PROCEEDINGS OF THE CONFERENCE ON NOISE SUPPRESSION NOISE CONTROL 85

FOREWORD

Several days after the INTER NOISE 85 Conference the International Conference on Noise Suppression – NOISE CONTROL 85 was held from the 24-th to the 27-th of September 1985 in Cracow. It was organized by the Committee on Acoustics of the Polish Academy of Sciences, Commission of Acoustics of the Hungarian Academy of Sciences, Institute of Mechanics and Vibroacoustics of the Academy of Mining and Metallurgy in cooperation with the Institute of Structural Engineering, the Central Institute of Work Protection and the Noise Control League.

The NOISE CONTROL 85 conference was organized parallely with the XXXII Open Seminar on Acoustics.

The present NOISE CONTROL 85 conference is the seventh conference organized in Poland every three years from the inspiration of Prof. Stefan Czarnecki. Initially these conferences were held for solely domestic participants. Begining from 1976 also specialists from other countries take part in this conference. In the framework of these conferences not only problems of existing threats of noise and vibrations in the work and life environment were discussed, but also scientific and technical achievements of various scientific and industrial centres, Polish theoretical works constituting a foundation for further research and original Polish developments, in many cases practically implemented. A Scientific Committee headed by the senior of Polish acousticians – Prof. Ignacy Malecki was engaged with the scientific problems of the conference. Prof. Tomás Tarnóczy from the Hungarian Academy of Sciences was the vice chairman of the Scientific Committee.

Both conferences were attended by 528 persons, 382 at the NOISE CONTROL conference and 146 at the XXXII OSA, including 96 participants from abroad, from 23 countries of the whole world, namely from: Australia, Austria, Belgium, Bulgaria, China, Czechoslovakia, Denmark, Egypt, France, Greece, Holland, Japan, Yugoslavia, Canada, Cuba, GDR, FRG, Roumenia, USA, Sweden, Hungary, Italy, USSR. 158 papers was presented in the framework of the NOISE CONTROL conference – 9 plenar and 11 on poster sessions.

Plenar papers were presented by:

1. I. BALLO, N. SZUTTOR – Institute of Materials and Machine Building of the Slovak Academy of Sciences, "*Active systems in the vibration control of operators seats of heavy-weight machines*".
2. C. CEMPEL – Institute of Applied Mechanics of the Poznań Technical University, "*Tribo-vibro-acoustical model of a machine*".
3. R. CIESIELSKI, J. KAWECKI – Institute of Structure Mechanics of the Cracow Technical University, "*Diagnostic foundations of the national standard PN–85/B–02170*".
4. Z. ENGEL – Institute of Mechanics and Vibroacoustics of the Academy of Mining and Metallurgy in Cracow, "*Progress in noise and vibrations control in Poland*".
5. F. INGERSLEV – President of the International Institute of Noise Control, Technical University, Lyngby, Denmark, "*International cooperation in acoustics*".
6. R. LYON – Massachusetts Institute of Technology, Cambridge, USA, "*Diagnostics and noise control in industry*".

Thesis: Diagnostics is a proper field for noise control engineers.

7. H. MYNCKE – President of the International Commission of Acoustics, Catholic University, Leuven, Belgium, "*Some considerations on common noise criteria*".
8. J. P. NAGY – Technical University of Budapest, "*Sound insulation requirements with respect to room absorption*".

9. J. Sadowski – Institute of Structural Engineering, Warsaw, "*Acoustic problems in multi-storey residential buildings raised by means of industrialized technology methods*".

Besides the plenary papers, the following papers were invited:

1. P. Francois – Electricite de France, Clamart, France, "*Labelling and reduction of the noise level of house-hold appliances*".
2. Z. Jagodziński – Technical University of Gdańsk, "*Ultrasonic Doppler vibrometer*".
3. J. Lang – Versuchsanstalt für Wärme- und Schalltechnik am Technologischem Gewerbemuseum, Vienna, Austria, "*Sound insulation requirements in office buildings and in schools*".
4. A. Lawrence – University of NSW, Sydney, Australia, "NOISE CONTROL – *the architects and planners roles*".
5. Z. Maekawa – Kobe University, Japan, "*Simple estimation methods for noise reduction by various shaped barriers*".
6. V. M. A. Peutz – Nijmegen, Holland, "*Reverberation times and reverberation levels*".
7. E. J. Richards – University of Southampton, England, "*Using the energy accountancy equation to modify punch design in presses*".

Scientific problems with which the Conference dealt, were grouped in 11 sessions, namely:

A. Computer techniques in vibroacoustics,
B. Physical foundations of sound emission and propagation,
C. Machine and equipment noise,
D. Industrial noise,
E. Noises in building objects,
F. Communication noise,
G. Measurement and analysis methods of vibroacoustical signals,
H. Materials in noise and vibrations suppression,
I. Vibrations,
J. Noise and vibration hazards to man,
K. Vibroacoustical diagnostics.

After analysing the programme it has to be stated that the NOISE CONTROL 85 Conference has served its purpose. Many papers, 11 sessions, wide discussions have proved the great activity of Polish acousticians. The Conference made possible the further integration of the Polish Academy of Sciences, universities, institutes and scientific centres, design offices and industrial plants. Several original papers were presented at the Conference, as well as results of research conducted in the framework of various research problems. Basic and theoretical investigations constituted a correct foundation for the utilitarian work. Discussions indicated the necessity of conducting fundamental research, which should be the ground for subsequent technical and implementation works.

All papers were published in a two-volume publication titled: Proceedings NOISE CONTROL 85.

It seems purposeful to conduct subsequent NOISE CONTROL Conferences with the participation of foreign guests every three years.

Z. Engel

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ACTIVE SYSTEMS IN THE VIBRATION CONTROL OF HEAVY MACHINE DRIVER'S SEATS

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The paper deals with vibration damping properties of passive vibration control means used in driver's seats of heavy earth moving machines.

The significant improvement of vibration control reached by an active vibration control system of an electrohydraulic type is described. The advantages and perspectives of electropneumatic active vibration control systems for driver's seats is stressed.

1. Introduction

In heavy working, earth-moving and building machines as well as in tractors, the seat is the main vibration control equipment protecting the driver-operator. Among the favourable consequences of good vibro-isolation for the driver, emphasis is mainly laid upon the protection of his health. On most of the contemporarily used seats, however, acceleration values lie deeply below the limits of the increased health hazard, defined for example by the international standard ISO 2631. Hence the protection of the driver's health should not be a problem today. It should be a matter of course.

Attention is therefore focussed upon satisfying a more severe criterion. The acceleration values on the seat are required to be below the limit of the driver's increased fatigue, as presented by ISO 2631, by standard SEV and by the national health care regulations.

There still exists a third, significant aspect of the driver's good vibro-isolation. It follows from the linkage between the effective value of acceleration on the seat $a_{sef}[\mathrm{ms}^{-2}]$ and the speed of the vehicle in its direct run $v[\mathrm{ms}^{-1}]$, equat. (1) [14]. The man-driver or the man-operator, respectively, makes use of this linkage because he tends to rise the speed of the vehicle under conditions of good vibration control. Conversely, when insufficiently protected, he will lower the speed until effective acceleration value on the seat drops to an acceptable value. This state, however, often means running at a speed that lies deep below the technical capability od the engine

$$a_{\mathrm{sef}} = k\sqrt{v} \tag{1}$$

where k denotes a coefficient, depending on the construction of the vehicle and seat and on the unevenness of the track.

The real driver seats with a passive or semiactive vibroisolating system are of complex structure. From the stand point of mechanics this complexity is of dual nature. Firstly, the seat has several eigenfrequencies and a corresponding set of cubic eigenfunctions. Secondly, there are some elements in the construction of the seat with nonlinear properties. For example: clearances in the kinematic joints, damping devices etc.

Nevertheless, many real seats appear as a system close to a linear one in a limited frequency band.

In estimating the dynamic properties of a seat a passive or semiactive vibration control system, the cabin base motion being stochastically perpendicular, the amplitude characteristic $|F_S|$ Fig. 1 is decisive. Its significance is related to the fact that at a given power spectrum density of the acceleration of floor vibration $\Phi_B[(\mathrm{ms}^{-2})^2/\mathrm{Hz}]$ it determines the power spectrum density of acceleration on

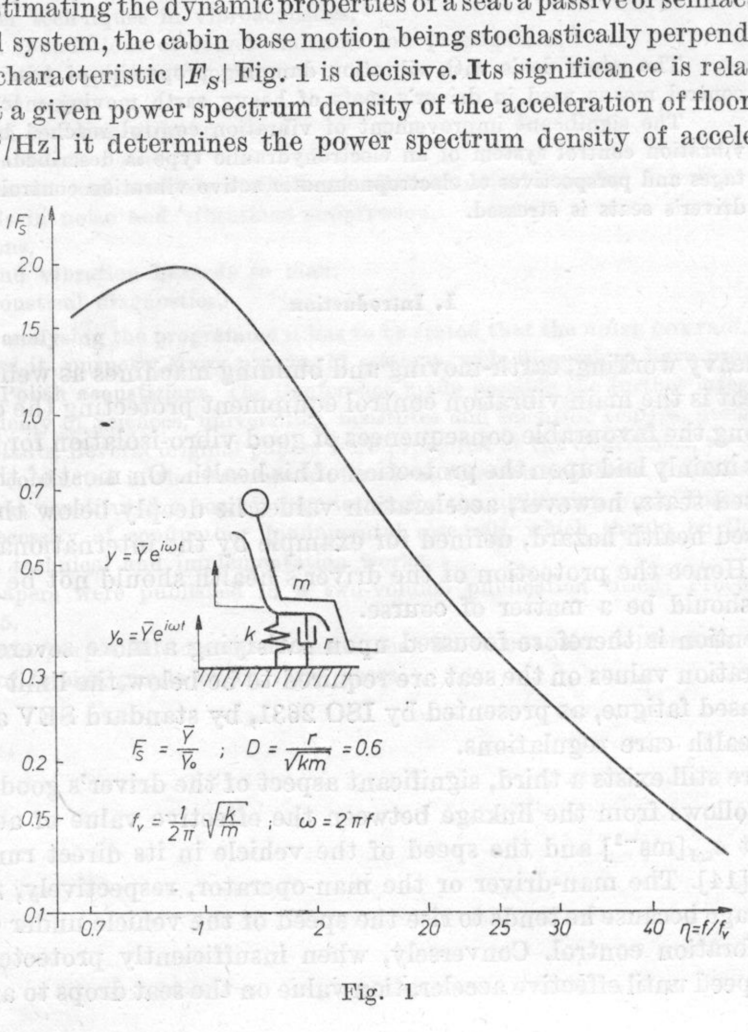

Fig. 1

the seat saddle $\Phi_S\,[(\mathrm{ms}^{-2})^2/\mathrm{Hz}]$, in terms of the relation

$$\Phi_S = |F_S|^2 \Phi_B. \tag{2}$$

Some results calculated according to this equation, are presented in Fig. 2.

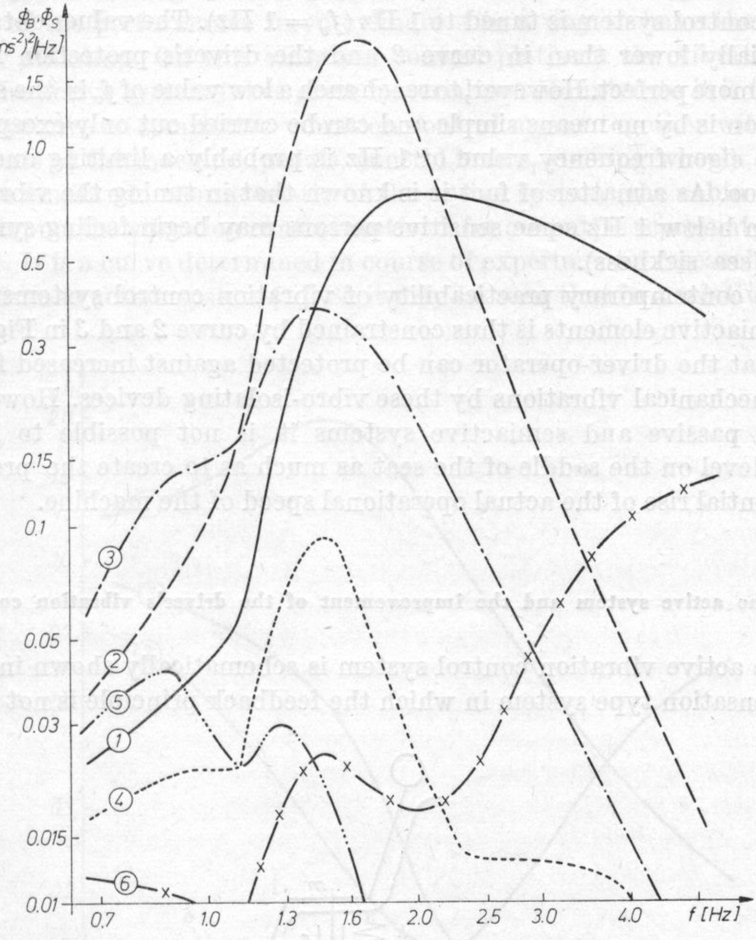

Fig. 2

Curve *1* on this figure shows the typical behaviour of the power spectrum density of perpendicular acceleration on the cabin floor. It is contained in the recommendation of ISO 7092 [12] for the 3rd class of earth movers, i.e. short-base machines with wheel undercarriage.

The power spectrum density of the acceleration of saddle vibration under passive vibration control is represented by curves *2* and *3*.

Curve *1* illustrates the situation when the eigen frequency of the passive vibration control system is $f_v = 1.5$ Hz. Many seats used at present have a vibration control system tuned to that value. They usually satisfy the criteria of the ISO 7096 standard, hence they are considered good.

Curve *2* shows, however, that in the ferquency band up to 2 Hz the power spectrum density of acceleration is higher on the saddle than on the floor, even if the area beneath the curve, that is the effective value of acceleration, may be lower than on curve *1*.

Curve *3* shows the behaviour of the power spectrum density when the vibration control system is tuned to 1 Hz ($f_v = 1$ Hz). The values obtained here are essentially lower than in curve *2* and the driver's protection is also intrinsically more perfect. However, to reach such a low value of f_v in the actual seat construction is by no means simple and can be carried out only exceptionally.

The eigen frequency value of 1 Hz is probably a limiting one on other grounds, too. As a matter of fact it is known that in tuning the vibration control system below 1 Hz some sensitive persons may begin feeling symptoms of kinetosis (sea sickness).

The contemporary practicability of vibration control systems with passive or semiactive elements is thus constrained by curve *2* and *3* in Fig. 2. It can be seen that the driver-operator can be protected against increased fatigue caused by mechanical vibrations by these vibro-isolating devices. However, with the aid of passive and semiactive systems it is not possible to lower the vibration level on the saddle of the seat as much as to create the prerequisites for an essential rise of the actual operational speed of the machine.

2. The active system and the improvement of the driver's vibration control

The active vibration control system is schematically shown in Fig. 3. It is a compensation type system in which the feedback principle is not utilized in

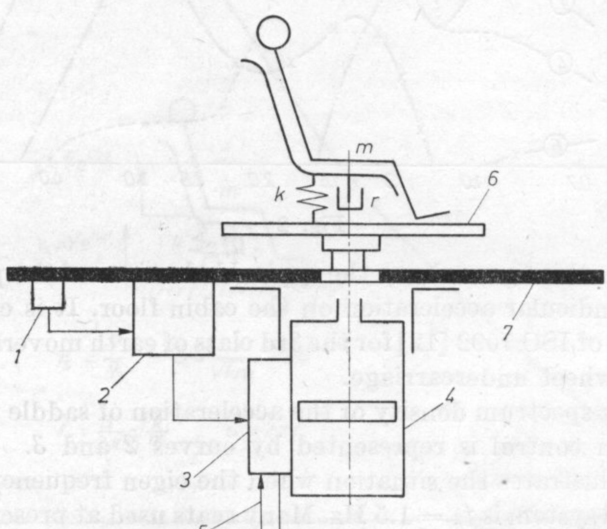

Fig. 3

the frequency band above 0.5 Hz. This improves the noise conditions in the system, removing the problems connected with its stabilization.

To the floor of a working machine *7* an accelerometer is mounted *1*. The signal from this sensor, adapted in the electronic regulator *2* is passed to the transducer *3* controlling the inflow and outflow of the working medium *5* into the organ of the performance servo-cylinder *4* in a way to have the platform *6* practically standing. On this vibration control platform the vibration control effect of the active system takes place materializes. Mounted to it is the driver's seat with passive or semiactive vibration control means together with the control elements of the machine (pedals, control levers, steering wheel).

The amplitude characteristic of the active system (the cabin floor being taken for input, the vibration control platform for output) is represented by curve *2* in Fig. 4. It is a curve determined in course of experimental research into a real electrohydraulic active system [1]. So it may be seen that in a relatively narrow

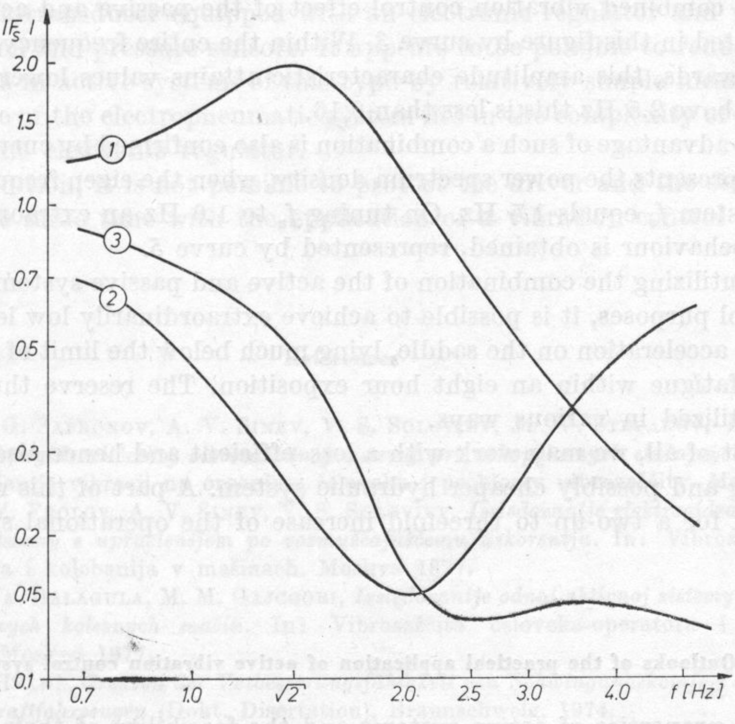

Fig. 4

frequency band around 2 Hz very favourable, low values are attained. However, the amplitude characteristic rises both towards low and high frequencies. Towards the low values this is caused by the real properties of the electronic regulator, towards the high ones by the properties of the hydraulics.

From the viewpoint of vibration control especially the rise of the characteristic towards high frequencies is very unfavourable, because in the frequency band from 4 up to 8 Hz, where the human body's sensitivity to vibration is greatest, the vibration control effect of the active system is considerably limited. This fact is illustrated by curve *6* in Fig. 2. The curve indicates a high power spectrum density value on the seat saddle within the band above 4 Hz.

This unfavourable property of the active system might be restricted by improving the dynamic properties of the hydraulics. At the same time, however, its price would rise and — this being extraordinarily negative effect — its power demands, too.

Another way of improving the properties of an active system is to link it up with a passive vibration control system. Figure 3 indicates a situation, where a seat with a passive vibration control is mounted to the vibration controlled platform. Considering a currently used seat whose eigen frequency, f_v, is about 1.5 Hz, curve *1* in Fig. 4 has been calculated in order to represent its amplitude characteristic.

The combined vibration control effect of the passive and active system is represented in this figure by curve *3*. Within the entire frequency band, from 0.2 Hz upwards, this amplitude characteristic attains values lower than 1. In the band above 2.5 Hz this is less than 0.16.

The advantage of such a combination is also confirmed by curves in Fig. 2. Curve *4* represents the power spectrum density, when the eigen frequency of the passive system f_v equals 1.5 Hz. On tuning f_v to 1.0 Hz an extraordinarily favourable behaviour is obtained, represented by curve *5*.

By utilizing the combination of the active and passive systems for vibration control purposes, it is possible to achieve extraordinarily low levels of perpendicular acceleration on the saddle, lying much below the limit of the driver's increased fatigue within an eight hour exposition. The reserve thus obtained may be utilized in various ways.

First of all, we may work with a less efficient and hence also less power demanding and possibly cheaper hydraulic system. A part of this reserve may be utilized for a two-up to threefold increase of the operational speed of the machine.

3. Outlooks of the practical application of active vibration control systems

The properties of active systems and the feasibility of their application have already been examined by several authors [1–10]. Most of them presumed that the executive element of the active system would be a hydraulic servocylinder completed by an electrohydraulic linear transducer controlling the transflux of the fluid into and from the servocylinder.

The advantage of this conception lies in its relative simplicity, especially of its electronic part. It may be expected that subsequent development will allow the reduction of the system power demand from the contemporary 3 up to 5 kW to hundreds of Watts by order of magnitude.

The disadvantage of the electrohydraulic variant lies in the fact that the active portion of the vibration control system is to be combined with a current seat with a passive vibration control, as it is schematically shown in Fig. 3. This increases both the price and the sophistication of the system. Further, the electrohydraulic transducer is a relatively delicate device requiring, in addition to other measures, also sufficient oil filtration.

According to the survey paper [10], based on analog modelling, some of the flaws of the electrohydraulic system could be removed if a pneumatic executive organ would be used. An especially significant is fact that both active and passive protection can be implemented by a single air spring in which pressure would vary in order to compensate the exciting effects from the cabin floor. This implies that the contemporary design of the driver's seat with an air spring would remain unchanged. The air spring would only be linked to an electropneumatic transducer equipped with an electronic regulator and with a set of acceleration and pressure sensors. It appears to be possible to reduce the power demands in active systems of this type by relatively simple measures. The disadvantage of the electropneumatic system lies in the complexity of the transducer and the electronic regulator.

In addition, it is not possible to protect the driver and the steering elements at the same time with the application of a vibration control system of this kind.

