization of the noise source strength. A much more useful quantity is the equivalent monopole source strength L_S, expressed as the equivalent sound pressure level at a reference distance of 1 metre.

L_S is very useful for characterization or rating of a seismic vessel or its single noise sources with regard to radiated noise. The source strength is characteristic for the vessel or the noise source in question and does not depend on the measurement site.

The source strength level may be determined from:

$$L_s = L_p + TL \qquad (1)$$

where,

L_p is the sound pressure level measured at a certain distance with an omnidirectional hydrophone and TL is the corresponding transmission loss at the measurement location, e.g. measured by means of small TNT explosives as mentioned above. By placing the TNT explosives at the same depth as the noise source on the ship the measured TL includes the directivity correction caused by the sea surface.

PREDICTION OF NOISE ON A HYDROPHONE ARRAY

The equivalent monopole source strength L_s may be used for prediction of the expected background ship noise L_{arr} on a hydrophone array at a specific prospecting area L_{arr} may be calculated from

$$L_{arr} = L_s - TL + DI \qquad (2)$$

where DI is the array directivity in dB, see Figure 4. TL is the transmission loss from the actual ship source, placed at a certain depth below the sea surface, to each hydrophone section in the towed streamer. The transmission loss may be calculated theoretically by means of a full wave equation model, e.g. a fast field model. An example of the result from such a calculation is shown in Figure 3 as function of frequency.

The predicted noise level may be applied for evaluation of the expected S/N ratio and for determination of the most appropriate seismic source and setting of array parameters.

CONCLUSION

The noise level from a seismic vessel should be characterized as an equivalent monopole source strength. This quantity is characteristic for a specific ship and is independent of the measurement site and the hydrophone cable geometry. The monopole source strength may be applied for prediction of the expected background noise from the ship at a specific prospecting area if the transmission loss from the ship to the hydrophone cable can be predicted, e.g. by means of a FFP full wave equation model.

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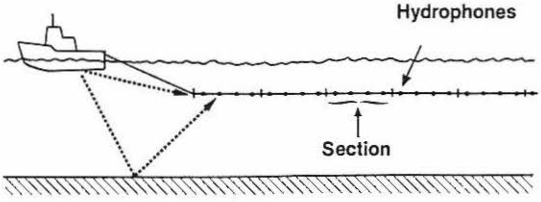

Figure 1.
Sound is radiated from the ship hull and the propeller of the ship and is transmitted via the sea bottom to the hydrophone arrays.

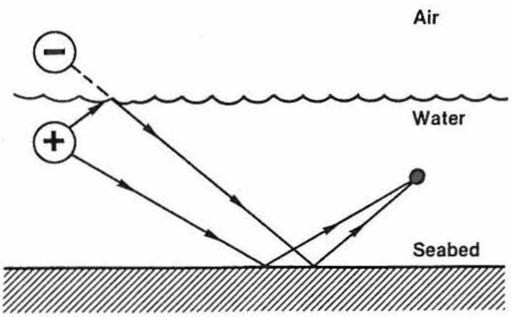

Figure 2.
The radiation of noise from a source on a seismic vessel is equivalent to radiation from a dipole source because of reflections from the sea surface.

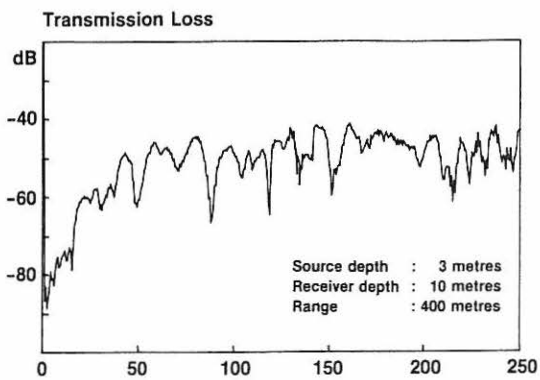

Figure 3.
Transmission loss as function of frequency at a range of 400 metres. Calculated by means of a FFP-model. Bottom properties: compression sound velocity 2250 metres, shear sound velocity 600 m/sec.