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COMPARISON OF PREDICTION METHODS FOR ROAD TRAFFIC NOISE

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The Sanitary-Epidemiological Station of County Győr-Sopron has accomplished some analysis of road traffic noise. On the basis of these investigations an answer should be given to the question of how far prediction methods for road traffic noise are reliable and to what extent they can be used for planning. The investigations will take 2 years (1984–1985). Research work was sponsored by the National Authority for Environment Protection and Nature Conservation. This report outlines the investigations performed in 1984, and the data published therein may be modified as required according to the 1985 investigations.

1. Method of investigation

In 1984 there were 11 locations in residential areas selected for the purpose of the measurements. At each location there was a basic measuring point at a distance of 7.5 metres from the centre line of the outmost traffic lane of the road. All the other measuring points, 2 to 6 for each location, were arranged inside the residential area. In each case the measurements were done simultaneously by using two instruments; one instrument was used at the basic point, while the other was used at one of the measuring points inside the residential area. The instruments used were BRÜEL et KJAER Noise Level Analyzer, type 4426, and Precision Integrating Sound Level Meter, type 2218.

The equivalent continuous A-weighted sound pressure levels were determined from measurements. The noise measurements were completed with a traffic census. The number of the vehicles was recorded for each traffic lane, and this figure devided according to the categories of vehicles. The duration of each measurement was 10–15 minutes. The equivalent continuous A-weighted sound pressure levels were also predicted theoretically for each investigated measuring point. Five prediction methods were applied.

The results obtained at 7.5 m from the centre line of the outermost lane of the public road and those obtained inside the dwelling area were evaluated separately. The different calculating methods supply the equivalent continuous A-weighted sound pressure levels, calculated from the traffic data, for different distances. All this data has been converted into figures relating to a distance of 7.5 m from the centre line of the outermost lane, so that they could have been compared.

2. Prediction methods used for the evaluation

There were altogether five prediction methods used in the investigation. Three of these methods had been published in Standards or Technical Reports, while two of them had been elaborated by the authors. The five methods are as follows:

A. Method. *MI–07.3704–81.* Reduction of traffic noise by road design methods. Hungarian State Technical Report [4].

B. Method. Calculation method for the prediction of road and railway traffic noise [2].

C. Method. Investigation and evaluation of road traffic noise [1].

D. Method. *DIN 18005,* Part 1. Noise control in town planning. Draft, 1976 [3].

E. Method. The computing model for road traffic noise Nordic method [5].

An abstract of methods A, B and C will be given, while we refer to the literature as far as methods D and E are concerned.

Method A. The equivalent continuous A-weighted sound pressure level at a distance of 7.5 m from the centre line of the outermost traffic lane is

$$L'_{Aeq} = 10\log Q + L_{eq1} + 10\log\left(\frac{p_1}{100} + \frac{p_2}{100}\cdot 10^A + \frac{p_3}{100}\cdot 10^B\right),$$

where

$$A = \frac{L_{eq2} - L_{eq1}}{10}; \quad B = \frac{L_{eq3} - L_{eq1}}{10},$$

and

$$L_{eq1} = 39.6 - 0.1\log v_1; \quad L_{eq2} = 46 - 0.6\log v_2; \quad L_{eq3} = 54.4 - 3.7\log v_3,$$

and Q is the density of traffic related to one hour of the busiest eight hours, vehicle/hour; p_1 is the percentage of passenger cars, %; v_1 is the average speed of passenger cars, km/h; p_2 is the percentage of heavy vehicles with two axles, %; v_2 is the average speed of heavy vehicles with two axles, km/h; p_3 is the percentage of heavy vehicles with more than two axles, %; v_3 is the speed of heavy vehicles with more than two axles, km/h.

The calculation should be performed separately for each traffic lane and the results should be summed. The equivalent continuous A-weighted sound pressure level at a distance more than 7.5 m from the centre line of the outermost traffic lane is

$$L_{Aeq}d = L'_{Aeq} - \Delta L_d - \Delta L_L - \Delta L_G - \Delta L_B - \Delta L - \Delta L_E,$$

where ΔL_d is the noise attenuation due to the distance; ΔL_L is the noise attenuation due to the absorption of the air; ΔL_G is the allowance for the ground effect; ΔL_B is the noise attenuation due to the plants; ΔL is the noise screening effect of barriers; ΔL_E is the effect of other factors.

Method B. The equivalent continuous A-weighted sound pressure level, L'_{Aeq}, at a distance of 25 m from the centre of the road is

$$L_{Aeq}(25 \text{ m}) = 36 + 10\log N,$$

where N is the average traffic density in passenger car-unit vehicle/hour (p.c.U./h) during the reference time interval.

The formula applies to a speed of $v \leqslant 60$ km/h. The traffic in passenger car-unit vehicles (p.c.U.): passenger car, or scooter = 1 p.c.U.; heavy motor-vehicle = 6 p.c.U.; motorcycle = 3 p.c.U.; tram = 6 p.c.U.

The equivalent continuous A-weighted sound pressure level, L'_{Aeq}, of the road traffic at a random point of the invesigated area is

$$L_{Aeq} = L_{Aeq}(25 \text{ m}) + (K_s + K_t + K_f + K_b + K_e + K_h + K_r).$$

The individual terms of the equation are as follows:

a) correction depending on the speed of the vehicles, K_s, dB. When the speed, v, is: $v \leqslant 60$ km/h: $K_s = 0$; $v > 60$ km/h: $K_s = 26.9\log v - 47.8$.

b) correction depending on the distance between the road and the observer, K_t, dB. It is calculated for the case of a perpendicular distance r [m], from the road as follows

$$K_t = 10 \log 25/r.$$

In the case of soft ground at a distance over 25 m, if the height, h, of the observer from the ground surface is less than one-tenth of the distance r

$$K_t = 13.33 \log 25/r.$$

The first formula should be used, if K_f and K_h are applied.

c) correction depending on a forest belt or a sparse construction, K_f, dB

$$K_f = -0.05 d_f,$$

where d_f is the distance measured through the forest belt, or the sparse construction.

The correction can be used, when $d_f = 30$ m at least, however, K_f can not be lower than -10 dB.

d) correction depending on reflection, K_b, dB. In the vicinity of larger surfaces (e.g. facades) $K_b = 3$ dB. In a street with a closed construction on both sides

$$K_b = 3 + 10 \log\left(1 + \frac{h}{b}\right),$$

where h is the mean building height; b is the distance of the facades.

e) correction referring to the incline of the road and crossings, K_e, dB: up to 6 % incline of the road $K_e = 0$; over 6 % incline of the road $K_e = 2$ dB; within 100 m measured from the centre of the crossing $K_e = 2$ dB.

f) correction depending on screening, K_h, dB. The value of correction K_h should be determined as illustrated in Fig. 1.

g) correction referring to the angle of view, K_r, dB:

$$K_r = 10 \log \beta_i / 180,$$

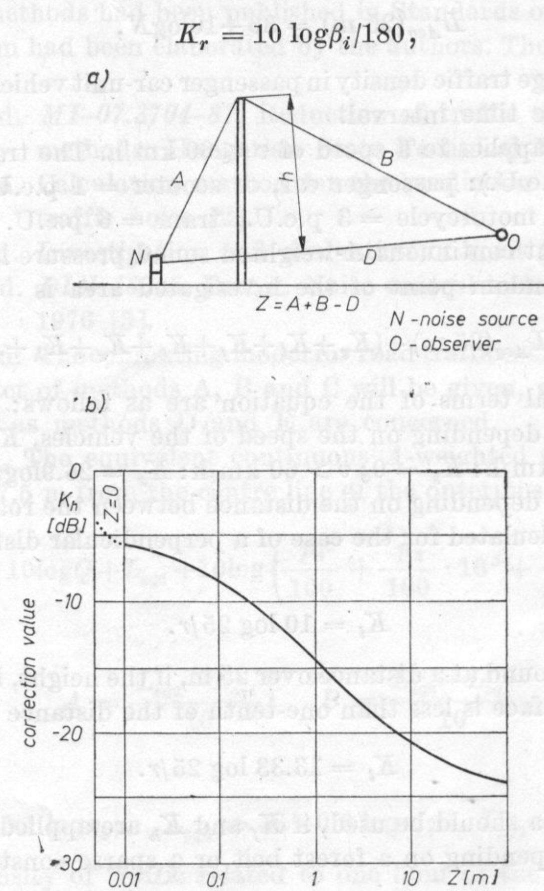


Fig. 1

where β_i is the angle of view in degrees of the ith section.

The equivalent continuous A-weighted sound pressure levels of each section should be summed.

Method C. The equivalent continuous A-weighted sound pressure level at 7.5 m from the centre line of the outermost traffic lane is

$$L_{Aeq} = 24 + 22 \log N - 2.2 (\log N)^2,$$

where N is the average traffic density expressed in passenger car-unit vehicle/hour, as defined in Method B.

Correction referring to the distance r, K_l, dB:

$$K_l = 10 \log \frac{7.5}{r}.$$

All further calculations are as described in Method B.

3. Results of the investigation

The analyzed locations were of different construction. There are two locations shown in Fig. 2 and 3, as examples of the arrangement of measuring points (B is the basic point).

Altogether there were 102 measurements done on the 11 locations at a distance of 7.5 m from the centre line of the outermost lane. Measurement results and those of the calculations were averaged for each location, thereafter the mean and the standard deviation of the differences between the calculated and

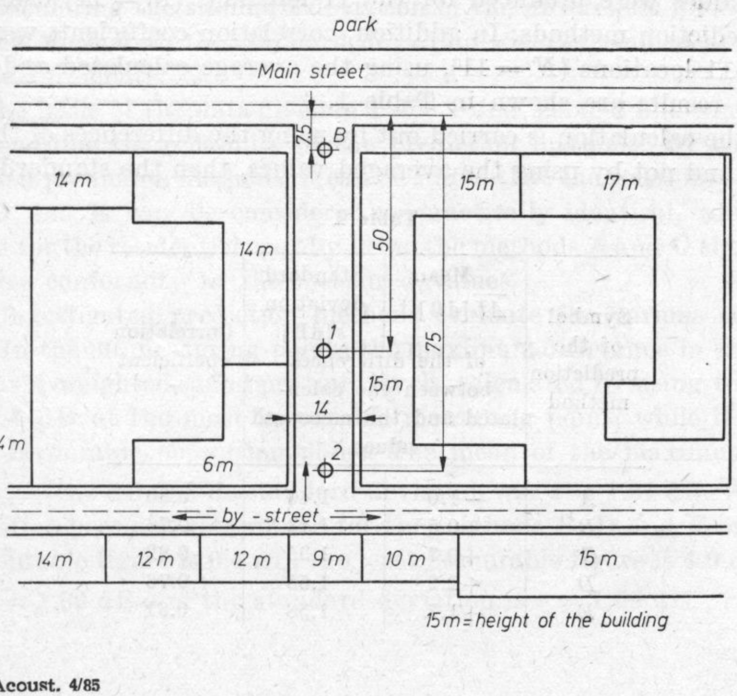
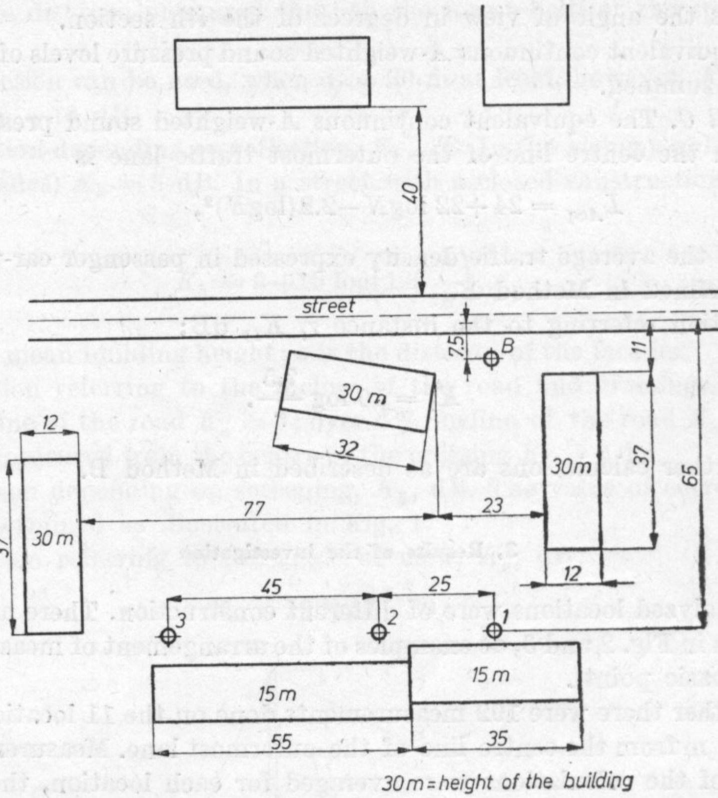


Fig. 2



30 m = height of the building Fig. 3

measured values were produced for the 11 locations ($N = 11$) using the five different prediction methods. In addition, correlation coefficients were calculated for the 11 locations ($N = 11$), using the average calculated and measured values. The results are shown in Table 1.

When the calculation is carried out by using the differences of the 102 measurements and not by using the averaged values, then the standard deviation

Table 1

Symbol of the prediction method	Mean $\overline{\Delta L}$ [dB] of the differences between the calculated and the measured values	Standard deviation s [dB] of the differences between the calculated and the measured values	Correlation coefficient r
A	-1.0	1.40	0.80
B	$+0.5$	1.43	0.76
C	$+0.7$	1.35	0.82
D	$+2.2$	1.55	0.76
E	$+1.7$	1.38	0.81

of the differences amounts to $s = 2.33$ dB instead of $s = 1.43$ dB, when Method B is used.

The calculated and measured values obtained at the measuring points inside the residential area were evaluated in the same way. The evaluation was performed for 30 measuring points at the 11 locations, the number of the measurements was 55. At some measuring points several measurements were done, the results of which were averaged first. Thereafter the differences between the calculated and measured levels were determined and the mean and the standard deviation of 30 differences was calculated. In addition, the correlation coefficients of the calculated and measured levels were also determined. The results are shown in Table 2. The maximum deviations from the measured values are also indicated in the table.

Table 2

Symbol of the prediction method	Mean $\overline{\Delta L}$ [dB]	Standard deviation s [dB]	Maximum deviation [dB]		Correlation coefficient r
	of the differences between the calculated and the measured levels				
A	−2.5	1.91	+0.8	−7.7	0.92
B	+0.7	1.39	+3.2	−1.4	0.96
C	−0.9	1.57	+1.7	−3.5	0.93
D	+0.9	1.47	+3.5	−2.5	0.96
E	+0.9	1.36	+3.0	−2.3	0.95

By calculating the standard deviation of the differences for each measurement using Method B, the standard deviation was determined at $s = 2.19$ dB for 55 measurements.

On the basis of the data given in Table 1 and Table 2 one can draw a conclusion regarding the reliability of the prediction methods. By comparing the investigated prediction methods, it can be stated that the reliability of the methods B, D and E can be considered as practically identical, while the levels calculated for the residential area by using the methods A and C showed a somewhat worse conformity to the measured values.

The investigated prediction methods evaluate the various situations differently. In the 30 measuring points the maximum difference in the equivalent continuous A-weighted sound pressure levels, calculated by using the five methods, is 2.4 dB at the most favourable measuring point, while it is 8.6 dB at the least favourable measuring point. The mean of the maximum differences was $\overline{\Delta L_{\max}} = 4.6$ dB and its standard deviation was $s = 1.51$ dB. When the maximum differences are determined for the methods B, D and E only, then the most favourable figure is 0.4 dB, the least favourable figure is 4.0 dB, the mean is $\overline{\Delta L_{\max}} = 1.80$ dB and the standard deviation is $s = 1.09$ dB.

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ANALYSIS OF DIRECTIONAL RADIATION PATTERNS OF A SYSTEM OF FLAT PLANE SOUND SOURCES

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This paper presents a method of determining directional radiation patterns of a system of arbitrary flat surface sound sources, source vibration velocities phase shifted by an arbitrary angle. Also the results of verification studies of this method for a system of two circular pistons are presented.

1. Introduction

One of the main causes for noise emission by machines and industrial installations are the material vibrations of their elements.

During the past few years several methods, allowing quantitative evaluation of noise emitted in industrial conditions, have been developed [4]. Research has been also conducted on predicting the vibroacoustical power emitted by a sound source in certain conditions [2]. The second problem becomes quite complicated in the case of simultaneous sound emission by several sound sources. This is due to the occurrence of interactions between source and air acoustical waves, which cause changes in the acoustical power system in relation to the power emitted by these sources independently (sources mutually isolated). Conducting quantitative research on acoustical energy emission requires the development of an effective method of determining acoustical self-impedance and mutual impedance of the sources. One of the acoustical impedance determination methods is based on the integration of the square root of the source directional radiation pattern modulus.

This paper presents a method of determining directional radiation patterns of a system of two arbitrary plane sound sources, vibrating out of phase with equal frequencies.

2. Directional radiation pattern of a system of flat plane sound sources

Mathematical analysis of an acoustical field produced by a source is a complex problem. An effective solution to this problem is possible in a case of flat plane sources vibrating in an infinite acoustic baffle in a free field. It is a theoretical case, rarely met in practice, but its solution can be applied with small corrections in the analysis of existing sound sources. Assuming an ideal acoustic baffle, the acoustic pressure in the Fraunhofer zone for a monoharmonic wave can be determined from the Fraunhofer integration formula

$$p(r, \theta, \varphi) = \frac{ik\varrho_0 c}{2\pi} \frac{e^{-ikr}}{r} \int_S v(\vec{r_0}) \exp[ikr_0 \sin\theta \cos(\varphi - \varphi_0)] dS_0 \quad [\mathrm{Pa}], \tag{1}$$

where $i = \sqrt{-1}$, k – wave number $[\mathrm{m}^{-1}]$, $\varrho_0 c$ – specific acoustical impedance of the medium $[\mathrm{kg\,m^{-2}s^{-1}}]$, r, θ, φ – space polar coordinates of the observation point, r_0, φ_0 – polar coordinates of a point laying on the source surface, S – source surface $[\mathrm{m}^2]$, $v(\vec{r_0})$ – normal component of the vibration velocity on the source surface in a point determined by vector $\vec{r_0}[\mathrm{m\,s^{-1}}]$.

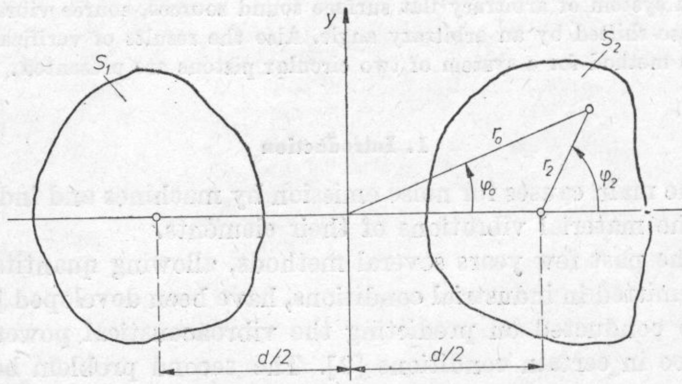


Fig. 1. Coordinate system applied for the determination of the directional radiation system of a system of flat plane sound sources

When the acoustical wave is emitted by a system of sources, S_1 and S_2, presented in Fig. 1, then the acoustic pressure at the distance r from the centre of the coordinate system in the Fraunhofer zone is expressed by the following formula:

$$p(r, \theta, \varphi) = \frac{ik\varrho_0 c}{2\pi} \frac{e^{-ikr}}{r} \Big[\int_{S_1} v(\vec{r_0}) \exp\big(ikr_0 \sin\theta \cos(\varphi - \varphi_0)\big) dS_0 +$$

$$+ \int_{S_2} v(r_0) \exp\big(ikr_0 \sin\theta \cos(\varphi - \varphi_0)\big) dS_0 \Big] \quad [\mathrm{Pa}]. \tag{2}$$

Acoustical pressure on the direction normal to the source surface $\theta = 0$, determined from (2), is

$$p_0(r) = \frac{ik\varrho_0 c}{2\pi}\frac{e^{-ikr}}{r}\left[\int_{S_1} v(\vec{r_0})\,dS_0 + \int_{S_2} v(\vec{r_0})\,dS_0\right] \quad [\mathrm{Pa}]. \tag{3}$$

Defining the directional pattern as a ratio of the acoustical pressure on the given direction to the pressure on the direction normal to the source surface, can be troublesome, because the expression in brackets in formula (3) can in certain cases equal 0. For this reason, the directional radiation pattern in the further part of this paper will be understood as the ratio of the acoustical pressure on the given direction to the acoustical pressure on the direction normal to the surface of an imaginary source, where the velocity distribution is expressed as

$$\hat{v}(\vec{r_0}) = |v(\vec{r_0})| \quad [\mathrm{m\,s^{-1}}]. \tag{4}$$

In a case of a system of sources, the acoustical pressure on the direction normal to the surface of an imaginary source is equal to

$$\hat{p}_0(r) = \frac{ik\varrho_0 c}{2\pi}\frac{e^{-ikr}}{r}\left[\int_{S_1} |v(\vec{r_0})|\,dS_0 + \int_{S_2} |v(\vec{r_0})|\,dS_0\right] \quad [\mathrm{Pa}]. \tag{5}$$

Substituting in formula (5) the following expressions

$$[v_1] = \frac{1}{S_1}\int_{S_1} |v(\vec{r_0})|\,dS_0 \quad [\mathrm{m\,s^{-1}}], \quad [v_2] = \frac{1}{S_2}\int_{S_2} |v(\vec{r_0})|\,dS_0 \tag{6}$$

we obtain

$$\hat{p}_0(r) = \frac{ik\varrho_0 c}{2\pi}\frac{e^{-ikr}}{r}\left([v_1]S_1 + [v_2]S_2\right) \quad [\mathrm{Pa}]. \tag{7}$$

As relation

$$r_0\cos(\varphi-\varphi_0) = \frac{d}{2}\cos\varphi + r_2\cos(\varphi-\varphi_2) \tag{8}$$

is valid for source 2, and for source 1,

$$r_0\cos(\varphi-\varphi_0) = -\frac{d}{2}\cos\varphi + r_1\cos(\varphi-\varphi_1) \tag{9}$$

where d — distance between the centres of the sources [m],
r_1, φ_1 — local polar coordinates of source 1,
r_2, φ_2 — local polar coordinates of source 2, then substituting expressions (8) and (9) in formula (2) and dividing it by expression (7), an expression for the

directional radiation pattern for a system of sources is obtained

$$D(\theta, \varphi) = \frac{\exp\left(-ik\frac{d}{2}\cos\varphi\sin\theta\right)S_1[v_1]D_1(\theta, \varphi)+\exp\left(ik\frac{d}{2}\cos\varphi\sin\theta\right)S_2[v_2]D_2(\theta, \varphi)}{S_1[v_1]+S_2[v_2]}. \tag{10}$$

Assuming that the sources are identical, they vibrate in the same phase with equal velocities, we obtain a known formula [6]

$$D(\theta, \varphi) = D_1(\theta, \varphi)\cos\left(\frac{kd}{2}\sin\theta\cos\varphi\right). \tag{11}$$

If sources are identical and vibrate with equal amplitudes of velocity, but the vibration velocity of the second source is out of phase at an angle of φ in relation to the vibration velocity of the first source, then after some simple conversions we achieve

$$D_2(\theta, \varphi) = e^{i\psi}D_1(\theta, \varphi). \tag{12}$$

Therefore the directional pattern of the system will be

$$D(\theta, \varphi) = \frac{1}{2}D_1(\theta, \varphi)\left[\exp\left(-ik\frac{d}{2}\sin\theta\cos\varphi\right)+\exp\left(i\psi+\frac{d}{2}\sin\theta\cos\varphi\right)\right] \tag{13}$$

while the pattern modulus will be expressed by

$$|D(\theta, \varphi)| = |D_1(\theta, \varphi)|\cos\left(\frac{kd}{2}\sin\theta\cos\varphi+\frac{\psi}{2}\right). \tag{14}$$

In a case of a system of rigid, circular vibrating pistons, the pattern modulus will equal

$$|D(\theta, \varphi)| = \frac{2J_1(ka\sin\theta)}{ka\sin\theta}\cos\left(\frac{kd}{2}\sin\theta\cos\varphi+\frac{\psi}{2}\right), \tag{15}$$

where a – piston radius [m], J_1 – first type Bessel function of the first order.

3. Physical model of a system of two piston sources

A system of two circular pistons vibrating in a rigid acoustic baffle has been analysed in the course of experimental investigations. Pistons with smooth and flat surfaces, mounted to two backing plates of electromagnetic heads of vibration inductors (RTF 11075) were the main element of the emission source excitation system. The bases of the inductors have been mounted to a soundproof casing (closed, wooden box), placed on a platform of an anechoic chamber.

Piston plates vibrated in circular openings of the acoustic baffle which separated the faces of the inductor heads and the bases of the inductors from the half-space above the pistons' surface. The vibrations of the pistons were forced kinematically by the head of the inductor.

Two identical pistons, of $2a = 0.14$ m diameter, forced to vibrate with a constant amplitude of vibration velocity in specified frequency ranges, were applied in investigations:

$v = 0.02\,[\mathrm{ms^{-1}}]$ for frequency range 200–500 [Hz];

$v = 0.004\,[\mathrm{ms^{-1}}]$ for frequency range 500–750 [Hz];

$v = 0.002\,[\mathrm{ms^{-1}}]$ for frequency range 850–2000 [Hz].

Frequency band 750–850 [Hz] was not taken into account, because of the occurrence of the effect of self-resonance of the inductor with the mounted piston. A piston of $b = 0.005\,[\mathrm{m}]$ thickness was placed in a square acoustic baffle with side length $l = 2\,[\mathrm{m}]$ and wall thickness $g = 0.07\,[\mathrm{m}]$, attached to the soundproof casing of the inductors.

Fulfiling the condition $l > \lambda$, the lower measuring frequency was fixed at 200 Hz.

The condition, $ka = 2$, for the analysed piston corresponds to a frequency of 775 Hz.

With the above system parameters, there should be a conformity between the calculation and measurement results in the far field in the frequency range 200–775 Hz.

In order to check quantitatively the discrepancies in the higher frequency range, the measuring range was widened to 2000 Hz. Preliminary tests of the distribution of piston vibrations have been conducted in order to determine to what extent the physical model complies with the theoretical model of two rigid pistons.

To this end following measurements were performed:

– measurement of the attenuation diagram of the vibrations acceleration, velocity and displacement, for the midpoint of a single piston,

– measurement of the characteristic of phase shifts between the vibration velocities of two points on the piston surface.

Measurement results have led to the following conclusions:

– the resonance frequency of the system equals 1100 Hz,

– the phase shift angle between the vibration acceleration of two points on the piston surface is constant and equals 0° in frequency range 200–750 Hz. Only for several narrow frequency bands it achieves the values between 10°–15°. At a frequency of 600 Hz a clear local change of the phase shift angle, achieving 90°, occurs,

– quick, transient changes of the phase shift angle in frequency range 750–850 Hz,

– the phase shift is constant and equals 180° in the range 850–2000 Hz; only in the range 1070–1120 Hz sudden changes of the phase shift occur, caused by resonance,

— the phase shift between the accelerations of vibrations of piston midpoints is constant in frequency ranges 200–750 Hz and 1250–2000 Hz,

— transient changes of the phase shift angle occur in the range 750–850 Hz.

Research has resulted in accepting the following measuring ranges: 200–750 Hz and 850–2000 Hz.

4. Theoretical and experimental investigations of the acoustical field distribution

The aim of the research was to determine the true and theoretical directional radiation patterns for a given model of a system of two pistons vibrating in an acoustic baffle. The measurements of the true patterns were conducted in an anechoic chamber. Sources were excited to vibrate monoharmonically with an equal amplitude by a system consisting of a sinusoidal signal generator which allows simultaneous generation of two signals shifted by an arbitrary angle, and two power amplifiers. Piston vibrations were controlled by a system of two accelerators and charge amplifiers, and a two-channel oscilloscope. Additionally, the phase shift angle between the vibration velocities of the pistons was measured

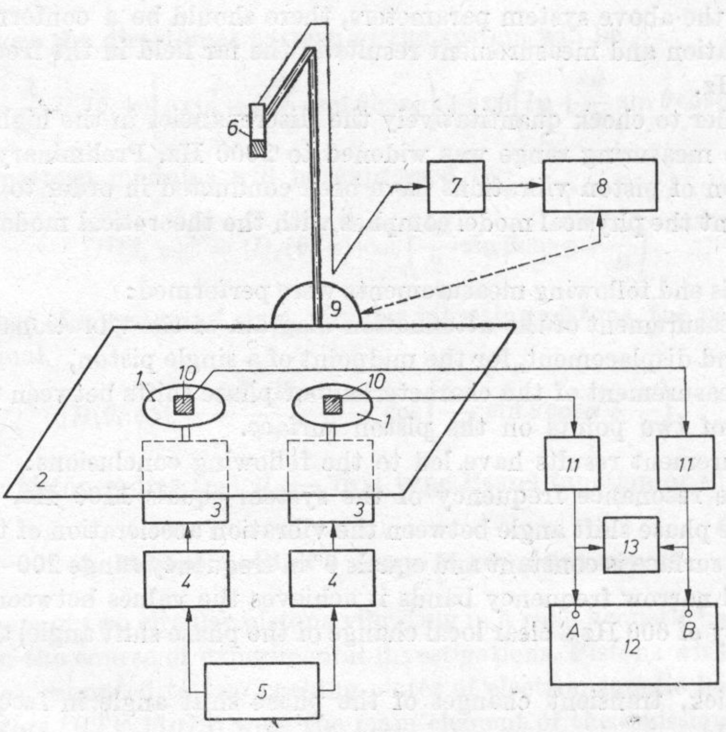
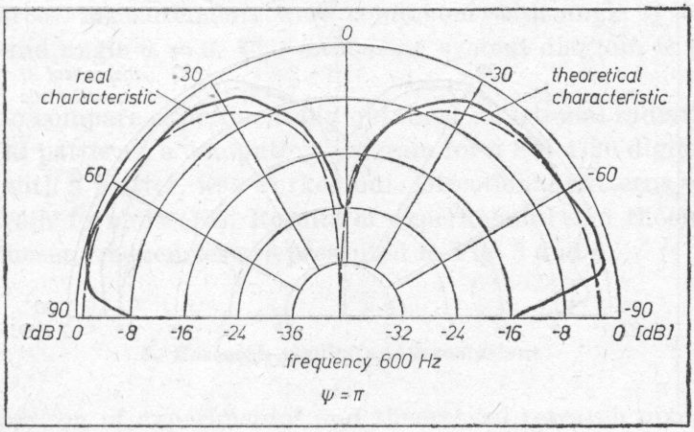
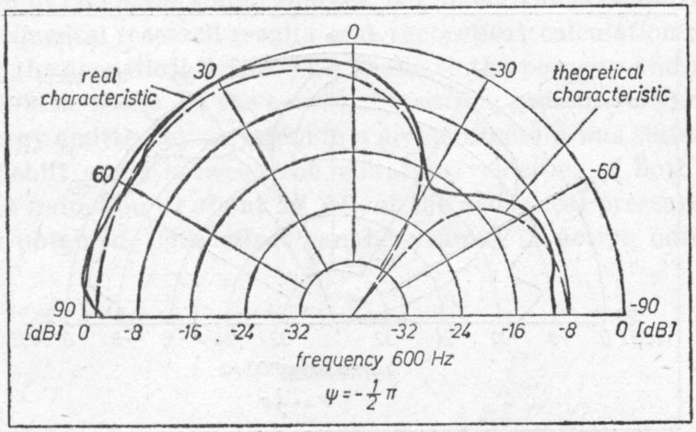


Fig. 2. Measuring system diagram: *1* — pistons, *2* — acoustical baffle, *3* — vibrations inductor, *4* — power amplifier PO–21, *5* — generator EMG 1117/6, *6* — microphone 4133 B&K, *7* — measuring amplifier 2607 B&K, *8* — level registrator 2307 B&K, *9* — rotary table 3921 B&K, *10* — accelerometer 4343 B&K, *11* — charge amplifier 2635 B&K, *12* — oscilloscope STD 501 XY, *13* — phasemeter 2971 B&K





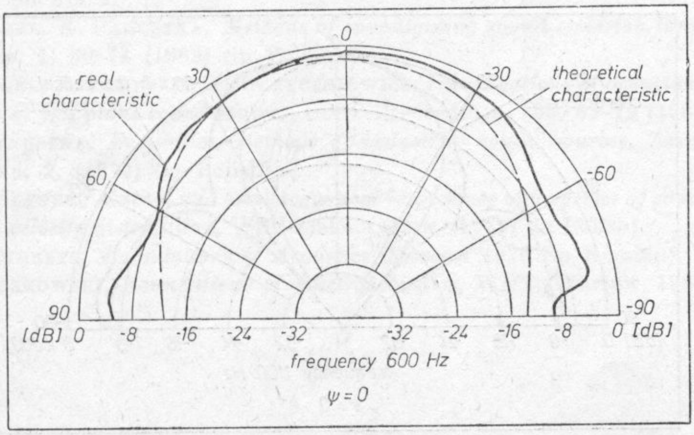


Fig. 3. Directional patterns of a piston system for a frequency forcing vibrations of 600 [Hz]

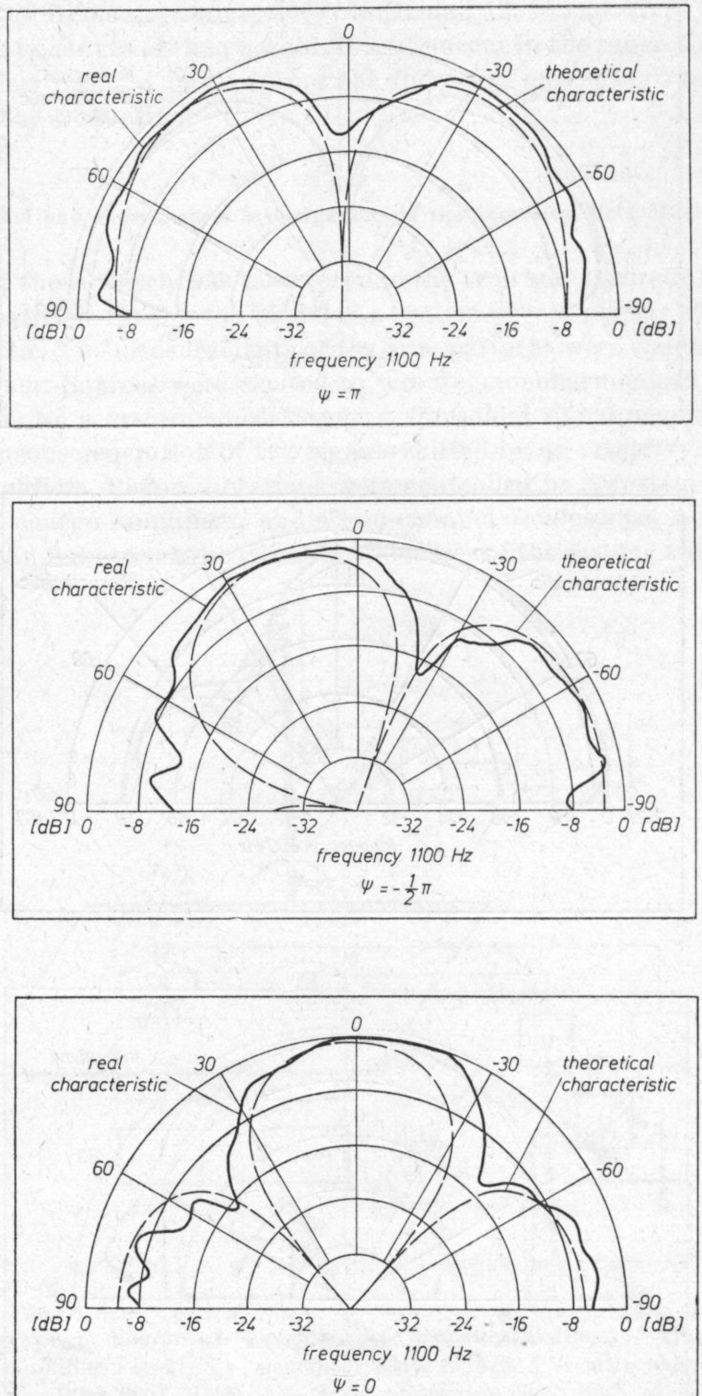


Fig. 4. Directional patterns of a piston system for a frequency forcing vibrations of 1100 [Hz]

by a phasemeter. The microphone descirbed a circle, 1 m in diameter, around the sound sources. Measurements were conducted with angle Q varying from $-\pi/2$ to $\pi/2$ and angle $\varphi = 0$. The measuring system diagram is presented in Fig. 2.

In order to compare experimentally obtained directional radiation patterns with theoretical patterns, a computing program for a SM 4/20 digital computer, collaborating with a plotter, was worked out. Directional patterns were calculated directly from formula (15). Results of experimental and theoretical investigations for chosen frequencies are presented in Fig. 3 and 4.

5. Research results and conclusions

The comparison of experimental and theoretical research gives their good qualitative conformity. Therefore we can see that the applied mathematical model of a system of flat plane sound sources, is proper. The greatest discrepancies between experimental research results and theoretical calculation results were observed near the acoustical baffle. This is due to the porosity and flexibility of the applied acoustic baffle. In the course of research, great diversification of the acoustical energy emitted by a system in a given direction was stated depending on the phase shift angle between the vibration velocities of both sources. In certain cases a reduction of about 20 dB of the acoustical pressure on a given direction was obtained. This effect can be utilized in active noise reduction methods.

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ACTIVE NOISE AND VIBRATION CONTROL (*ANVC*): CURRENT TRENDS, PERMANENT AIMS AND FUTURE POSSIBILITIES

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The main purpose of this report is to provide a deaper and more adequate understanding of *ANVC*. A literature of about 500 papers is now available and has been analysed: *ANVC* is now developed in many directions; many experiments have been done, proving that several kinds of noise and vibration can be attenuated substantially by active devices. Industrial and commercial exploitation seems a matter for tomorrow.

An epistemological obstacle is emphasized, which may be called "anticausality" or teleonomy. A shift of paradigm (in KUHN's sense) seems to be useful or even necessary to get a fair understanding of how to solve a practical active control problem.

The author recalls the origins of his own method, which had been applied and developed in acoustics by G. MANGIANTE and G. CANÉVET, becoming then the *JMC* theory. This theory has been recently reformulated in a set of three theorems on "field reshaping". The first theorem of Field Reshaping is analysed. It leads to a very general "designer's equation" which can be considered as the key to most active control systems. Two appendixes (Fig. 2–4) show how to meet some requirements of the *JMC* theory: a) location of the main sensor in the closest neighborhood of the antisource, i.e. using a feedback suppressor; b) digitizing the system for easier adaptative checking.

Plan:

1) Introduction, 1.1. Vocabulary and abbreviations, 1.2. A first practical reason for promoting *ANVC*, 1. 3. Preliminary sketches.

2) Current trends in *ANVC*, 2.1. Evolution of interest in *ANVC*, 2.2. *ANVC* once and now, 2.3. The most productive authors, 2.4. Keyword analysis.

3) A short excursion into epistemology, 3.1. WALDHAUER's anticausal ana-

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lysis, 3.2. Technology as the science of teleonomy and *purposeful planning*, 3.3. Alternation causality-anticausality.

4) *JMC* theory and Field Reshaping Theorems, 4.1. Origin and evolution of the *JMC* group, 4.2. The scenario of the first Field Reshaping Theorem, 4.3. The special case of linear acoustics and *ANC*, 4.4. Equivalence of the *JMC* tripole with other unidirectional doublet sources.

5) Aims and possibilities.

1. Introduction

1.1. Vocabulary, notations and abbreviations

ANC: Active Noise Control, or: Active Noise Cancellation

ANA: Active Noise Absorption, but **not** "Active Noise Attenuation"!

ANR: Active Noise Reflection, but **not** "Active Noise Reduction"!

AVC: Active Vibration Control, or: Active Vibration Cancellation

AVD: Active Vibration Damping (for vibrations which propagate across elastic media)

AVA, (resp. *AVR*): Active Vibration Absorption (resp. Reflection)

S': primary source of noise or vibration

F': primary field (radiated by S')

S'': secondary source (*s*), introduced either by the Huygens Principle (HP) or by the Principle of Active Absorption (PAA)

F'': secondary field (radiated by S'')

OP: *Physical* operator connecting a field F with its source (*s*): $OPF = S$ (hence also: $OP\,F' = S'$ and $OP\,F'' = S''$)

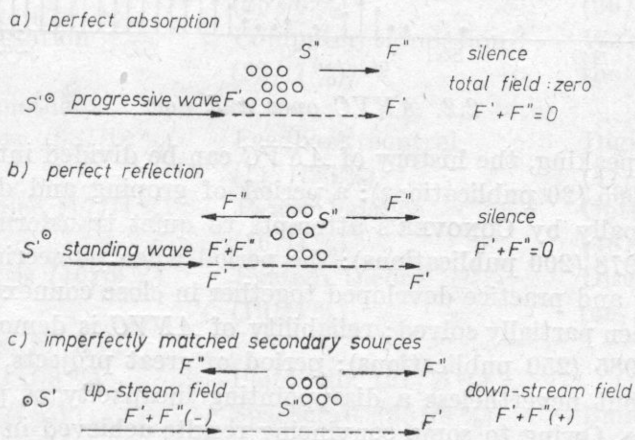
OR: *Radiation* operator expressing a field F in terms of its source (s): $OR\,S = F$ (hence also: $F' = OR\,S'$ and $F'' = OR\,S''$).

1.2. A first practical reason for promoting ANVC

The conventional approach to noise and vibration abatement is to passively absorb the vibratory energy, i.e. convert it into heat via viscous flow. In most cases, the total energy involved in noise or unwanted vibrations is very small, so that passive absorption will have negligible impact on the ambient temperature. The technology for passive attenuation is well developed and is extensively employed in mufflers, absorptive wall panels and damping coatings for vibrating structures. The absorption efficiency for a given passive device generally increases with the frequency of the noise; and for frequencies above 200 Hz-passive absorption is the usual and cheapest solution. But for very low frequen, cies, passive attenuators become very large and costly. Hence, we reach the first reason for supporting active methods of noise and vibration abatement.

1.3. *Preliminary sketches of wave control by secondary sources*

Three cases are emphasized: perfect absorption, perfect reflection and imperfect matching. It is important to note that these sketches are valid for passive as well as for active control. Passive secondary sources are considered to be induced in some absorbing or reflective medium by a convenient natural process. Active secondary sources are to be created artificially with the help of some adequate device.



Some degree of mismatch is allowable provided that the downstream field $F'+F''(+)$ remains small comparatively with the "incident" field F'; *up-stream* and *down-stream* is meant relatively to the set of S'' sources. The hypothesis that the set of S'' is perfectly transparent to the field F' is not essential: suffice to consider that in the down-stream area F' means the field radiated by S' with account of the presence of the set S''.

2. Current trends in *ANVC*

The most extensive compilation on *ANVC* literature has been drawn up by D. Guicking, who recorded 457 reference papers from 351 authors between 1933 and the end of 1984. Analysis of this bibliography provides good information on the evolution and present state of *ANVC*.

2.1. *Evolution of the interest in ANVC*

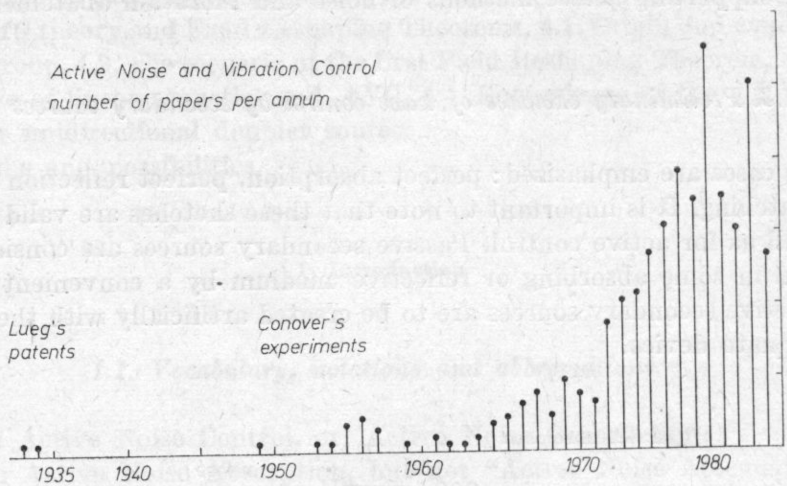


Fig. 1

2.2. *ANVC once and now*

Roughly speaking, the history of *ANVC* can be divided into 3 periods:

1° 1933–1965 (20 publications): a period of groping and disappointment, marked principally by Conover's attempts to quiet transformer hum.