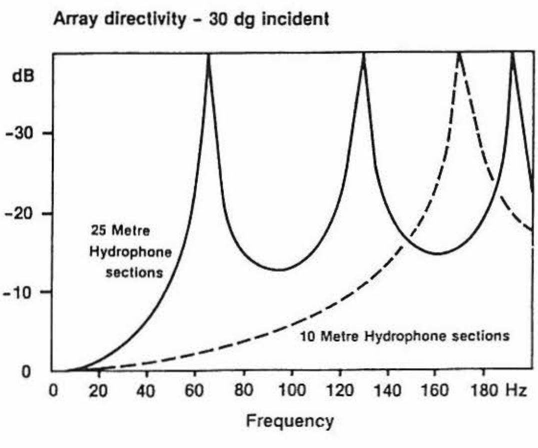

Figure 4.
Frequency response of a 10 metres and a 25 metres hydrophone section. Angle of incidence 30 degrees to the cable axis.

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THE STAGE FLOOR - SUPPORTING RESONANT BODY OR SOUND TRAP FOR THE DOUBLE BASS?

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Introduction

The cello and the double bass are supported on the floor via an adjustable metal pin, the end pin or the peg. This arrangement may have acoustic implications. One hypothesis is that part of the body vibrations of these instruments could be transmitted down through the end pin, setting the stage floor or riser into vibration. The vibrating stage floor would then act as an enlargement of the instrument body, and contribute to the radiated sound.

Informal questioning reveals that the acoustic support from stage floor and risers as perceived by the musicians can be considerable, provided the vibrational properties of the floor are favorable. The literature on architectural acoustics also gives evidence of attention being paid to floor vibrations in concert halls. The Neues Gewandhaus in Leipzig (built in 1896, destroyed in the last war) was famous for the "powerful" sound of the cello and double bass sections (Bagenal & Wood 1931, Beranek 1962), an effect which was assumed to be due to vibrations in the thin wooden parquet floor on which the entire stage was resting.

Experiments

A series of acoustic measurements was carried out in the Berwald Hall in Stockholm, in order to determine the practical influence of stage risers. This hall was particularly well suited for the experiments. The stage floor is constructed from muninga hardwood, bonded in asphalt to a concrete foundation resting on bedrock. This construction yields a very stiff stage floor, almost totally resistent to vibrations, which could be used as a reference case.

Two professional double bass players were asked to perform, first with the instruments placed directly on the stage floor, in a second session with the instruments supported on risers, and finally on the stage floor again. Two types of risers were used in the experiments, one smaller size with the dimensions 1 x 1 m approximately, and a larger size with the dimensions 1 x 2 m, with a cross-bar at the middle of the long side. Both types were covered with 1/2" plywood. The players were placed stage right at their usual position in the orchestra. The recordings were made with several microphones in different positions. The results presented in the following refer to a near-field microphone very close to the players (about 1 m), and a far-field microphone above the audience, at a distance of about 12 m. The reverberation radius of the hall is approximately 3.5 m.

The players performed chromatic scales with a compass from the low E-string to the open G-string, covering a fundamental frequency range from 41 Hz to 98 Hz. The instruction to the players was to play all scales identically with regard to dynamic level, tempo, bow velocity, bow force and distance between bow and bridge. The players chose to play at a mezzoforte level.

The recordings were analyzed by computing the Long Time Average Spectra (LTAS), in which the audible frequency range (20 - 20 000 Hz) is covered by 30 one-third octave bands.

Results

The acoustic influence of the risers was determined by comparing LTAS from performances made on the floor and on the two types of stage risers used, Fig. 1. Each curve represents the average of several scales, corresponding to between four and two minutes of analyzed sound. The figure shows that playing on the risers gives a gain in a considerable part of the frequency range. The gain is different for the two types of risers used. The largest differences

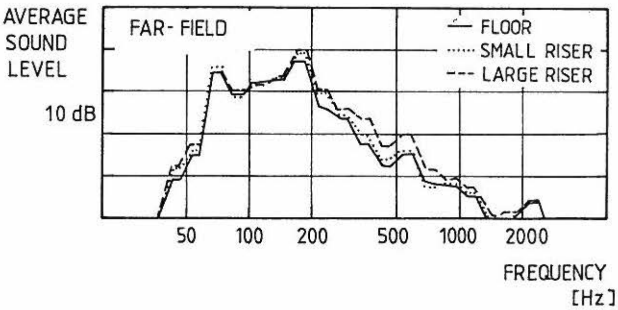

Fig. 1: Long Time Average Spectra (LTAS) of chromatic scales played on two double basses in a concert hall, as recorded by a microphone in the far-field. The basses were positioned on the stiff stage floor (full lines), small risers (dotted lines) and large risers (dashed lines). Each curve represents an average of several repeats, amounting to a couple of minutes of analyzed sound.