2° 1966–1978 (200 publications): the period when pioneering became successful; theory and practice developed together in close connexion; major problems have been partially solved; reliability of *ANVC* is demonstrated.

3° 1979–1985 (250 publications): period of great projects, impressive demonstrations but nevertheless a disappointing incapacity of producing commercial devices. Owing to some convincing results achieved in the preceeding period, *ANVC* is (definitively?) recognized as a technique of real importance. Several new systems have been developed; more impressive demonstrations have been conducted; serious programs are under way in several noise-abatement hardware companies and industries with noise problems show an increasing interest in *ANVC*.

2.3. *The most productive authors*

(name)	(country)	(number of papers)	(speciality)
Balas	USA	13	Feedback control of structures
Bschorr	D	16	*ANC*
Eghtesadi	Iran	13	*ANC* (experiments)
Fedoryuk	USSR	8	*ANC* (theory)
Guicking	D	11	*ANC*
Jessel	F	20	*ANVC*
Kido	J	10	*ANC* (automatic control)
Leventhall	GB	17	*ANC*

MANGIANTE	F	12	*ANC*
MEIROVITCH	USA	17	Active control of dynamic systems
OZ	USA	11	Active control of dynamic systems
TARTAKOVSKII	USSR	16	*ANVC*
WARNAKA	USA	8	*ANC*

2.4. *Keyword analysis*

a) Kind of papers

Bibliography (19//4 %)	Historical (38//8 %)	
Overview (59//13 %)	Context papers (24//5 %)	
Fundamentals (76//17 %)	Theory (226//50%)	Problems (18//4 %)
Experimental (144//32 %)	Model experiment (25//5 %)	Modelling (26//6 %)
Model computation (46//10 %)	Computer simulation (32//7 %)	Waveform synthesis (18//4 %)

b) Means and methods

Control theory (83//18 %)	Feedback control (117//26 %)	Digital control (47//10 %)
Energy balance (17//4 %)	Power balance (16//4 %)	Optimization (46//10 %)
Modal analysis (40//9 %)	System theory (19//4 %)	Discretization (46//10 %)

c) Special topics in *AVC*

Beam vib. (38//8 %)	Plate vib. (21//5 %)	
Aircraft (34//7 %)	Spacecraft (35//8 %)	Buildings and bridges (31//7 %)
Structures (94//20 %)		
Vib. suppression (121//27 %)	Vib. dampers (20//4 %)	Vibr. insulation (18//4 %)

d) Special topics in Acoustics and *ANC*

Transformer noise (24//5 %)	Structure borne sound (37//8 %)	Noise (37//8 %)
Monopole (39//9 %)	Dipole (31//7 %)	Tripole (25//6 %)
1D (one-dimensional 117//26 %)	2D (5//1 %)	3D (91//20 %)
Duct (58//13 %)	Exhaust (20//4 %)	Turbulence (20//4 %)
Loudspeaker (18//4 %)	Headset (18//4 %)	

2.5. *Towards a better understanding of ANVC*

The prime aim of this report was to provide a better understanding of active confinement of noise and vibration. Usefulness of such an intention becomes evident as soon as one takes a look at the above panoramic view upon the work done currently in the field of *ANVC*. In spite of many encouraging results, its present state does not seem fully satisfying: effort is too dispersed and too fragmentary. An increase of interdisciplinary cooperation, among individuals with different backgrounds and skills, would surely prove very helpful for future development. Nevertheless a basic epistemological obstacle remains still to be removed: active control is a new sort of generalized feedback control; both appeal to a basic notion of intentionality and finality, whose vital importance in RESEARCH and DEVELOPMENT cannot remain over-looked any longer. Hence the next short excursion into epistemology.

3. A short excursion into epistemology

3.1. WALDHAUER'S "*anticausal analysis*"

In a book entitled "Feedback", Fred D. WALDHAUER introduces a new way of building up feedback theory and design. Instead of computing the output as a function of the input (as in the main causal analysis), he considers the output as known or given and expresses therefore the input in terms of the output. This new approach he calls "anticausal analysis" and shows in detail over more than 600 pages the benefits of it in designer's comfort and synthesis efficiency. Unfortunately he writes only a few lines on the philosophical aspects of his innovation. In fact WALDHAUER'S method introduces really a *paradigm shift* (according to KUHN'S terminology). Novelty and importance of such an epistemological revolution can be safely compared to that of COPERNICUS. It is only to be hoped that anticausal analysis will be accepted by the world of science and technology in less time than the heliocentric system!

3.2. Technology as the science of teleonomy and "purposeful thinking"

Epistemology is the science of sciences. In the same sense, technology should be not only the scientific justification of practical know how and technica practice, but also the very science of all the applied sciences, including their industrial and commercial developments. As in marketing and other commercial or economical sciences, anticausal approach becomes essential. Evidently Noise and Vibration Control belongs to applied mechanics. As control becomes active, with feedback effect, acceptance of teleonomy and anticausality must follow inavoidably.

3.3. Alternation causality-anticausality

Anticausal analysis, which proceeds from effect to cause, is fairly complemental to the common deductive causal analysis which proceeds from cause to effect. Both can be coupled and used alternatively in close connexion. Such an alternating approach will be emphasized in the next section in the framework of a very general problem on wave propagation. Thus will be obtained a very robust and versatile *designer's equation* for *ANVC*.

4. *JMC* theory and field reshaping theorems

Over 30 papers were published between 1968 and 1980 by the *JMC* group composed by JESSEL, MANGIANTE, CANÉVET and a few other students and engineers working on *ANC*.

4.1. Origin and evolution of the JMC group

As soon as 1954, while he was working on diffraction for a thesis under the direction of Louis de Broglie (the inventor of wave mechanics) and RENÉ LUCAS (one of the discoverers of the diffraction of light by ultrasound), JESSEL realized that Huygens' Principle (HP) leads by complementarity to a principle of active wave confinement. In 1965 with the help of RENÉ LUCAS, JESSEL was appointed at the Marseilles CNRS Laboratories with a view applying his idea of active confinement in acoustics. The *JMC* group started its work in October 1967 and soon obtained positive results on active absorption of a sinus sound wave in a duct. The group carried on its activity during about ten years, then it has been progressively dismantled for quite not scientific reasons. Since 1980 a different direction has be given in the research done in Marseilles on *ANC* and since 1981 JESSEL works alone on qure theoretical subjects: general system theory, holophony and *field reshaping*. The latest subject, which embeds very many applications, provides now the most general approach to *ANVC* and will perhaps lead to a more convincing understanding of *ANC* and *AVC*.

4.2. The scenario of the first Field Reshaping Theorem

As already told above, anticausal reasoning will alternate here with normal causal deduction.

a) The first step is causal: one is given a primary source S' and its radiated field F' with the connecting equation

$$OP\, F' = S'. \tag{1}$$

But one observes that F' is not convenient in some part of the space where it is measured (or computed by the inverse connection $F' = ORS'$).

b) Second step: introduction of *finality*, by defining a "required" field F which may be given everywhere or even only in the parts of space where it is different from F'. The problem now is to obtain F not from completely new sources S but from the source S' (possibly modified in a convenient way) with the help of some additional (secondary) sources S''. Therefore we now introduce an operator M "modifier" which can be applied to S' as well as to F': we deduce M from F and F' by writing

$$F = M\ F' \tag{2}$$

and, if necessary, we give M a convenient extension so M can be applied to S', leading then to the new source MS'.

c) Third step: anticausality leads us now to look for an equation whose solution is the required field F: the nature of the physical phenomena remaining inaltered, this equation will use the same operator OP. We consider therefore $OP\ F = OP\ M\ F'$. But than we must observe that usually $OP\ M \neq M\ OP$: the two operators do not commute. Therefore we have to introduce another set of sources: secondary sources S''. Then we will get

$$OP\ F = OP\ M\ F' = M\ S' + S'' = M\ (OP\ F') + S'' \tag{3}$$

which leads to

$$S'' = OP\ M\ F' - M\ OP\ F' = (OP\ M - M\ OP)F'. \tag{4}$$

Eq. (4) can be called the ***designer's equation*** for many technical problems including those of *ANC* and *AVC*.

Note that the approach contains really a good amount of anticausality, as we start here from a given field F in the course of looking for the sources (the causes) that will be the most convenient to produce it.

d) Following steps. The alternation can be carried on: usually the sources S'' can be put together only approximatively; such approximate sources S''_0 can in turn be used as primary sources for a second iteration. And so on ...

e) From (3) and (4), together with an explicit knowledge of $OR = OP^{-1}$, we obtain a formula which generalizes KIRCHHOFF's one

$$F = MF' = OR[MS' + (OP\ M - M\ OP)F']. \tag{5}$$

Observe also that GREEN's formula is useless here. In fact another and simpler identity plays here the role of the GREEN's formula

$$OP\ M\ F' = M\ OP\ F' + (OP\ M\ F' - M\ OP\ F'). \tag{6}$$

4.3. The special case of linear acoustics and ANC

For meeting linear acoustics, one has to specialize conveniently the operator OP, the field F' and the sources S'. The can be done as follows

$$F' = \begin{pmatrix} p' \\ v' \end{pmatrix}, \quad S' = \begin{pmatrix} q' \\ f' \end{pmatrix}, \quad OP = \begin{pmatrix} \delta_t & \text{div} \\ \text{grad} & \delta_t \end{pmatrix} \tag{7}$$

which corresponds to the system of equations

$$\delta_t p' + \operatorname{div} v' = q' \quad \text{and} \quad \delta_t v' + \operatorname{grad} p' = f'. \tag{8}$$

As for operator M, it will be given the constant value 0 in a part E_0 of space which is to be silenced. In another part E_1 it will, say, be given the constant value 1, as we cannot or will not alter the primary field in this part E_1 of space. We therefore write M as a matrix

$$M = \begin{pmatrix} m & 0 \\ 0 & m \end{pmatrix}, \tag{9}$$

where m is a continuous function with value 0 in E_0 and 1 in E_1.

Formula (4) provides now the "*designer's equations*

$$q'' = v' \cdot \operatorname{grad} m \quad \text{and} \quad f'' = p' \cdot \operatorname{grad} m. \tag{10}$$

This means that the *ANC* secondary sources S'' are made, according to the *JMC* theory, of monopole sources q'' and of force vector sources f'', the latest being equivalent to dipoles. When necessary, we shall emphasize them as *JMC sources* or *JMC antisources.*

JMC antisources may be more complicated than "tripoles" (monopole+dipole). When the velocity vector v' is no more collinear with grad m in (10), one will have to complete the system (8) by an equation of irrotationality for v': rot $v' = 0$. Then, by reshaping of v', a quadripolar source will appear: $g'' = v' \cdot \operatorname{grad} m$.

4.4. *Equivalence of the JMC tripole with other unidirectional sources*

M. A. SWINBANKS' double ring source can be symbolically presented as a sum of two unequal monopoles: $[Q_1(x-a)+Q_2(x+a)]$. The *JMC* tripole can likewise be $[q_1(x-a)+q_0(x)-q_1(x+a)]$. It is possible to shift over the short distance a the central monopole q_0 of the *JMC* source. Mathematically, the shift can be accounted for by replacing the diagonal mm of the matrix (9) by $m_p m_v$ i.e. by reshaping p' and v' in slightly different manners. The *JMC* tripole would thus become $[q_1(x-a)+q_0'(x+a)-q_1(x-a)]$ result which can easily be identified with the SWINBANKS' doublet. Note that ANGEVINE also simplified the *JMC* tripole to a couple of asymmetric loudspeakers.

5. Aims and possibilities

The main question concerning Active Control, on Vibrations as well as noise, is that of its feasibility and economical interest. For its theory cannot be falsified, except by requiring the falsity of one or another of the firmest axioms in mechanics and classical physics. But the connectio is new and apparently some taboo subject has been evoked. Aims and claims of the *JMC* theory are straightforward conclusions directly deduced from its *designer's equations*: Wave Reshaping and consequently *ANC* and *AVC* must work for mechanical waves of any kind and in any medium, and for non-linear phenomena as well as for the linear ones. As for the waveforms, the easiest to cancel is of course the sines and the sum of sines. Then the stationary noises with relatively narrow frequency bands and eventually the non-stationary broadband noise. The difficulties of materializing the *ANC* antisources are of technical origin. For tight

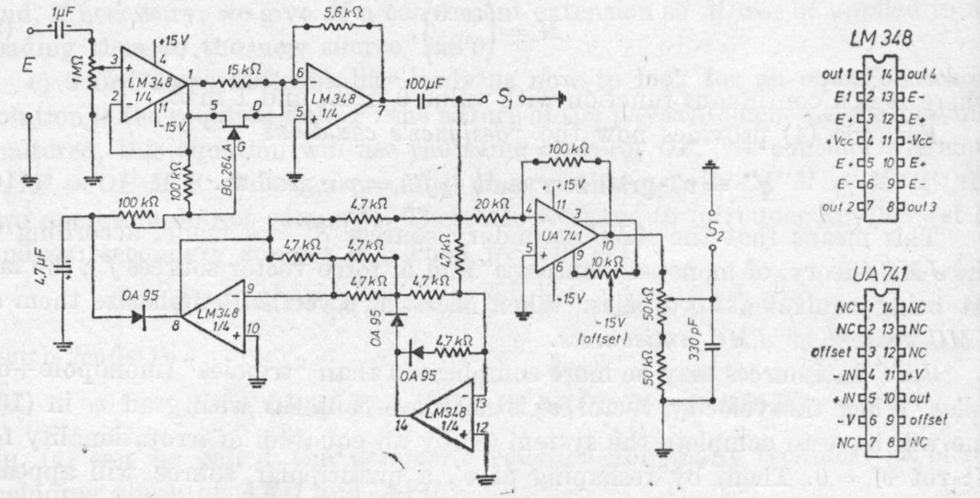


Fig. 2. Amplifier with feedback suppressor, after TH. ANGELINI (formerly in the *JMC* group *E*: input; S_1 output without suppression; S_2 output with suppression

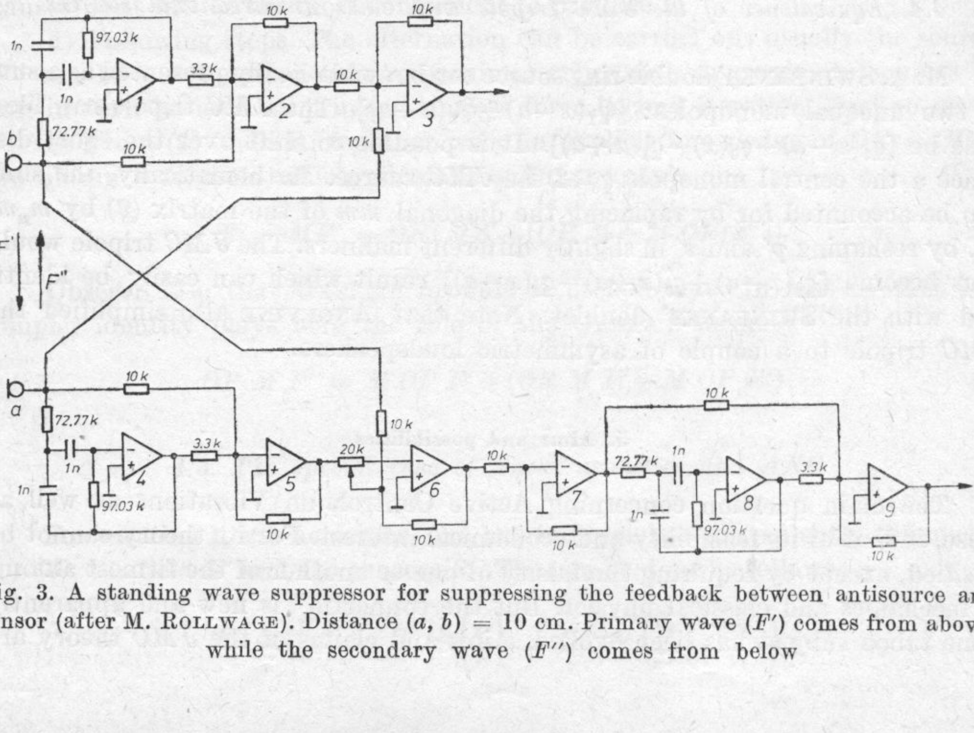


Fig. 3. A standing wave suppressor for suppressing the feedback between antisource and sensor (after M. ROLLWAGE). Distance $(a, b) = 10$ cm. Primary wave (F') comes from above, while the secondary wave (F'') comes from below

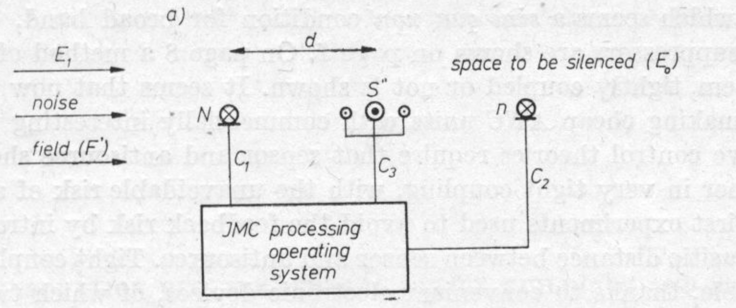


Fig. 4a. The system of antisources S'' is fed through a convenient operating system whose example is given in *b*). N is the main sensor (possibly a set of microphones measuring not only the incident pressure p' but also the particle velocity v'). n is another sensor or microphone for checking the efficiency of the antinoise system (N, S'') and possibly improving it by adaptative action. Channels *C1*, *C2* and *C3* (or C_1, C_2 and C_3) are respectively for the data concerning N (main *JMC* sensor), n, the checking microphone and S'', the *JMC* antisource

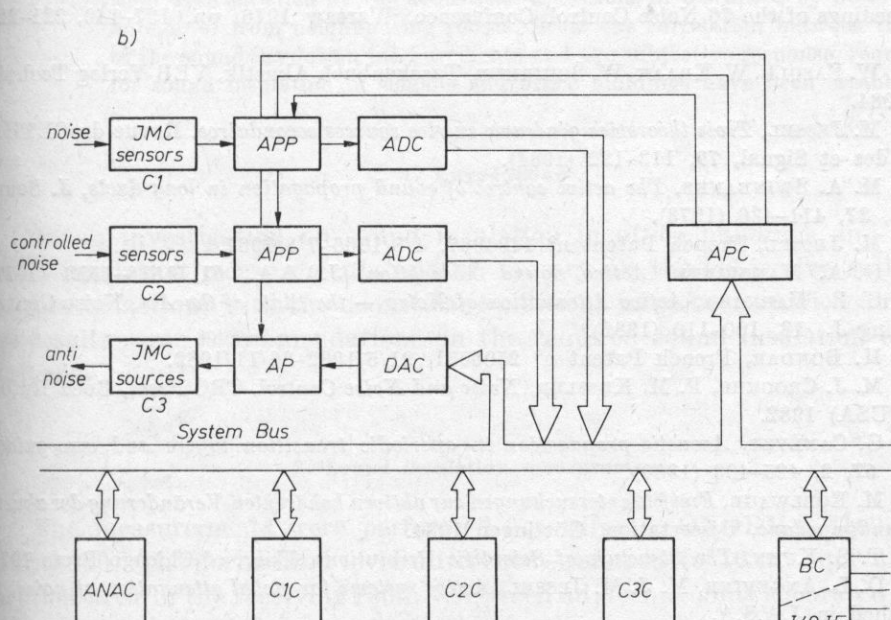


Fig. 4b. Details of the *JMC* operating system (according to G. MANGIANTE). Preprocessing means filtering, sampling and preamplifying (*APP*). Channels *C1*, *C2* and *C3* can operate in parallel, to speed up the process. Abbreviations: *APP*: Analogic "PreProcessor"; *ADC*: Analogic-Digital Converter; *AP*: Analogic Processor; *DAC*: Digital-Analogic Converter; *APC*: Analogic Processor Controller; *ANAC*: Active Noise Absorber Controller; *C1C*, *C2C*, *C3C*: Channel 1, 2, 3 Controller; *BC*: Bus Controller; *I/O IF*: Input/Output InterFace

coupling, which seems a *sine qua non* condition for broad band, examples of feedback suppressors are shown on page 7. On page 8 a method of digitzong a *JMC* system, tightly coupled or not is shown. It seems that now all elements exist for making cheap *ANC* units with commercially interesting possibilities. Most active control theories require that sensor and antisource should be placed together in very tight coupling, with the unavoidable risk of a ringing feedback. First experiments used to avoid the feedback risk by introducing a so to say parasitic distance between sensor and antisource. Tight coupling becomes now possible, thanks to convenient electronic devices, of which two examples are shown on this page. Papers on tight coupling have also been published by H. G. Leventhall and W. Hong. Rollwage's device was used in an active system for changing the acoustical impedance of a wall: the wall can be made perfectly absorbing as well as reflecting to a required degree!

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Comments:

Jessel's patent [7, 1966] concerns an *ANC* system with adaptative checking.

Bondar's patent [10, 1982] concerns a cold plasma loudspeaker which is now developed in an improved form. Such systems of loudspeakers, with very good performances also at very low frequencies, would be very convenient as *ANC* antisources.

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SOUND INSULATION REQUIREMENTS IN OFFICE BUILDINGS AND SCHOOLS

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By order of the Austrian Ministry for Building and Technology the sound insulation between adjacent rooms in schools and office buildings has been measured. Additionally all teachers teaching in classrooms of schools and all employees working in office rooms were asked with a questionnaire to what extent they were satisfied by the acoustical conditions or disturbed by noise from the outside or from neighbouring rooms. From the correlation between the results of the sound insulation measurements and the subjective response, requirements for sound insulation in schools and office buildings have been established.

1. Introduction

In an investigation on sound insulation in office buildings and schools, sound insulation measurements and an inquiry on the subjective satisfaction (or dissatisfaction) with the acoustical conditions have been carried out. From the results some recommendations on the required sound insulation could be derived.

2. Sound insulation measurements

The measurements were performed according to ÖNORM S 5100 (similar to ISO 140). The normalized level difference —referred to 10 m² equivalent absorption area in the receiving room —between adjacent rooms separated by partition walls or floors

$$D_n = L_1 - L_2 + 10 \log 10/A,$$

D_n — normalized level difference in dB, L_1 — sound level in the source room in dB, L_2 — sound level in the receiving room in dB, A — equivalent absorption area in the receiving room in m², was measured in third octave bands from 100–3150 Hz and the weighted normalized level difference D_{nw} was calculated to *ISO 717*. For floor also the impact sound insulation was measured by the normalized impact sound level — octave band levels in third octave steps referred to 10 m^2 equivalent absorption area in the receiving room — and the "Trittschallschutzmaß" (impact protection margin) was calculated by using the reference curve[1].

The tested rooms were sampled according to different structural details, e.g. different types of walls, different junctions of partition walls and external walls, to cover all existing types of sound insulation in the buildings. Especially rooms, in office buildings, separated by a partition wall, with or without a door, had to be considered separately.

3. Questionnaire on subjective satisfaction or dissatisfaction with acoustical conditions

A questionnaire has been worked out to ask the following details:

In the buildings:

Disturbance felt in 3 gardes: *slightly disturbed*, *disturbed*, *highly distrubed*

- when discussing with colleagues
- when listening to the telephone
- when performing mental work

caused by activities in the adjacent rooms (generally 4 rooms) as

- speaking, telephoning
- typing noise
- impact noise, chairs and similar

and caused by outside noise sources (e.g. traffic)

- with windows open
- with windows closed.

In school buildings:

Annoyance felt (in 3 grades as before)

- when teaching
- when pupils are speaking
- when tests have to be performed by the pupils

caused by activities in the adjacent classrooms (generally 4 rooms) as

- talking
- singing
- walking
- moving chairs

and caused by outside sources (e.g. traffic, sport areas)

- with windows open
- with windows closed.

The sound insulation measurements and the inquiries have been carried out in 6 schools and in 4 office buildings.

[1] In Austria the impact sound level is still measured in octave bands and weighted by a shifted reference curve (with $L_n = 70$ dB at 100 Hz).

4. Sound insulation in office buildings

From the results of the inquiry (in summary 736 questionnaires) and of the sound insulation measurements the following recommendations can be derived: The main disturbance in office rooms is caused by traffic noise. E.g. in one building exposed to an A-weighted equivalent sound level of 72–75 dB 22 % of the employees felt disturbed and 50 % highly disturbed with windows open, and with double glazed windows closed 47 % felt disturbed at the telephone and 60 % at mental work, even 20 % disturbed or highly disturbed.

In average the disturbance is the same for all floors. Therefore, designing buildings

– traffic noise at the site

– the required sound insulation of windows and facades according to this as prescribed in ÖNORM B 8115 (see Table 1)

– the necessary air change via a mechanical ventilating system or silenced air-intakes and outlets

have to be considered.

Table 1. Minimum sound insulation of facades of office buildings

A-weighted equivalent sound level [dB]	$\leqslant 65$	66–70	> 70
minimum weighted sound insulation index R_w[dB]	33	38	43

Talking (also at the telephone) in the neighbouring room and much less typing in the neighbouring room was the main noise source inside the building. Discussion with colleagues or telephoning is little disturbed by the neighbouring noise sources, disturbance during mental work was quoted much more; the evaluation of required sound insulation was therefore based on this.

The required sound insulation inside the building is shown by the following outcome of the inquiry for different buildings:

It turns out, that a normalized level difference $D_{nw} \geqslant 45$ dB between office rooms may be recommended. In several questionnaires the respondents mentioned, that avoiding disturbance is not as important as ensuring privacy for certain discussions in certain conference rooms or similar. To ensure this a higher normalized level difference of $D_{nw} \geqslant 55$ dB (the minimum requirement between dwellings) is recommended.

The disturbance caused by activities in rooms separated by a partition wall with a door turned out to be very small, though the measured sound level difference was only 24–25 dB. A minimum sound insulation between office rooms connected by a door does not seem necessary.

Table 2. Sound insulation and disturbance in office rooms

Weighted normalized level difference D_{nw} [dB]	Percentage of employees disturbed slightly disturbed highly disturbed disturbed		
	by noise in office rooms separated by a partition wall		
30–36	42	3	–
31–34	45	12	3
34–39	19	5	3
34–39	57	7	2
37–41	30	6	2
	by noise in office rooms separated by a floor		
44–46	very slightly disturbed		
47	no disturbance		
47–48	no disturbance		
61–63	no disturbance		
61–63	no disturbance		

In several questionnaires however disturbance caused by talking and walking in the corridors was mentioned. Therefore it seems necessary to require a minimum sound insulation between office room and corridor with $D_{nw} \geqslant 35$ dB (walls with doors) and $D_{nw} \geqslant 45$ dB (walls without doors).

The required impact sound insulation between office rooms can be derived from the following results of the questionnaires:

Table 3. Impact sound insulation and disturbance in office rooms

Trittschallschutzmaß [dB]	Percentage of employees disturbed slightly disturbed highly disturbed		
+ 3	27	—	—
+ 8	no disturbance		
+11	5	5	—
+30	no disturbance		

An impact sound insulation margin of $\geqslant 10$ dB may be recommended. This should be also ensured between corridor and office room at the same level (impact sound insulation in horizontal direction).

5. Sound insulation in school buildings

From the results of the inquiry (in summary 411 questionnaires) and the sound insulation measurements the following recommendations can be derived:

Disturbance caused by noise outside the building (with open windows) is considerable; not only traffic noise is the sources but also noise produced by the activities of the pupils and the personnel such as sport, kindergarten, parking lots for motorcycles, lawn mawers and similar. When designing school buildings the ground plan should be adjustd to avoid disturbance caused by these sources.

Disturbance from classroom to classroom via the open windows was mentioned in school buildings set around a courtyard. If a ground plan with a courtyard is chosen (e.g. to protect the classrooms against traffic noise) care has to be taken to avoid "cross talking" via the courtyard.

Talking, music and singing (if relevant) in the neighbouring classroom is the main noise source inside the building. The disturbance is quoted about equally during teaching, listening to pupils and at tests.

The required sound insulation within the school building can be derived from the outcome of the inquiry as follows:

Table 4. Sound insulation and disturbance in classrooms

Weighted normalized level difference D_{nw} [dB]	Percentage of teachers quoting disturbance
	by noise in classrooms separated by a partition wall
24–26	100% quote to be disturbed or highly disturbed
29–30	100% quote to be highly disturbed
32–34	75% quote to be disturbed
33–39	single guote to be disturbed, during tests also to be highly disturbed
38–39	25% quote to be disturbed, 15% also to be disturbed or to be highly disturbed by music activities
38–45	single quote to be disturbed, especially by singing
44–47	20% quote to be disturbed
50	15% quote to be disturbed
	by noise in classrooms separated by a floor
44–46	no disturbance
47–48	single quote to be disturbed by singing
47–48	single quote to be disturbed by music
48–49	no disturbance
51	no disturbance
52–53	single quote to be disturbed by singing

It turns out that the requirement $D_{nw} \geqslant 55$ dB according to ÖNORM B 8115 is ensuring sufficient sound insulation between classrooms, also taking into account music and singing. $D_{nw} \geqslant 55$ dB seems comparably high and it is therefore – according to the usually existing absorption in classrooms – recommended to sustain the requirement of $D_{nw} \geqslant 55$ dB, but to refer it to an equivalent sound absorption area of 25 m^2 instead of 10 m^2.

The impact sound insulation gives the following assessment:

Table 5. Impact sound insulation and disturbance in classrooms

Trittschall schutzmaß [dB]	Percentage of teachers quoting disturbance
−3 dB	15 % quote to be disturbed or to be highly disturbed by walking and moving chairs
+5 dB	single quote to be disturbed by walking, 20 % quote to be disturbed or to be highly disturbed by moving chairs
+11 dB	single quote to be disturbed by walking, 25 % quote to be disturbed or to be highly disturbed by moving chairs
+14 dB	single quote to be disturbed by walking, moving chairs
+14 dB	no disturbance

An impact sound insulation margin $\geqslant 15$ dB, required according to ÖNORM B 8115, seems sufficient to avoid disturbance by walking and moving chairs.

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DIAGNOSTICS IS A PROPER FIELD FOR NOISE CONTROL ENGINEERS

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The noise and vibration produced by machines have Tong been of concern to noise control engineers. The emphasis of such concern has been the effect of noise on people, whether hazardous or simply objectionable. The use of vibration signals to reveal operating characteristics of the machine is a new and requires a new set of measures of both the characteristics of the noise signals and the effects of the machine structure on these signals. This paper discusses the relation between noise studies and diagnostics in terms of their similarities and differences, and suggests that diagnostics is a proper field of study and work for noise control engineers.

I. Comparison of diagnostics and noise reduction

When mechanism operate within a machine, forces are generated that produce vibratory motions. These vibrations are transmitted through the machine and result in exterior surface vibrations and radiated sound. Whether the sound or vibration is of concern as a noise problem or is to be used for fault detection in a diagnostic system, these features of excitation, transmission, and radiation are important. The similarity of these basic aspects of noise problems and diagnostics is evident [1].

The greatest differences between diagnostics and noise lie in their respective goals. A machine operating properly and without faults can still be very noise, and a machine that has developed a major fault can still operate quietly. Signal features that may be important for diagnostics, such as phase, are generally ignored in noise problems. Signal energy in the speech bands, so important in noise problems, has no special role in diagnostics.

Although both noise reduction and diagnostics represent established activities, there has been relatively little interaction between these fields. This results partly from a difference in goals and partly due to differences in the frequency range in the signals of interest. But more interest in source noise control and newer signal processing methods that can gain more information from broad band signals have given emphasis to areas that have features in common. It is natural, therefore, that these two specialities of engineering acoustics — noise control and diagnostic analysis — should become more closely related subjects of research and application.

II. Goals: noise reduction vs. diagnostics

The goals of a noise reduction program generally include one or more of the following [2]:

Reduction of hazard — such as hearing loss, hypertension, etc. This is a traditional goal of noise control — it is of lesser concern in some countries at present because of decrease in regulatory activity.

Reduction in functional impairment such as speech interference, task performance, etc. This area continues to be an important goal for noise reduction since users of machines must be able to function in the environment of the machine.

Increased acceptability of sound — such as consumer satisfaction: and the related goal — enhanced product image.

These last two goals are occasionally of interest to manufacturers of consumer products and represent areas in which noise reduction engineering could be of greater service in product design. Helping manufacturers to meet these goals will require new approaches in product sound evaluation — stepping away from *A*-weighted intensity measures to evaluation procedures that may be very product specific.

The goals of diagnostics are usually thought of in positive terms:

To determine presence of faults, or perhaps, more generally,

To determine operating parameters,

since out of bound parameter values (fuel charge, clearances, ignition timing) can also be considered to be faults. It is perhaps more accurate, however, to describe these goals in a rather negative manner:

Not to make mistakes.

There are two kinds of mistakes one can make in diagnostics:

Type 1 error: There is no fault, but a faulta condition is declared.

Type 2 error: There is a fault, but a safe condition is declared.

Depending on the situation, the relative penalties for making each kind of mistake can be quite different. Type 1 errors can be very costly if a production process is needlessly interrupted. Type 2 errors can be expensive in security systems.

It is clear that meeting psychoacoustical criteria is crucial to realizing the goals of noise control. Since listening to machines is a traditional and practical method of diagnostics, some diagnostics system designers have sought to "do what the person does" in detecting faults. Although this approach can be successful in particular instances, it may lead to even greater efforts to "diagnose" the listener, i.e., to learn what the person is doing in making diagnostic judgements. In general, it will be better to spend this effort learning enough about fault related vibration generation and transmission in order to design the diagnostic system to detect and recover the appropriate signals.

III. Generators of vibration

Both diagnostics and noise analysis use power spectra of excitation sources as signals of interest. Diagnostics has generally used line spectra of rotating machinery to detect imbalance, misalignment, and other faults. Because the relative strengths of the spectral lines that correspond to shaft multiples are important the frequency dependence of the transmission path for such vibrations is also important [4].

Broadband, fullor third- octave spectra are of special importance in noise analyses, but they can also be useful for diagnostics. For example, impact and roughness in bearings produce broadband excitations and simple energy detection in these high frequency bands can indicate faults. The distinction between line and continuous spectra is not perfect, of course, since jitter in the occurrence of a *once-per-rev* impact can a transition from a line spectrum at low frequencies to a continuous spectrum at high frequencies.

The major division among approaches to determining source signatures is *empirical* vs. *analytical*. This is best exemplified by studies of *piston slap*, a noise or vibration pulse produced by piston and cylinder wall impact in diesel engines [5]. There have been a number of studies of the dependence of piston slap noise on ring clearance, cylinder pressure, piston/cylinder clearance, etc. An analytical study of this problem allows one to determine both spectra and temporal waveforms of the interaction force between piston and cylinder wall. The temporal waveform of the excitation source can have significant diagnostic value.

Whether the source *signature* of interest is an excitation spectrum or a temporal waveform it is unlikely that analytical or empirical procedures will be able to determine all signatures of interest. As in noise analysis, diagnostic procedures may be required to determine source properties from "output" data, the external machine vibration or sound radiation.

IV. Structural response of the machine

Noise analysis has typically taken a rather simplistic approach to structural noise transmission. Since signal phase and spectral details have relatively litle importance for noise, models such as Statistical Energy Analysis that provide estimates of the magnitude of a frequency and spatially averaged transfer function are widely used. Details of phase and magnitude are important in the performance of isolators, and in the energy injected into a structure by narrow band sources such as rotating components, but often a frequency averaged estimate is satisfactory.

As noted above, diagnostic systems may (or should) require much more detail in the knowledge of the transfer function.

The amplitude of individual line components, such as the meshing frequency of a gear, compared to other levels can reveal important features of machine performance. If these relative amplitudes are modified by the transmission path, their diagnostic value is reduced. It is partly for this reason that changes of these amplitudes over time may have more diagnostic value and such changes are frequently tracked in practical systems [6].

The recovery of source signatures from output data requires that we know the magnitude and/or phase of the transfer function much better than we need to for noise analysis [7]. We have noted the importance of the magnitude spectrum for an accurate determination of line spectra. The recovery of temporal waveforms requires a detailed knowledge of the phase spectrum, an almost unexplored area in structural transmission studies. Recent research has shed more light on the phase properties of transfer functions, but it is still not clear how the information that is needed to recover source signatures is to be obtained.

The need for detailed information on structural transmission raises a related issue of variability. A diagnostic system may be installed in a variety of machines and both the transfer function for vibration and the transducer mounting impedance can vary greatly. Thus, vibration spectra on machines of the same general type but that are different models or from different manufacturers may not be at all comparable [10]. This makes the use of vibration criteria based on vibration levels alone unreliable, particularly at frequencies above the first few machine resonances.

It is perhaps more surprising that transfer functions measured on nominally identical machines should show a great deal of variability in magnitude and phase, but they do [11]. This means that a diagnostic system designed for a particular machine model may have to be in some way "tuned" to each individual machine to which it is applied. This is beyond the capabilities of systems that are currently on the market, but research on this topic is being actively pursued [12].

V. Signal processing of vibration data

The vibration sensor location for noise studies may be closely specified, such as above an isolation mount, or loosely specified, i.e., at an "average" position on a housing or cover. Frequently, the band levels at a number of locations will be averaged in order to predict noise from the machine, or to correlate with predicted values.

There is a great deal more experience in relating vibration levels to noise than there is to machine faults. Perhaps for this reason, sensor locations and mountings for diagnostics tend to be highly specified. Current systems tend to use one transducer per machine element using observed and pick-up locations typically at bearing caps or shaft journals. More advanced systems will be required to have sensors placed at a greater distance from the source (*s*) and to use multiple sensors, but fewer than the number of sources to be diagnosed. This requires multi-dimensional source waveforms or spectra.

Our group at MIT has been using inverse filtering to recover source spectra and waveforms. The design of inverse filters is a key element to the success of this approach. A single source (diesel engine combustion pressure, for example) and sensor (accelerometer on the engine casing) requires an inverse filter that has a response that is just the opposite of the transmission path through the engine structure, both in magnitude and phase.

There will be a number of narrow bands in which the structure transmits very little energy (notches), so that noise contamination at the sensor is a problem. An inverse filter will have a great deal of gain at these frequencies and may, therefore, amplify output noise. It is possible to develop a criterion for a maximum gain that the inverse filter is allowed to have based on the signal to noise ratio at the output as estimated from the coherence function.

There is also a multi-dimensional version of this criterion using singular value decomposition of the matrix of transfer functions from the various sources of vibration to each sensor. The singular values represent normalized gains of the inverse (matrix) filter at each frequency. A general criterion for the extreme values of singular values based on the coherence function can be developed that is a direct analogy to the above two-port example [13].

The recovery of source waveforms, important for the diagnosis of impact events, depends on knowing the phase delay in the structural propagation. Gaining this knowledge is not easy, however, and the problem of designing inverse filters for mechanical systems forms a major part of our current research.

VI. Future developments

Noise control and noise reduction design have been and will continue to be major engineering activities, but the use of vibration and acoustical information for diagnostic purposes is growing and will continue to do so. The trends towards automated processes in manufacturing, intelligent machines, and the push for higher quality places a requirement for systems that monitor themselves, can identify faults, and suggest corrective actions. The sound and vibration produced by the operation of machines are not the only available signals for diagnostics, nor should these signals be used to the exclusion of temperature, fluid flow, currents and voltages, etc. Nevertheless, sound and vibration are intimately connected with the operations of mechanisms, and will have a central role in the development of diagnostic systems.

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SIMPLE ESTIMATION METHODS FOR NOISE REDUCTION BY VARIOUSLY SHAPED BARRIERS

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Simple estimation methods for various noise barriers are reviewed from the practical point of view of noise control. Some simple charts for the barriers of various bodies are found by experimental and theoretical studies. It has been clear that the integral equation method is useful to predict the noise reduction by a semi-transparent barrier.

1. Introduction

The acoustic shielding may be achieved not only by a screen but also by many obstacles or barriers such as buildings, earth berms, or terrain that blocks the line of sight from the source to the observer. But the acoustical design of a barrier is not very easy due to the difficulty in the calculation of sound diffraction around the barrier. In this paper simple estimation methods for various noise barriers are reviewed from the practical point of view. Although the rigorous solutions of sound diffraction are presented by many authors, the simple and pure conditions which are needed for the rigorous solutions can scarcely be found in practice in citu. Therefore rough estimation methods are still useful in the practice of noise control.

2. Half-infinite thin screen for a point source

When a half infinite plane screen exists between a point source S and a receiver, P in free field, the well known chart in Fig. 1 is the simplest and most reliable method to obtain noise attenuation with reasonable accuracy, though the results generally have values lower by a few dB than those KIRCHHOFF'S approximate theory as shown in the Fig. 1.

In the region of $N \geqslant 1$, the attenuation is expressed by

$$[Att]_{1/2} = 10 \log (20\, N) \quad [\mathrm{dB}]. \tag{1}$$

It is proved that this expression is the first term of an asymptotic formula derived from the exact theory of diffraction by Keller. 2)

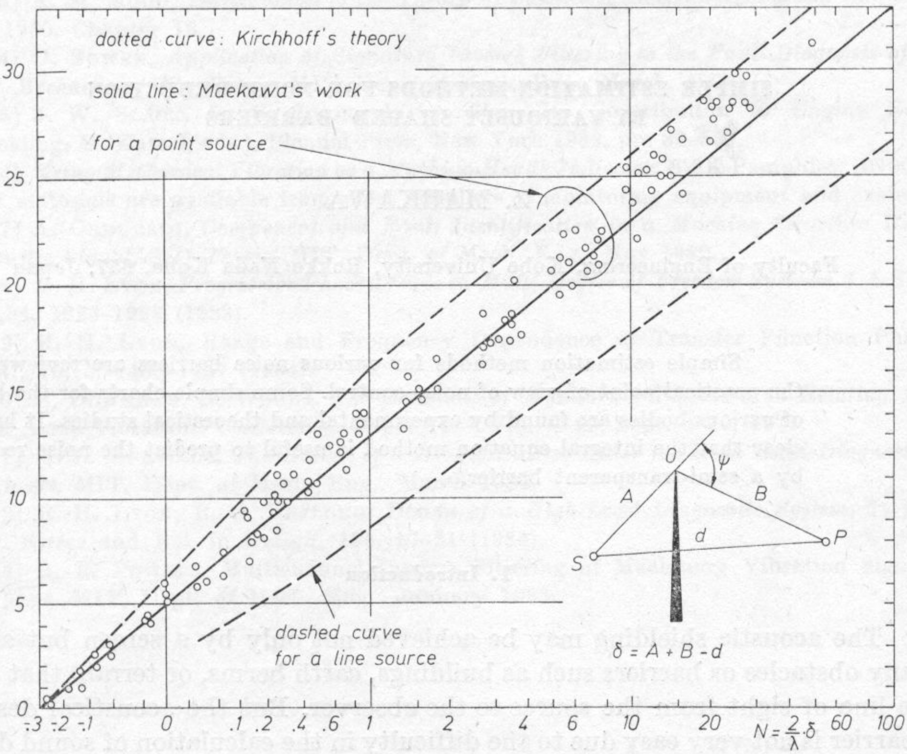

Fig. 1. Sound attenuation by a semi-infinite screen in free space. Horizontal scale, logarithmic scale in the region of $N > 1$, adjusted so that the experimental curve becomes a straight solid line in the region of $N < 1$. Depending on whether $N < 0$ or $N > 0$, the receiving point P lies in the illuminated region or in the geometrical shadow, respectively. Attenuation is relative to propagation in free space. o-experimental values measured by pulsed tone

For the entire range of N, Formula (2)

$$[Att]_{1/2} = 5 + 20 \log \frac{\sqrt{2\pi |N|}}{\tanh \sqrt{2\pi |N|}} \quad [\mathrm{dB}] \tag{2}$$

is convenient for calculation with the aid of a computer, though it has a small discrepancy, within 1.5 dB only in the range $N < 1$. 3)

The variable N is calculated with the value of δ, the path length difference, which may be simply obtained by geometry.