are observed with the large riser which gives a maximum gain in the order of 3 dB around 50 Hz, as measured in the far field. In the range 150 - 1000 Hz the observed gain is even higher, about 5 dB at the most. These differences in sound radiation with the instruments positioned on the floor and on risers respectively, are not negliable and can be assumed to have perceptible influence on the sound in the hall.

The question arises whether the observed differences are due to the risers, or if they rather reflect differences between the performances on the floor and risers, respectively. In fact, the musicians were skilled at making consistent repeats. The areas in Fig. 2 represent the variation in LTAS between several repeats on the floor and on the large riser. The figure shows that the rather small variations between repeats do not make the LTAS from the three playing conditions merge to the extent that they would render the observed differences bewteen the floor and risers meaningless. Consequently, the players could be considered capable of making repeats under the same conditions with satisfactory reproducability.

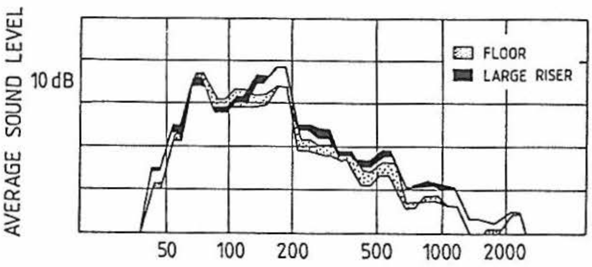

Fig. 2: Variation in LTAS between repeats (near-field microphone). The variation between seven repeats on the floor (dotted area) is compared with four repeats in two positions on the large riser (filled area).

Exchange of vibrations

Not all of the vibration energy in the floor will be radiated as sound. Apart from losses during the transmission in the floor, adjacent basses and celli in the orchestra could be assumed to consume energy from the floor by picking up vibrations through their end pins. In addition to this indirect transmission path via the floor, the instruments will also exchange vibrations directly via the sound transmission in the air.

The efficiency in this exchange of energy between instruments was investigated by comparing the vibrations in the bridge and in the support close to the end pin for two conditions, both on a riser. In the "active" condition the bass was played. The vibration levels (acceleration) in the bridge and in the riser close to the end pin of this "active" bass are compared in Fig. 3. The difference in vibration levels between bridge and riser is roughly 30 dB. As a comparison, the corresponding difference when the bass was supported directly on the floor was more than 50 dB. This confirms the initial assumption that the stage floor itself is almost rigid.

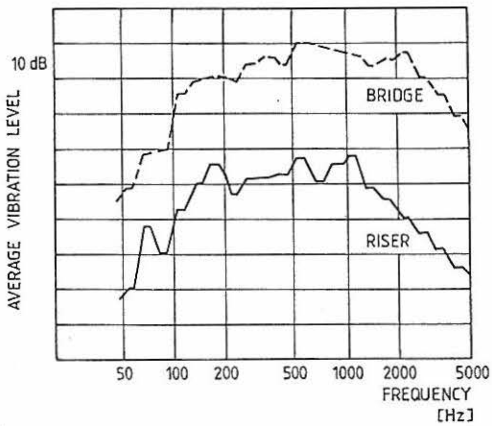

Fig. 3: The vibrations of a played double bass are partly transmitted down into the support via the end pin. The figure shows LTAS of the vibration (acceleration) in the bridge and in the support of an played ("active") bass, when positioned on a riser.

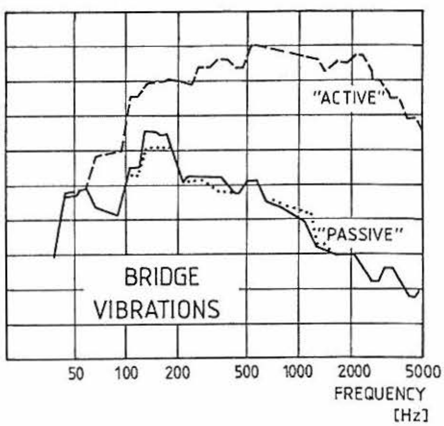

Fig. 4: The vibrations in the support together with the near-field sound will set adjacent basses into vibration. The figure illustrates the difference in the bridge vibrations when a bass is played on a riser ("active"), and when exposed to the excitation from another bass on the same riser ("passive").

In the "passive" condition, the same bass was placed on the same riser but only left exposed to the excitation from another bass, which was played on the same riser at a distance of about 1.5 m. The "passive" bass was now set into vibration by the excitation from the played instrument, Fig. 4. The figure shows the vibration level in the bridge of the "passive" bass compared to the corresponding level when it was "active". The difference is large at higher frequencies, more than 30 dB above 200 Hz. At lower frequencies, however, the transmission between the basses grows more efficient, and at approximately 60 Hz the bridge vibrations are of equal amplitude whether the bass is "active" or "passive". The high vibration level in the "passive" bass at low frequences is probably due to the Helmholtz resonance (approximately 65 Hz), which is easily excited externally.

The proportion between vibrations transmitted through the air and via the end pins was estimated by supporting the passive bass on a piece of foam rubber, with the aim of preventing the vibrations in the riser to reach the "passive" bass, The insertion of the foam rubber support did not notably change the vibration level in the passive bass, see Fig. 4. This result indicates that the air path is the dominating part in the coupling between the instruments.

It is an open question, whether the mutual exchange of vibrations within a double bass section, be it placed on risers or not, are important to the ensemble playing, for instance as regards intonation.

Body size and radiated partials

The conditions for sound radiation in the low register of the bowed instruments are akward. The bodies of the instruments are much too small in comparison with the wavelength of the low fundamentals in order to be efficient sound radiators. This is especially true for the double bass and cello, which ought to have their body lengths increased by almost 50 % in order to compete with the violin. As a consequence of the "small" body size, the radiated spectra for the lowest notes on all bowed instruments exhibit a very weak fundamental, typically 20 dB weaker than the second harmonic.

A strong fundamental is associated with a perceptually full tone quality. A wide open organ pipe (flute rank) is an example of a sound source in which the fundamental always is stronger han the other partials. When used in the low register this rank gives a perceptually full bass sound. Accordingly, a reinforcement of the lowest harmonics in the cello and the double bass by stage floor vibrations would promote a full sound quality.

Stage risers can have such an effect on the radiated sound of the double bass, as measured in spectra of low notes. The fundamental may gain as much as 10 dB on certain notes, and dips in the spectra at higher partials are leveled, when the instrument is played on stage risers.

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From Wooden Blank to Finished Violin Top Plate.

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1. Introduction.

The spruce blank for a violin top plate has well defined lower resonances. The resonances have been mapped by means of Vibravision (an electronic speckle pattern interferometer using a laser and presenting the vibration patterns on a TV-monitor (1). Furthermore the resonances have been proved to be related to the elastical properites of the wood in a simple way. After correction for mass the frequency of the first resonance gives the skew modulus, the frequency of the second resonance gives the longitudinal elasticity modulus and the frequency of the third resonance gives the transversal elasticity modulus (2). With the vibration patterns known the violin maker can obtain these resonance frequencies by simple tap tone test in his workshop and thus the elastical properties of his working material (1).
The properties of the free violin plate can be judged by the properties of its first, second and fifth resonances (3). The three resonances can also be used as guide to work the plate to wanted properties. Therefore we have studied relations between resonances (frequencies and vibration patterns) and plate properties (material properties and design) by FEM-calculations (4). The calculated results are compared with a violin plate made with known and the same material parameters.

2. Starting point - measured and calculated properties of the blank.

The blank had a "standard" size: 385 mm long, 215 mm wide, 7.5 mm thick along the edges and 20 mm along the center line (the glue joint). One side of the blank is flat and the other side as a roof with two tilting sides. Calculated and measured nodal lines show good agreement fig. 1. Calculated resonance frequencies showed minor differences (less than 2%). The material parameters obtained and used as reference in later calculations were a density of 482 kg/m^3, and 16 Gpa, 0.975 Gpa and 0.725 Gpa for the longitudinal elasticity modulus, the transversal elasticity modulus and the shear modulus respectively.