3. Simple estimation of the effect of ground reflection

When the long screen WO is erected on the ground between S and P, and distances between them are not as long as shown in Fig. 2, the sound pressure level at P can be predicted according to the following process:

(1) The sound level L_0 at the top of the screen is asigned as the reference value of the sound level at any point in the shadow zone of the screen. By this procedure both the directivity of the noise source and the reflection from the ground between S and the screen can be at a certain approximation neglected.

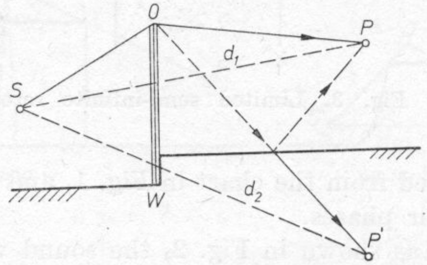

Fig. 2. Section of the long wall between a sound source S and a receiving point P

(2) The effect of ground reflection is calculated by the summation of the sound energy received at P and P', the image of receiver P, assuming perfect specular reflection on the ground and neglecting their phases. If the summation is expressed by L_3 in the level of attenuation, the sound level at P with the screen is obtained by

$$L = \left(L_0 - 20 \log \frac{d_1}{\overline{SO}}\right) - L_3 \quad \text{[dB]}. \tag{3}$$

(3) The shielding effect of the screen, however, must be obtained by the expression $(L_p - L)$ dB, where L is the value calculated by the method mentioned above and L_p is the measured value of the sound level at point P when the wall is absent. We call it the insertion loss of the screen, and it is variable owing to both the directivity of the sound source and the reflectivity of the ground.

4. Attenuation by a finite-size screen

In general, even if a wall or screen has any shape, the sound level in the shadow zone of the screen must be integrated from all the contributions from the open surface.

In the simple case where the length of the semi-infinite screen in free space is limited on both sides, as shown in Fig. 3, the open surface should be divided into three zones [A], [B], and [C]. Zone [A] is then a half-infinite empty plane, and both [B] and [C] are quarter-infinite zones.

The contribution of a half-infinite open surface, of course, can be obtained by the chart in Fig. 1. The contribution of a quarter-infinite open surface can be obtained by summing the two attenuation values of semi-infinite screens, according to the Fresnel-Kirchhoff's diffraction theory [1]. These values, of co-

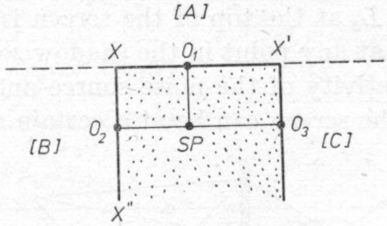

Fig. 3. Limited semi-infinite screen

urse, are easily obtained from the chart in Fig. 1, and added together with its energy neglecting their phases.

In the same way as shown in Fig. 2, the sound wave reflected from the ground is calculated at the point P', the image of P as shown in Fig. 4, conside-

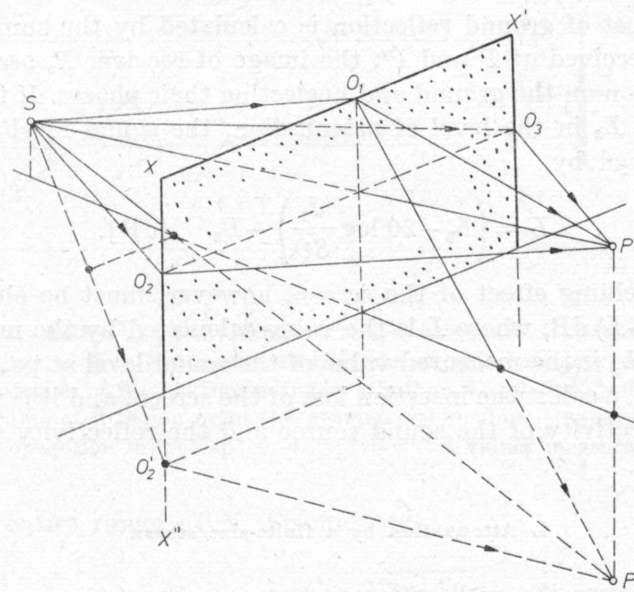

Fig. 4. Propagation paths around the finite size screen

ring the specular reflection of the ground. The sound energy at the receiving point P and that at the image P' should then be added together.

When a thin screen has a complex multilateral shape, the sound level in the shadow zone of it can be obtained by the same principle, as you can find in the literature [4].

5. Simple estimation for barriers of various bodies

Until now, the screen was assumed to have zero thickness. The barriers, however, have their own bodies as shown in Fig. 5. To calculate the noise reduction in the first approximation, we use the chart in Fig. 1 with the value of the path difference,

$$\delta = \overline{SO}+\overline{OP}-\overline{SP} \quad \text{or} \quad \delta = \overline{SX}+\overline{XY}+\overline{YP}-\overline{SP}.$$

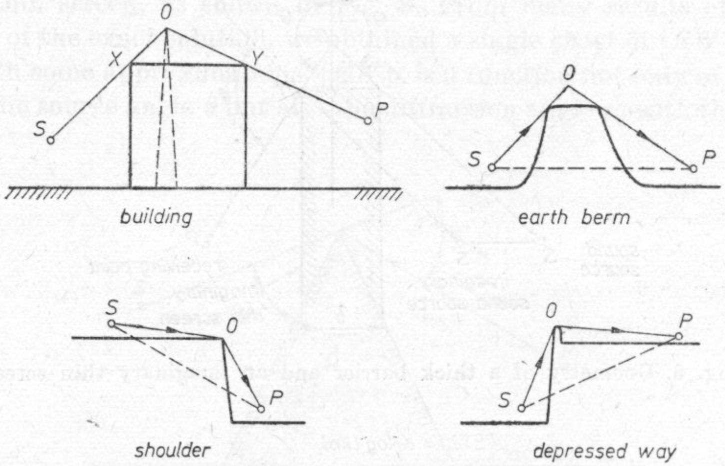

Fig. 5. Noise reduction can be obtained by Fig. 1 by using the path difference of each case

This is the simplest way, though there are more accurate methods as follows:

(*A*) *Effect of the thickness of a screen* [5].

According to many experimental data, the effect of the thickness of a screen should be negligible as long as the thickness is smaller than the wave length. A thick plate or wide barrier has two edges which increase the noise reduction by double diffractions.

The attenuation of a band noise by a thick barrier ${}_n[Att]_b$ is assumed to be composed of the attenuation by an imaginary thin screen ${}_n[Att]_0$ and the effect of the thickness of the barrier ${}_n[ET]_b$ expressed by

$${}_n[Att]_b = {}_n[Att]_0 + {}_n[ET]_b \quad [\text{dB}]. \tag{4}$$

Fig. 6 shows the geometry of a thick barrier and an imaginary thin screen. S and P are the sound source and the receiving point, respectively. SO is parallel to $S'Y$ and SS' is parallel to XY. ${}_n[Att]_0$ is the attenuation of sound by the imaginary thin screen on the path from S' to P passing Y.

After theoretical research, though the computation results of the exact solution show the resonance effect related to the thickness, b, with reasonable approximation, the effect of thickness for a noise having a considerable bandwidth is obtained from

$$_{n}[ET]_{b} = K\cdot\log(kb) \quad [\mathrm{dB}], \tag{5}$$

where K is the value given by the single chart in Fig. 7, and

$$k = \frac{2\pi}{\lambda}. \tag{6}$$

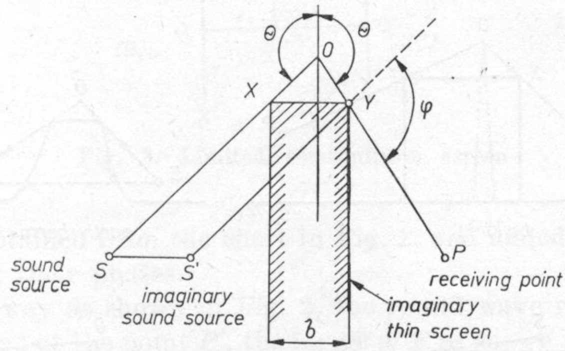

Fig. 6. Geometry of a thick barrier and an imaginary thin screen

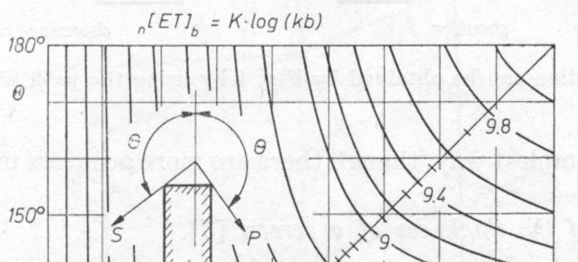

Fig. 7. The values of K in Eqn. (5) to calculate the effect of thickness of a noise barrier

(*B*) *Effect of the angle of a wedge* [*6*].

Many barriers of various bodies, have wedges at their corners, which cause sound diffractions. The theory of the diffraction by a wedge has been treated by many authors. Now, from the practical point of view, the effect of the wedge-angle Ω, which must be added to the attenuation by an imaginary thin screen, in a way similar to that of a thick barrier, is derived from the exact theory, as follows

$$[Att]_\Omega = [Att]_0 + [EW]_\Omega \quad [\text{dB}], \tag{7}$$

where $[Att]_\Omega$ is the noise attenuation by the wege which has wedge-angle Ω, and $[Att]_0$ is the noise reduction (obtained from the chart in Fig. 1) caused by the imaginary thin screen, as shown in Fig. 8. From many results of numerical calculations of the exact solution, we obtained a single chart of $[EW]_\Omega$ as shown in Fig. 9 with some approximations. $[EW]_\Omega$ is a function not only of the wedge-angle and the source angle θ but also the diffraction angle φ; with the notations

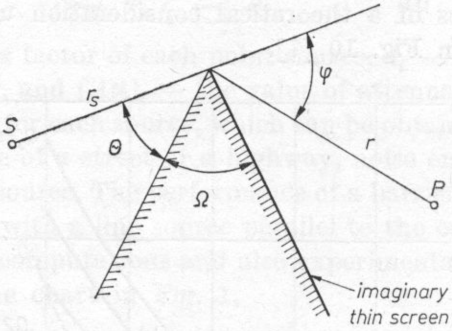

Fig. 8. Notations of the wedge diffraction

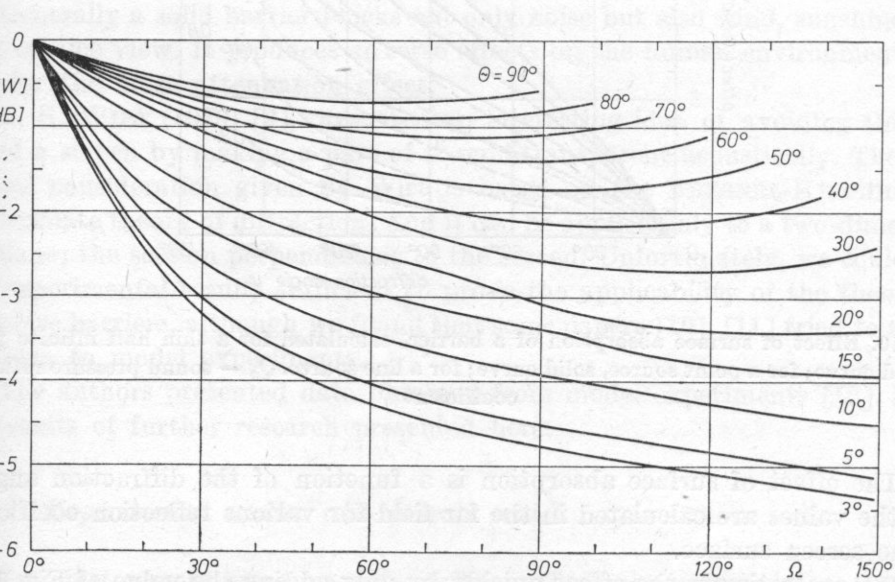

Fig. 9. Effect of wedge-angle Ω with a parameter of source angle θ

as shown in Fig. 8. But the effect of φ can be neglected, and the values of the curves show the negative largest values, in order to be on the safe side for the practical noise control. It is clear that $[EW]_{\Omega}$ decreases the barrier-attenuation from that of the thin screen, but not over -6 dB, and vanishes when $\Omega \to 0$.

Both methods of obtaining the effects of thickness and wedge angle are certified for their usefulness by many experiments.

6. Effect of surface absorption of the barrier

All barriers discussed above are assumed to have a rigid surface. But, the barriers treated by sound absorbing materials are widely used. The effect of surface absorption of a half infinite screen is given by using the exact theory of diffraction. The results of a theoretical consideration under some simplified condition are shown in Fig. 10 [7].

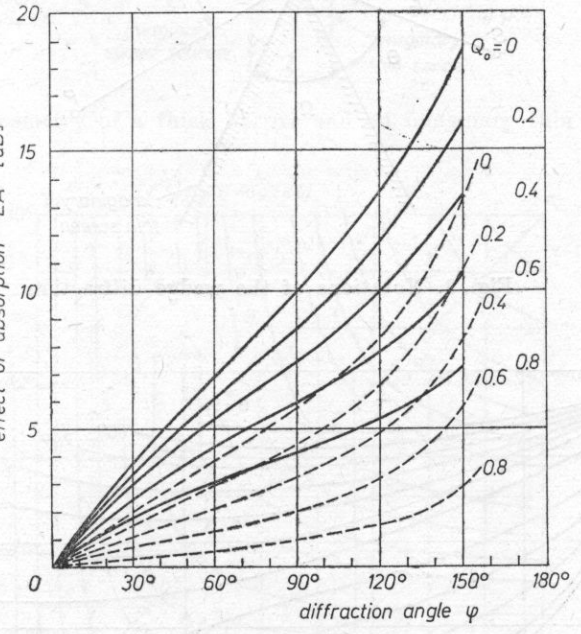

Fig. 10. Effect of surface absorption of a barrier, calculated for a thin half infinite plane. Dotted curve; for a point source, solid curve; for a line source. Q_0 — sound pressure reflection coefficient

The effect of surface absorption is a function of the diffraction angle φ, and the values are calculated in the far field for various reflection coefficients of the screen surface.

We can estimate the effect quickly by only adding the value of Fig. 10 to the value of attenuation by the reflective barrier.

7. Large extended noise source

It is a more difficult problem to obtain a theoretical solution of sound diffraction with a large source, because the wave front from the large source cannot be expressed exactly. There is a conventional method, however, if the large source can be replaced by one or more point sources. When the noises are emitted incoherently from virtual point sources, the sound energy received from each point source, which can be obtained as mentioned above, should be added together at the receiving point.

So that the shielding effect of the barrier for a group of sources can be expressed as follows [8]

$$[Att] = 10\log\left\{\sum_{i=1}^{n}\frac{K_i}{d_i^2}\Big/\sum_{i=1}^{n}\frac{K_i}{d_i^2}\log^{-1}\frac{-[Att]_i}{10}\right\} \quad [\mathrm{dB}], \tag{8}$$

where K_i – the power factor of each point source, d_i – the distance from each source to the receiver, and $[Att]_i$ – the value of attenuation due to the barrier at the receiving point for each source, which can be obtained as described above.

For a special case of a street or a highway, noise emission is often treated as an incoherent line source. The performance of a barrier against highway noise should be considered with a line source parallel to the edge of the barrier. The results of theoretical computations and also experimental studies are shown by a dashed-curve in the chart in Fig. 1.

8. Study of a design method of semi-transparent noise barriers

Generally a solid barrier blocks not only noise but also wind, sunshine and sight or nice view. It produces adverse effects on the human environment, except for the noise attenuation effect.

L. S. Wirt (1979) [9], presented an interesting idea of avoiding this defect of a screen by making a part of it semi-transparent acoustically. The theoretical consideration given by Wirt is based on the Fresnel-Kirchhoff's approximate theory of diffraction. And it can be applied only to a two-dimensional plane; the section perpendicular to the screen. Unfortunately, we could not find experimental results sufficient to prove the applicability of the theory to real noise barriers, although we found that some papers [10], [11] tried to apply this idea to model experiments.

The authors presented data obtained from model experiments [12]. Now, the results of further research presented here.

A) Experimental studies with a scale model

Two test screens were made from 3 mm thick aluminum plates. One has a straight line knife edge and the other has indentations in a saw-teeth shape, as shown in Fig. 11. The screen was set in an anechoic room to fill up the width of it.

A point source was located at a 0.3 m distance from the screen and was emitting 1/3 Oct. band noises. The sound pressure level distributions in the steady

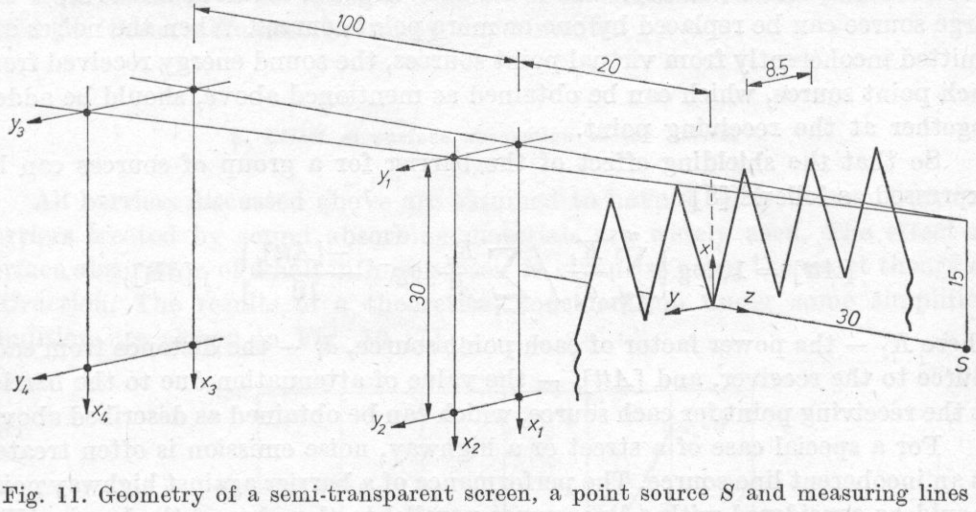

Fig. 11. Geometry of a semi-transparent screen, a point source S and measuring lines

state were measured by a 1/2 condencer-microphone moving on eight lines, $x_1 - x_4$ and $y_1 - y_4$, in the shadow zone of the screen as shown in Fig. 11. The results of the experiments are compared with the theoretical values in Fig. 13.

B) Theoretical calculations

The Helmholtz–Kirchhoff integral theorem gives the integral formula for the sound field of a point source with the approximate boundary condition, that

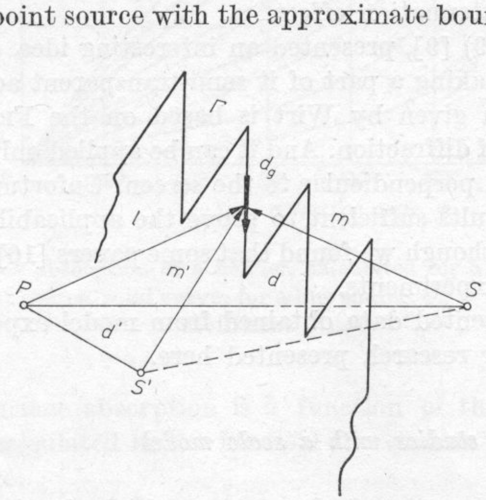

Fig. 12. Geometry and notation for Eqn. (9)

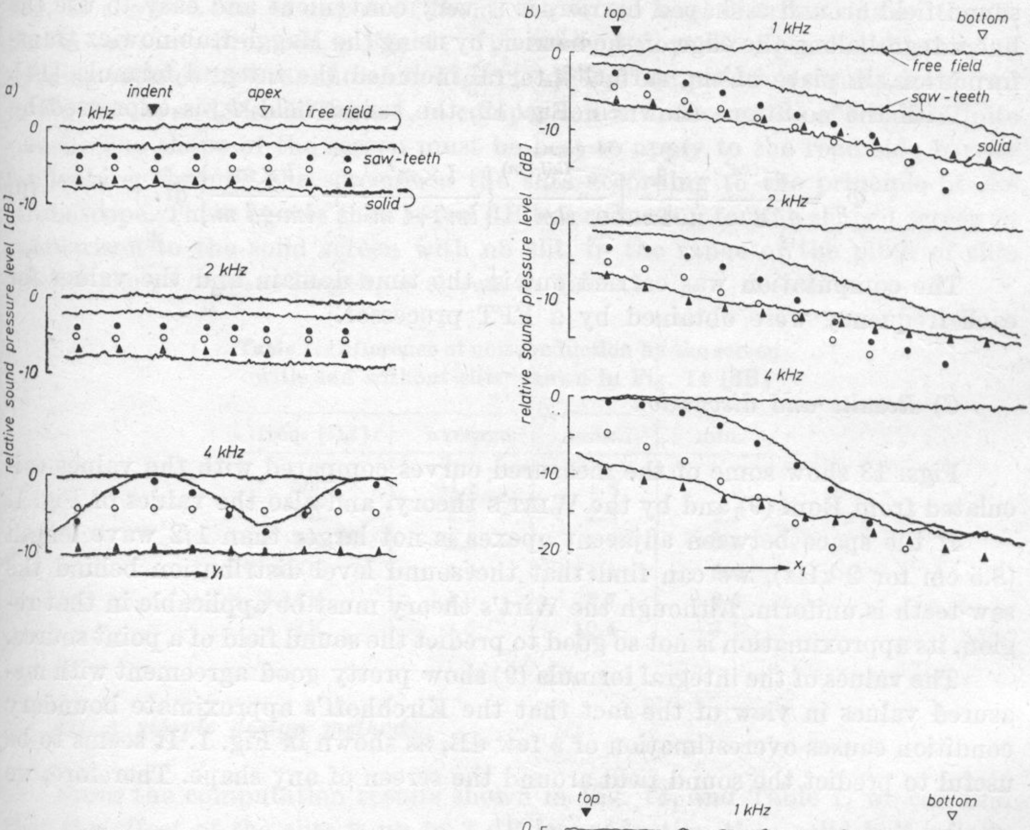

Fig 13. Experimental results, ⁓ – measured, and calculated values, ● – by the line-integral Eqn. (9), ○ – by WIRT's theory, ▲ – by the chart Fig. 1

is the velocity potential on the back surface of the screen is zero [13]. For the sound field around a shaped barrier, it is very convenient and easy to use the line integral along the edge of the barrier, by using the Maggi–Rubinowicz transformation, in place of the surface-integral included the integral formula [14].

With the notations shown in Fig. 12, the sound field Φ_p is expressed by

$$\Phi_p = \frac{e^{-ikd'}}{d'} + \frac{1}{4\pi}\int \frac{e^{-ik(l+m)}}{lm}\left\{\frac{l\times m'}{lm'+l\,m'} + \frac{l\times m}{lm+l\,m}\right\} dt. \tag{9}$$

The computation was carried out in the time domain and the values for each frequency were obtained by a FFT processor.

C) *Results and discussion*

Figs. 13 show some of the measured curves compared with the values calculated from Eqn. (9) and by the Wirt's theory, and also the values of Fig. 1.

If the space between adjacent apexes is not larger than 1/2 wave length (8.5 cm for 2 kHz), we can find that the sound level distribution behind the saw-teeth is uniform. Although the Wirt's theory must be applicable in that region, its approximation is not so good to predict the sound field of a point source.

The values of the integral formula (9) show pretty good agreement with measured values in view of the fact that the Kirchhoff's approximate boundary condition causes overestimation of a few dB, as shown in Fig. 1. It seems to be useful to predict the sound field around the screen of any shape. Therefore, we

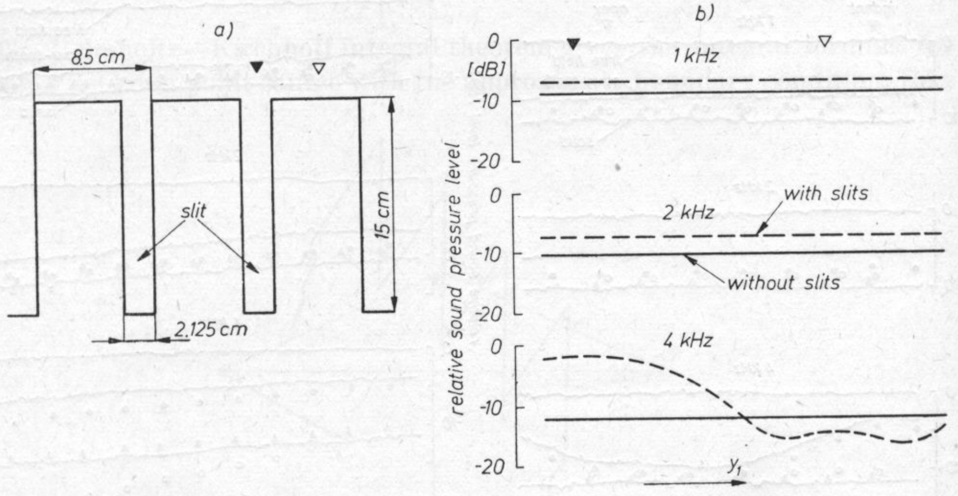

Fig. 14. Semi-transparent screen having 25 % opening ratio, (a) shape of the screen (b) sound level distribution on the y_l line in Fig. 11, calculated by the line integral Eqn. (9) — shaped screen, — solid half-infinite screen, refer to the value in free field

have computed the sound field in the shadow zone of several shaped screens. Fig. 14 shows a few examples of them, the sound level distributions on the Y_1 line in the same situation in Fig. 11. The screen has vertical parallel slits having opening ratio of 25 %, as shown in Fig. 14 (a). The computation results are shown in Fig. 14 (b) and also in Table 1, in comparison with the results of a half-infinite screen. The shape of the screen must be best to apply to the road-side barrier for looking through the screen via the slits according to the principle of the stroboscope. These curves show a few dB less reduction for the shaped screen in comparison to the solid screen with no slit, in the range of the pitch of slits smaller than 1/2 wave length, as shown in Table 1.

Table 1. Difference of noise reduction by the screen with and without slits, shown in Fig. 14 [dB]

freq. [Hz]	average	max.	min.
500	1.8	1.8	1.8
1K	2.2	2.3	2.1
2K	2.9	3.2	2.6
2.5K	3.4	4.4	2.2
3.15K	4.0	5.7	2.6
4K	3.5	10.4	−3.1

D) A simple design method

From the computation results shown in Fig. 14, and Table 1, we can find that the effect of the slits is up to 3 dB less reduction than solid half-infinite screen. The noise reduction of the solid half-infinite screen can be simply obtained by the chart in Fig. 1. Therefore, we can easily estimate the noise reduction by a semi-transparent screen with parallel slits, having 25 % opening ratio, by subtracting the average values of Table 1, provided that the pitch of slits is smaller than 1/2 wave length.

9. Conclusion

From the practical point of view, several simple methods of estimating noise reduction by various by shaped barriers are reviewed. These simple methods do not need the usage of an computer, but of course, they can be easily applied to a computer program [15].

The new application method of the simple chart to design a semi-transparent barrier is proposed. It has been clear that the line-integral formula is useful to predict the noise reduction and to obtain the correction terms for the simple chart to design the variously shaped barriers.

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SOME CONSIDERATIONS ON COMMON NOISE CRITERIA

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Until now the ideal noisedescriptor has not been invented. The purpose of this paper is to draw the attention of the user to some shortcomings of common noise criteria in extreme circumstances. Comments are made on the A-weighted sound pressure level L_A, the continuous equivalent sound pressure level $L_{A\text{eq}}$, the sound exposure level L_{AE}, and the percentile levels L_{AN}. These comments do not affect the usefulness of these descriptors.

1. The A-weighted sound pressure level dB(A).

The sound level meter which is the most simple instrument for noise measurements gives us a lot of possibilities.

A distinction must be made between:

Frequency weighting: dB(A) dB(B) dB(C) dB(D) (see Fig. 1).

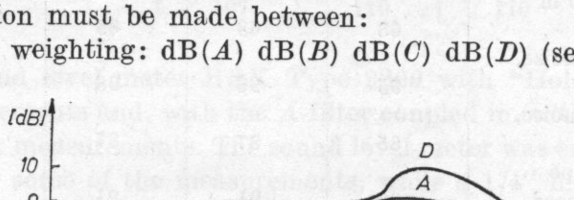

Fig. 1. Frequency weighting of the sound level meter

Time weighting: Slow: 1000 ms
Fast: 125 ms
Impulse: 35 ms (1500 ms)
Peak: 50–100 μs (see Fig. 2)

The difference in the readings will depend upon the choice of the frequency and the time weightings and on the character of the noise as show in Table 1 [1].

A number of measurements have been made in different industries with the

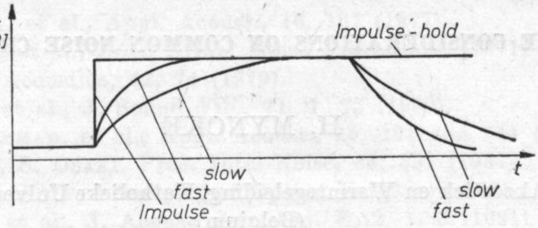

Fig. 2. Time weighting of the sound level meter

Table 1

Sound Source	Fast dB(A)	Imp. dB(A)	Imp. Hold dB(A)5 ×	Peak Hold dB(A)5 ×	Δ
1	2	3	4	5	6
Sinusoidal pure tone 1000 Hz	94	94	94	97	3
Beat Music from a gramophone	90	91	93	97	4
Modern music from a gramophone	102	103	103	105	2
Electric quitar from a gramophone	85	86	86	91	5
Motorway traffic, 15 m distance	80	80	81	89	8
Motorway traffic, 50 m distance	68	68	68	76	8
Train 70 km/h rail noise, 10 m distance	95	96	98	106	8
Train 70 km/h rail noise, 18 m distance	85	87	87	94	7
Noise in aircraft Type PA 23, cruising speed	90	91	91	100	9
Noise in aircraft Type Falco F 8, cruising speed	97	98	98	109	11
Noise in aircraft Type KZ 3, cruising speed	102	102	103	112	9
Noise in car Type Fiat 500, 60 km/h	78	79	79	93	14
Noise in car Type Volvo 142, 80 km/h	75	75	76	86	10
Lawn mower 10 HK, 1 m distance	97	99	99	116	17
Typewriter IBM (Head position)	80	84	83	102	19
Electric shaver, 2.5 m distance	92	92	92	107	15

1	2	3	4	5	6
75 HK diesel motor in electricity generating plant	100	101	101	113	12
Pneumatic nailing machine, 3 m distance	112	114	113	128	15
Pneumatic nailing machine near operator's head	116	120	120	148	28
Industrial ventilator 5 HK 1 m	82	83	83	93	10
Air compressor room	92	92	92	104	12
Large machine shop	81	82	82	98	16
Turner shop	79	80	81	100	19
Automatic turner shop	79	80	80	99	19
40 tons Punch press, near operator's head	93	98	97	121	24
Small automatic Punch press	100	103	103	118	15
Numerically driven high speed drill	100	102	103	112	9
Small high speed drill	98	101	101	109	8
Ventilator with filter	82	83	83	94	11
Machine driven saw, near operator's head	102	102	104	113	9
Vacuum cleaner Type Hoover, 1.2 m distance	81	81	81	93	12
Bottles striking each other	85	88	90	105	15
Bottling machine in brewery	98	99	101	122	21
Toy pistol (cap)	105	108	108	140	32
Pistol 9 mm, 5 m distance from side	113	114	116	146	30
Shotgun, 5 m distance from side	108	110	111	143	32
Saloon rifle, 1 m distance from side	107	110	110	139	29

use of the sound level meter B&K Type 2209 with "Hold" circuit for peak voltage measurements and, with the A-filter coupled in, with an averaging time of 30μs for peak measurements. The sound level meter was equipped with a 1/2″ microphone for some of the measurements, while a 1/4″ microphone was used for both high levels and the sharpest peaks. The 1/2″ microphone has a resonance frequency around 18 kHz but can be used up till 38 kHz as it is damped. The 1/4″ microphone has a resonance frequency around 34 kHz and can be used up till 65 kHz. The sensitivity was adjusted using a pistonphone as is customary.

The results are shown in Table 1. The measurements were all taken according to dB(A) "Fast" (125 ms), dB(A) "Impulse" (35 ms), and dB(A) "Imp. Hold" in which case the reading noted was the mean of 5 values measured with approximately 10 second intervals. Finally measurements were also taken with dB(A) "Peak Hold" (30 μs) with 5 to 10 second intervals and the mean of 5 measured values noted. The most interesting aspect is to ascertain how large the "Peak" value is above dB(A) "Fast" or dB(A) "Impulse". It is denoted by Δ in the Table. The larger the difference the more dangerous the noise. A pure sinusoidal tone has the same values for "Fast" and "Imp. Hold" while the "Peak" value should be and is 3 dB higher. It can be seen that beat music and other electronic music has very low peak values; the same is true for noise in aircraft and a number of (especially high speed) machine tools and woodworking machines. Larger differences are revealed by lown mowers, type writers, and naturally all types of percussion machines such as pneumatic nailing machines, bottling machines (bottles clattering against each other) and punch presses. Obviously direct impacts, gunshots and explosions manifest the largest differences. The dB(A) weighting curve corresponds more or less to the 40 phon equal loudness contour.

Mathematically we have the following relation:

$$P = a + b\, L_p + c\, L_p^2,$$

with L_p — the sound pressure level in dB of a pure tone with frequency f in Hz,

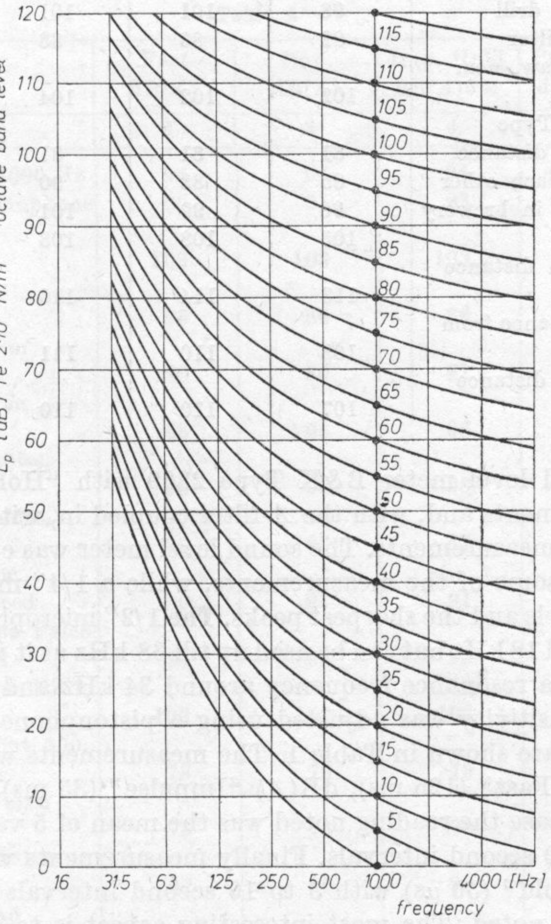

Fig. 3. Noise rating curves

P – the equal loudness level in phons, a, b, c – parameters depending upon the frequency.

Although the dB(A) is a very useful noise descriptor, one has to keep in mind that it is not always valid.

1.1. In the first example we make a comparison with the noise rating curves NR, given in the annex of ISO R 1996, which are commonly used in some countries (see Fig. 3). They correspond to the following equation:

$$L_p = a + b \times NR.$$

Again the parameters a and b depend upon the frequency as mentioned in the standard. Fig. 4 gives a comparison between the inverse of the dB(A)

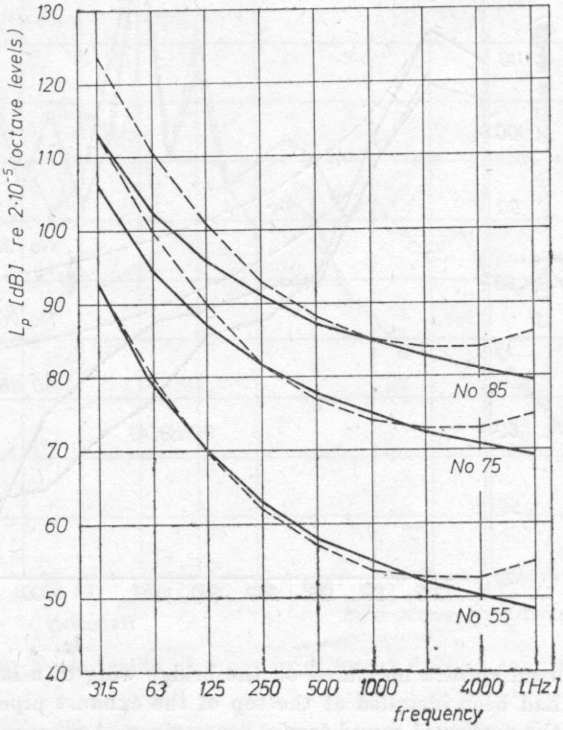

Fig. 4. A comparison between the inverse of the dB(A) weighting curve (dashed) and some NR-curves

weighting curve and some NR-curves [2]. In the case of a broad band noise we have

$$\mathrm{dB}(A) \approx NR + 5.$$

A noise with a sound pressure level of 85 dB in the 1000 Hz octave band and with appreciably lower levels in the other octave bands gets the NR-number 85.

The dB(A) reading is also 85 for there is no attenuation in this frequency region. There is full agreement between the two systems in this case.

Another noise, now having 113 dB in the 31.5 Hz octave band and less than 60 dB in the other ones, gets also the NR-number 85, which should mean that the annoyance caused by the two sounds is equal.

The dB(A) meter reads only 74 dB, as can be seen in Fig. 4, because the attenuation for this frequency band relative to 1000 Hz is 39 dB. A large difference in acceptability for the two noises is shown by the dB(A) rating, but the same rating is given for both if the NR-curves are used.

Conversely, two signals with equal sound levels A but different frequencies may show a difference of 10 NR-numbers. Figure 5 illustrates this possibility.

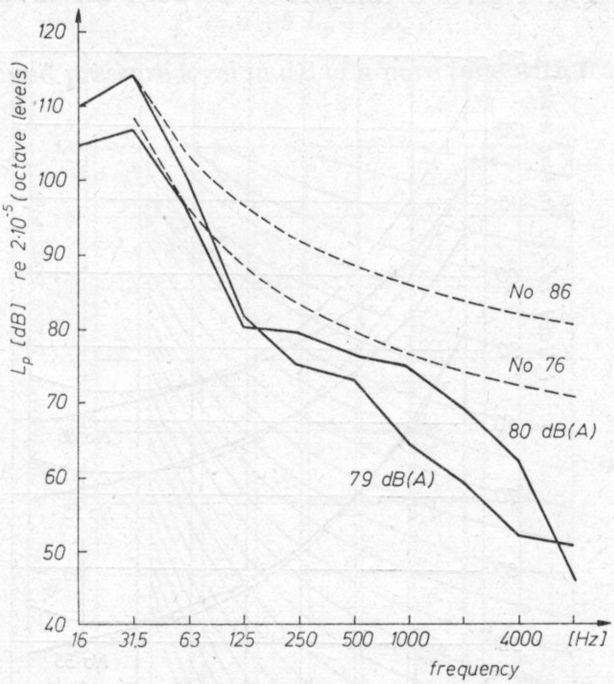

Fig. 5. Two octave band spectra measured on the bridge wing of a large motorship before and after a silencer had been installed at the top of the exhaust pipe of the main engine. Next to the spectra the measured sound levels A are given. A remarkable reduction of the noise annoyance had been noticed after installing the silencing construction (thick line)

Two octave band spectra measured on the bridge wing of a large motorship before and after a silencer had been installed at the top of the exhaust pipe of the main engine:

	NR	dB(A)
without silencer	86	79
with silencer	76	80

From this example we may conclude that in the case of high level noises (e.g. in wheelhouses and on bridge wings of motorships) a great disagreement between the two rating methods may occur. Moreover the *NR* system will give a better correlation between the physical measurements and the acoustical comfort.

1.2. The second example emphasises the advantage of the use of loudness expressed in sones for the determination of the acoustical comfort inside a motorcar (Fig. 6) [6]. Different driving conditions give the same dB(A) value but a difference of 10 sones in loudness, respectively 53 and 43 sones.

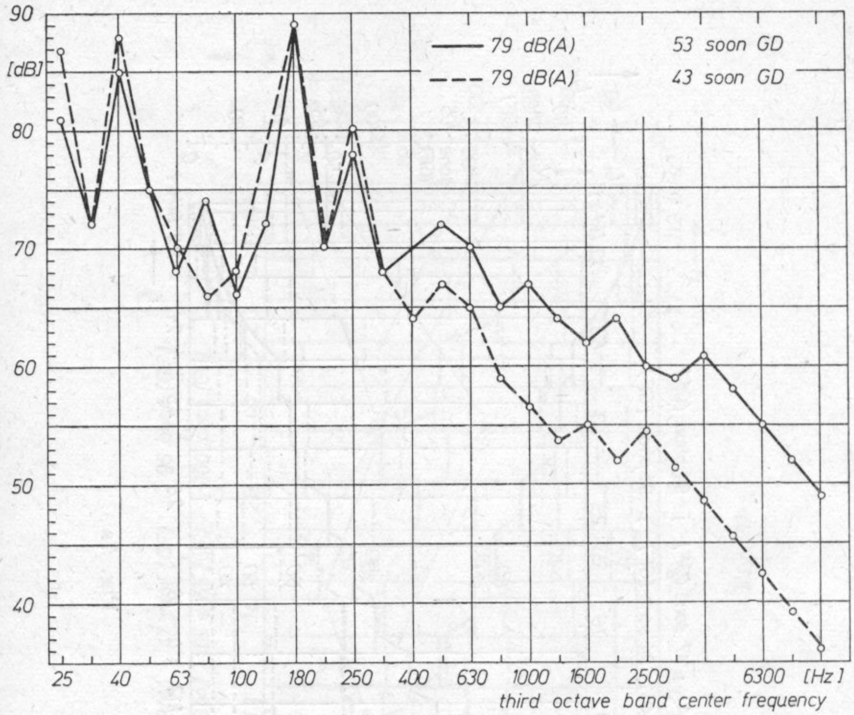

Fig. 6. Noise inside of a car in different driving conditions

1.3. The third example (see Fig. 7) [7] concerns the noise produced by a motorcycle. A comparison is made between the variation of dB(A), phone (GF) and sone in the following conditions:

Two motorcycles with a similar noise spectrum but at a different distance. There is a good agreement between the three values.

Two motorcycles with different spectra. There is no variation in the dB(A) reading.

Two extreme case with a spectrum giving a lower dB(A) value but a higher loudness in sone.

The different results are given in the Table at the right side of the spectra.