3. Calculated and measured properties of the free top plate.

A FE-model, symmetrical along the glue joint (and resembling of a Stradivarius top plate) was used in the calculations, see fig. 2. The model consisted of 190 triangular three-node flat elements. FEM-calculations were made for different thicknesses and thickness distributions, different arch height, and variations in single material parameters. Furthermore a physical top plate was made of the blank with geometrical measures and material parameters of one of the FE-models. Calculated and measured resonance frequencies and nodal lines are given in fig 3.
The calculated and the measured frequencies showed good agreement (maximum differences 8% for the third and fourth modes, 5% for the fifth mode and 2% for modes one and two. The calculated and measured nodal lines show good agreement too. Thus we concluded that the results of the numerical experiments were to be trusted. We investigated modes no 1, 2 and 5 more in detail and we thereby found:
The importance of the different factors can be ranked as thickness, density, arch height and elastical modulii (when each factor is changed 10%).
Mode no 1 is most sensitive to the thickness in the center and the shear modulus,
mode no 2 is most sensitive to the thickness in the center, the transversal and the longitudinal elasticity modulii, and
mode no 5 is most sensitive to the thickness in the upper and lower parts, the longitudinal elasticity modulus, the shear modulus and the transversal elasticity modulus (the difference in magnitude of influence is small for elastical parameters and mode no 5 though).
The thickness influences modes no 1 and 2 more than mode no 5, and the arch height seems to influence mode no 5 more than modes no 1 and 2 (the influence of the different parameters are approximately the same though).

4. Conclusions

One aim of this report was to investigate whether "deviating" material properties can be "corrected" for by adjusting thickness and arch height, and how this correction could be achieved.
The analysis of the calculated (and partially verified by experiments) have given:
If all three elastical parameters were increased by the same amount, their influence could be counteracted by thinning the plate uniformly or by decreasing the arch height. If, however, only one of the parameters were changed, this effected the modes differently. Therefore we believe that single parameter deviations are best dealt with by thinning the plate unevenly. For instance, mode no 1 will increase in frequency with an increase of the shear modulus. This increase is counteracted by thinning the plate in the center. The thinning has, however, also the side effect to decrease the frequency of mode no 2, which is little influenced by the shear modulus. This type of complication seems to be general - a major correction must be followed by minor corrections for side effects.
We also made a pilot study on the influence of the boundary condtions. Comparisons of experiments and calculations indicated that the plate should be treated as approximately clamped. It was, however, evident that a good understanding of the boundary conditions is most important. The different locked boundary conditions gave larger influence than those observed for variations in material properties.

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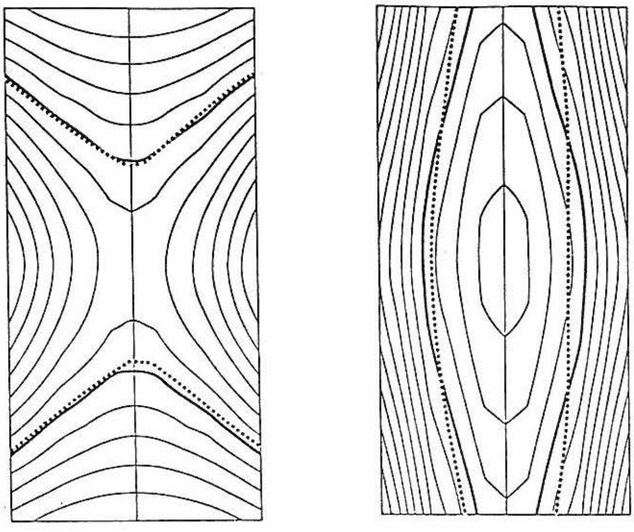


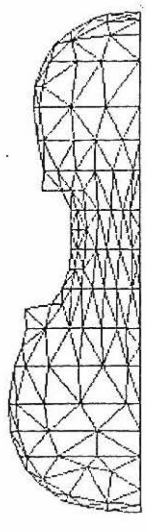
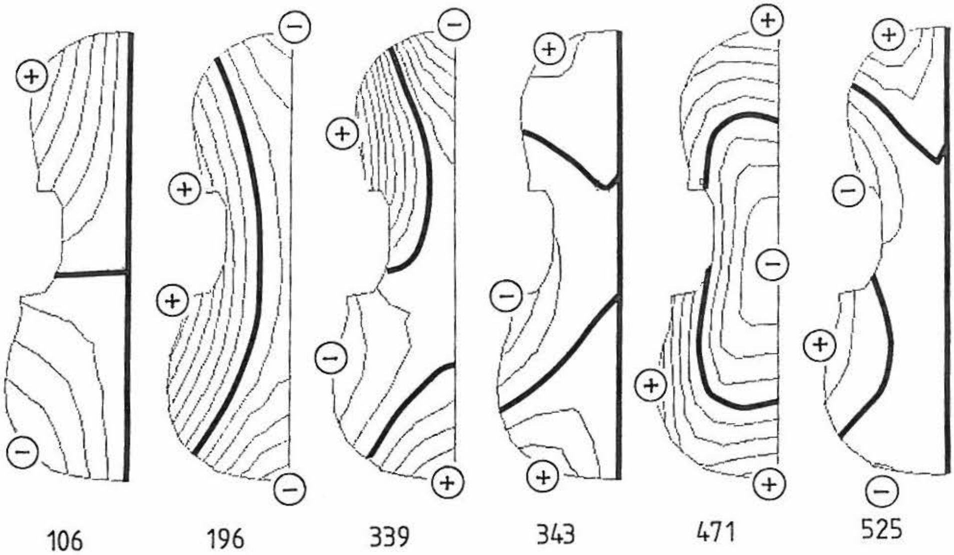


Fig. 1. Calculated iso-amplitude lines and measured nodal lines for the spruce blank - calculated/measured frequencies 272/272 (not shown), 573/562, and 637/631 Hz, iso-amplitude lines and glue joint are marked with thin lines, calculated nodal lines with thick lines and measured nodal lines with broken lines.

Fig. 2. FE-model of one half of the violin plate.



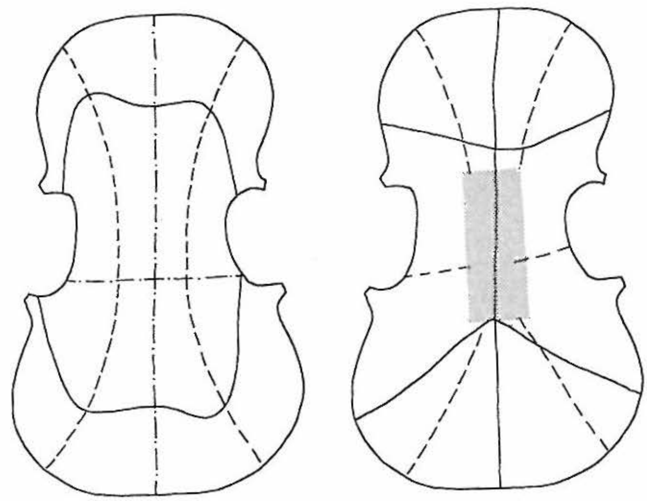


Fig. 3a. FEM-calculated modal shapes - numbers mark frequencies in Hz, thin lines mark iso-amplitude lines, thick lines mark nodal lines, and plus and minus signs mark antinodes of opposite vibration phases.

Fig. 3b. Measured nodal lines - (left figure) mode no 1 at 105 Hz with dotted line, no 2 at 200 Hz with broken line and no 5 at 448 Hz with full line, and (right figure) mode no 3 at 332 Hz with broken line and no 4 at 317 Hz with full line.

AUTHOR INDEX

This book includes 95 papers presented during the 16th meeting of the Nordic Acoustical Society at Aalborg University on August 20-22 1986.

The contributions cover a wide range within the field of acoustics: building acoustics, electroacoustics, speech analysis and synthesis, room acoustics, psychoacoustics, noise abatement, vibrations, instrumentation and legislation.

Most of the contributions are of interest also to people outside the Scandinavian countries, and the Organizing Committee has suggested the authors to give the written version of their paper in English. A large number of contributors have followed this request.

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