ΔL_A dB(A)	ΔL_{NGF} phon (GF)	ΔN sone
identical spectrum Gelijkaardig spectrum		
-6	5	-13 - 28%
(87–81)	(96–91)	(47–34)

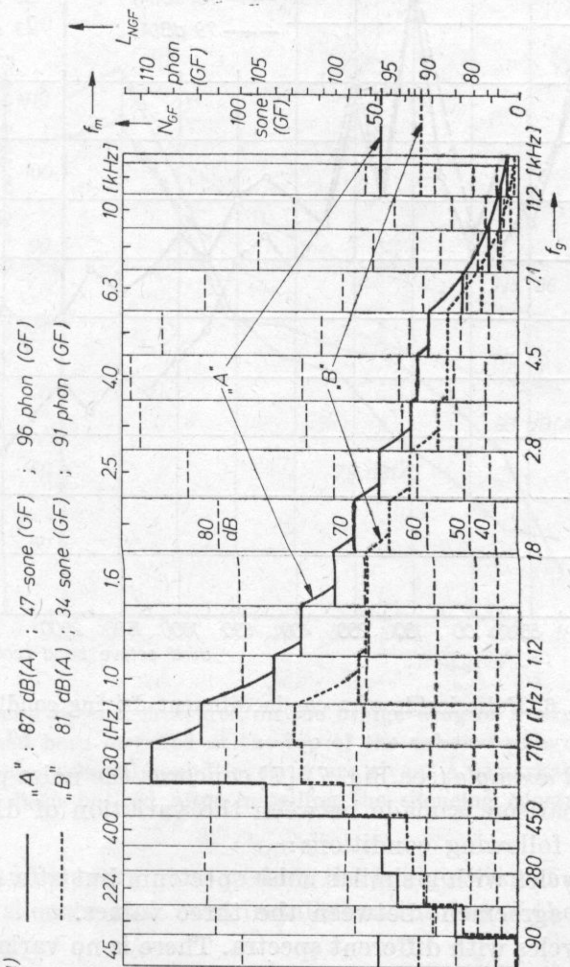

Fig. 7a

ΔL_A dB(A)	ΔL_{NGF} phon(GF)	ΔN sone
same L_A - broader spectrum Zelfde L_A - breder spektrum		
0	+4	+18 38%
(87→87)	(96→100)	(47→65)

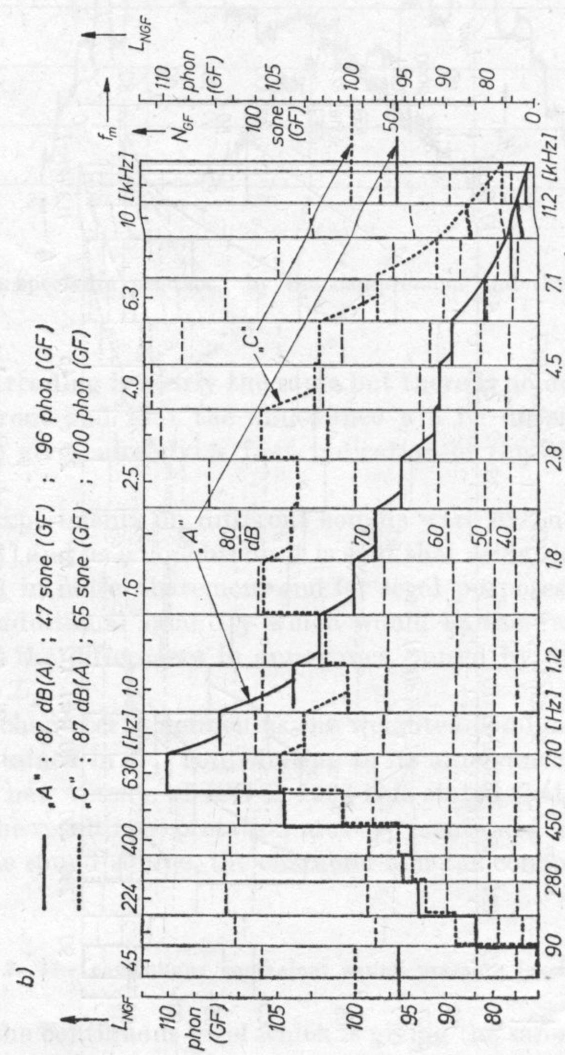

Fig. 7b

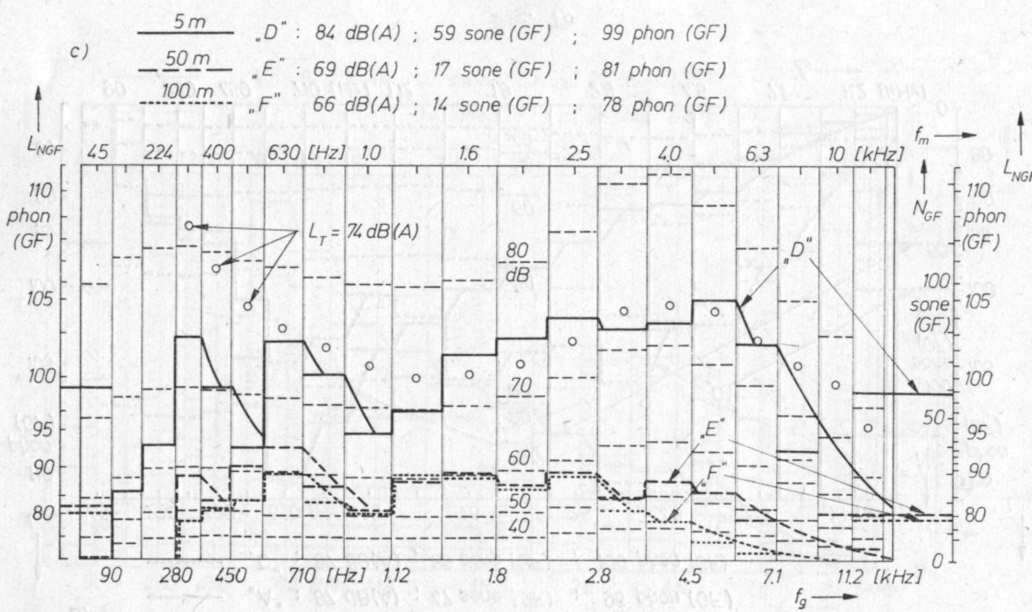

Fig. 7a-c. Spectra of motorcycles

ΔL_A dB(A)	ΔL_{NGF} phon(GF)	ΔN sone
lower L_A - encreased spectrum at higher freq. Lager L_A - naar hoge frekwen. verbreed spek.		
-3 (87→84)	+3 (96→99)	+12 26% (47→59)

1.4. The character of the sound

Fig. 8 shows the spectra of two different sounds (compression and expansion of air), presenting a maximum at respectively 100 and 10 000 Hz.

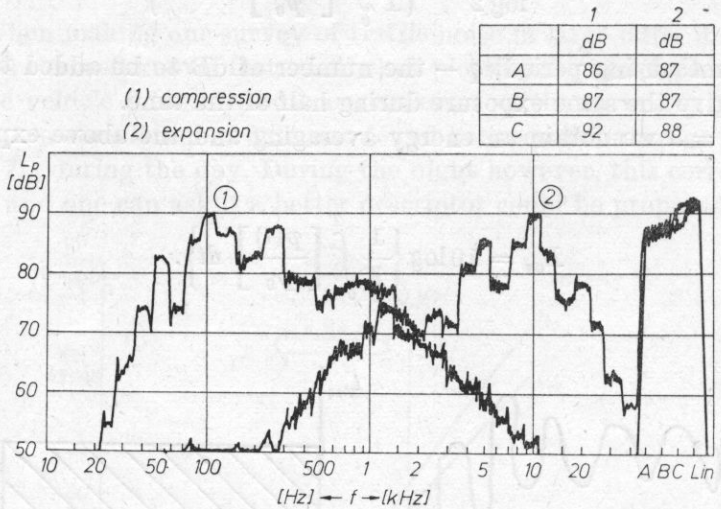

Fig. 8. Noise spectrum produced by the compression and the expansion of air

The dB(A) reading is nearly the same but there is no doubt that both sources sound different and that the annoyance will be different. The difference dB(C) − dB(A) gives already a first indication of the different character of both noises.

Listening experiments on different sounds were executed at the Institute of Perception [3] and as a conclusion it is said that aside the L_A readings being widely accepted in noise abatement and for legal purposes, there seems to be a need for an additional quantity which would be the "sound character", to take account of the differences in annoyance caused by different sounds that have the same L_A.

The sound character is defined as the weighted combination of all acoustic factors, not contained in L_A, contributing to its annoyance.

Also in the new version of ISO R 1996 it is stated that data which may be significant for the result interpretation must be mentioned in the report, namely the nature of the sound source, the character and the connotation of the sound.

2. The continuous equivalent sound pressure level L_{eq}

The L_{eq} is the continuous level which is giving the same exposure as a fluctuating noise during the observation period (Fig. 9). The most general formulae is the following

$$L_{\mathrm{eq}} = \frac{q}{\log 2} \log\left\{\frac{1}{T}\int_0^T \left[\frac{p(t)}{p_0}\right]^{20\log 2/q} dt\right\},$$

where T — measuring period, q — the number of dB to be added to the noise in order to give the same exposure during half of the time.

With $q = 3$ we obtain an energy averaging and the above expression becomes

$$L_{\mathrm{eq}} = 10\log\left\{\frac{1}{T}\int_0^T \left[\frac{p(t)}{p_0}\right]^2 dt\right\}.$$

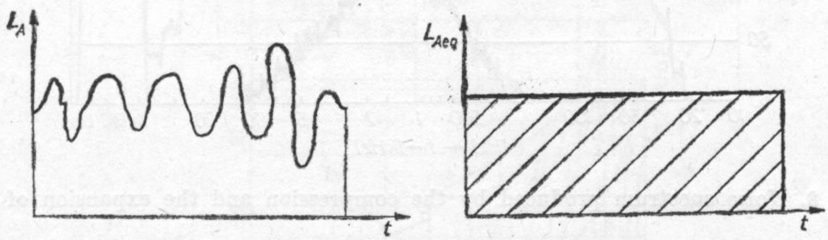

Fig. 9. Continuous equivalent sound pressure level (A-weighted or not)

The L_{eq} as a noise descriptor has been widely used during last years. In the new version of ISO R 1996 its use is highly recommended. A typical example of its usefulness is given in Table 2 where some idealized sounds are expressed in L_{eq} and in percentile levels L_N. From the last column we see that the L_{eq} varies over a range of 40 dB allowing a good measurement of different sounds.

People investigating annoyance know however, that the correlation between

Table 2. Noise descriptor values for various idealized sounds

Nature of the sound	L_1	L_{10}	L_{50}	L_{eq}
Steady sound, 40 dB	40	40	40	40
Steady sound, 40 dB, except 80 dB, 0.2 percent of time (3 min/24 h)	40	40	40	53
Steady sound, 40 dB, except 80 dB, 2 percent of time (30 min/24 h)	80	40	40	63
Steady sound, 40 dB, except 80 dB, 20 percent of time (5 h/24 h)	80	80	40	73
Steady sound, 80 dB, 100 percent of time	80	80	80	80

the physical value of the L_{eq} and the annoyance score is not always perfect. In some cases other descriptors have to be added or are to be used. Again we let follow some examples.

2.1. When making our survey of traffic noise in large cities in Belgium [4] we observed the well known fact that, when plotting L_{eq} and L_{10} levels as a function of the vehicle intensity, both curves cross as shown in Fig. 10. Above 20 vehicles/hour a correlation of 0.86 was found between the average factor scores and L_{eq} or L_{10} during the day. During the night however, this correlation is far under 0.20 and one can ask if a better descriptor could be proposed.

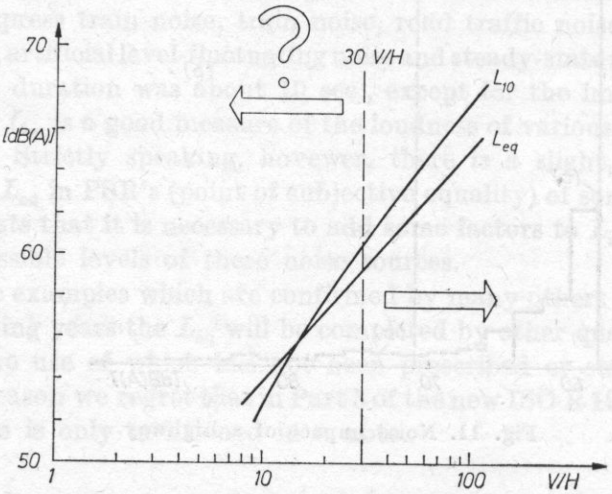

Fig. 10. Variation of L_{eq} and L_{10} as a function of traffic intensity (vehicles per hour)

2.2. Another example is this of the determination of the impact of a highway on the acoustical environment of a community in the neighbourhood. For an exact determination a distinction must be made between:

a) the background level in the community,

b) the level due to the local traffic,

c) the level due to the presence of the highway.

In order to determine exactly the impact of the highway on the acoustical environment in the community we have to eliminate $L_{eq}(b)$ from our measurement and the real impact will be given by the difference $L_{eq}(c) - L_{eq}(a)$ (Fig. 11). We realize that taking the L_{eq} of the background noise and not its lowest value as foreseen in ISO R 1996 may not be accepted by everybody. But in the example we just described it is the only way to give the right answer to the question.

2.3. When using acoustical barriers along a highway the sound attenuation is often expressed in L_{eq}. Plotting the sound level variation as a function of time we observe that behind a barrier the diffderence between the maxima and

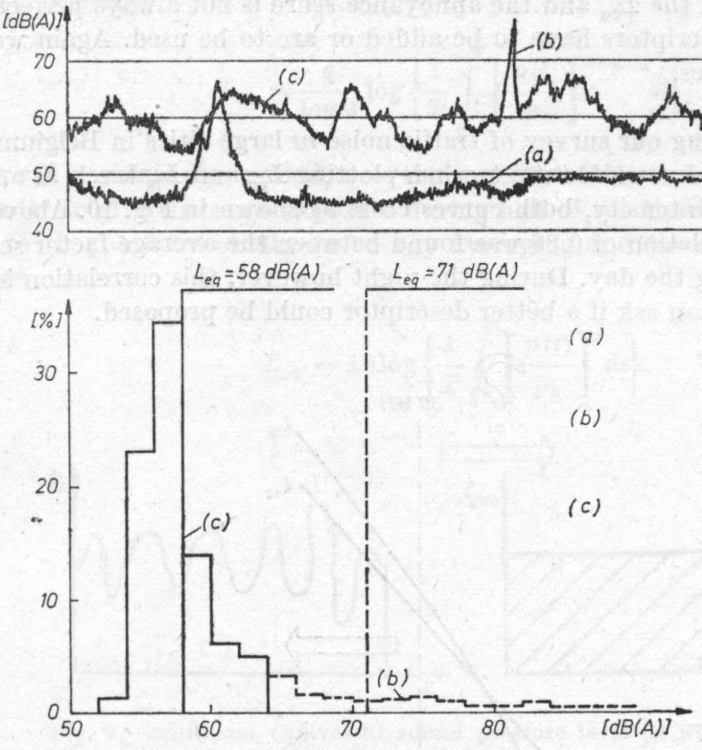

Fig. 11. Noise impact of a highway

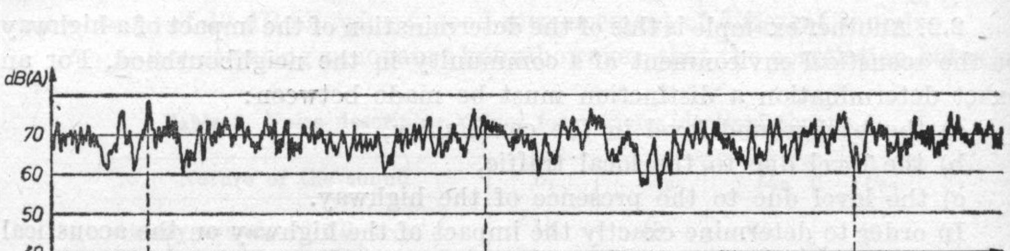

Fig. 12. Variation of the A-weighted sound level as a function of time

the minima will be lower (see Fig. 12). This lower "dynamic" of the noise will also reduce the annoyance. In this case the use of a percentile level such as L_{10} will be very useful.

2.4. In a quite different field, namely this of factory noise, a recent investigation, we made in our city, brought us to the conclusion that the L_{90} level produced by the factory was, in the considered case a better descriptor of the noise impact on the neighbourhood than the L_{eq} level.

2.5. In a recent Japanese study [5] the application of L_{eq} as a measure of the loudness of various noises was investigated. Nine kinds of noise source — aircraft noise, super express train noise, train noise, road traffic noise, speech, music, impulsive noise, artificial level-fluctuating noise and steady-state noise — were used as stimuli. The duration was about 10 sec., except for the impulsive noise. It was found that L_{eq} is a good measure of the loudness of various noises, as a first approximation. Strictly speaking, hovewer, there is a slight, but systematic deviation from L_{eq} in PSR's (point of subjective equality) of some noise sources. This fact suggests that it is necessary to add some factors to L_{eq} in order to decide the permissible levels of these noise sources.

From these examples which are confirmed by many others we can conclude that in the coming years the L_{eq} will be completed by other quantities we know already, but the use of which has not been prescribed or standardised until now. For this reason we regret that in Part 3 of the new ISO R 1996 the use of the percentile levels is only mentioned in a note.

3. The sound exposure level L_{AE}

This descriptor also called the single event exposure level, was firstly introduced in ISO 3891 for the evaluation of aircraft noise. Its use has recently

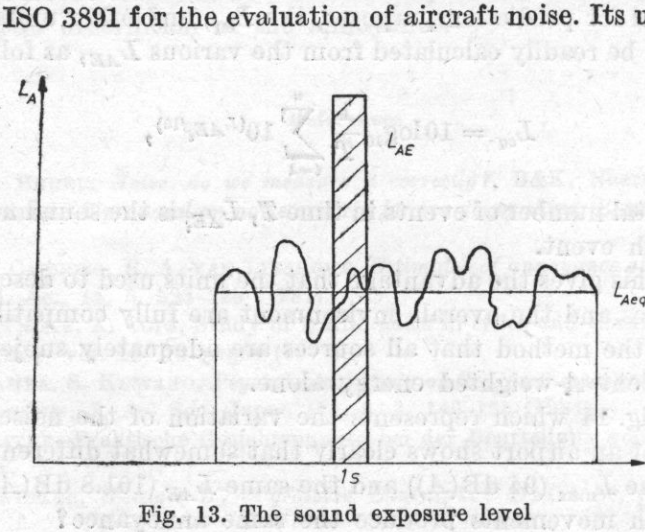

Fig. 13. The sound exposure level

become more general and it is also taken up in the new version of ISO R 1996 in respect to community noise (see Fig. 13).

L_{AE} is defined as the constant level which, if maintained for a period of one second, would deliver the same A-weighted noise energy to the receiver as the actual event itself. This is, then, basically a L_{eq}, which is normalised to a time period of one second. Mathematically we have

$$L_{AE} = 10\log_{10} \int_{-\infty}^{\infty} \left[\frac{p_A(t)}{p_{\text{ref}}}\right]^2 \frac{dt}{\tau_{\text{ref}}},$$

where $p_A(t)$ is the instantaneous A-weighted sound pressure, p_{ref} is the reference pressure, 20 micropascals, τ_{ref} is the reference time, i.e. 1 s.

In practice the following is often used:

$$L_{AE} = 10\log_{10} \int_{t_1}^{t_2} 10^{(L_A(t)/10)}\, dt,$$

where $L_A(t)$ is the instantaneous A-weighted sound pressure level, t_1 and t_2 define the time interval in which the level remains within 10 dB of its maximum during the event.

The usefulness of this concept becomes most apparent when dealing with an environment in which a number of different types of noise events occur. These may differ because of the operating conditions or individual characteristics of the same general type of source, such as aircraft, or the occurrence of two or more totally different types of a noise source.

In either case, the knowledge of the normalised sound exposure level, L_{AE}, for each type of event, further categorised in terms of operating conditions where applicable, has many advantages. When describing any noise environment in terms of the equivalent continuous sound level, L_{eq}, or designing a mathematical model for prediction purposes, the L_{eq}, and other units based on it, such as L_{dn}, can be readily calculated from the various L_{AE}, as follows

$$L_{eq} = 10\log_{10} \frac{1}{T} \sum_{i=1}^{n} 10^{(L_{AE_i}/10)},$$

where n is the total number of events in time T, L_{AE_i} is the sound noise exposure level for the i'th event.

Therefore, this gives the advantage that the units used to describe both the individual sources and the overall environment are fully compatible, although it is implicit in the method that all sources are adequately subjectively rated by their equivalent A-weighted energy alone.

However, Fig. 14 which represents the variation of the noise level in the neighbourhood of an airport shows clearly that somewhat different movements can give the same $L_{\max}$ (94 dB(A)) and the same L_{AE} (101.8 dB(A)). The question is, will both movements produce the same annoyance?

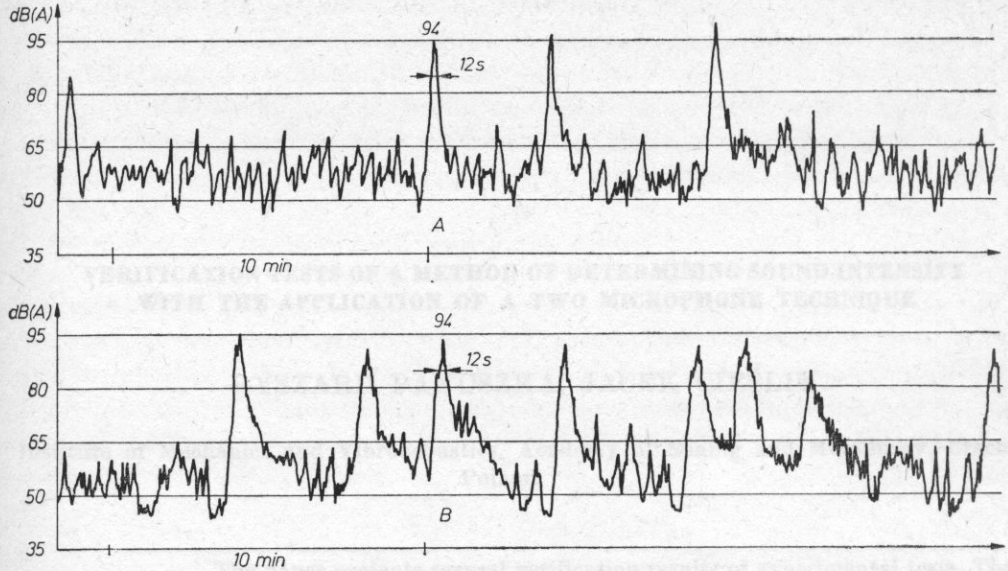

Fig. 14. Variation of the sound level as a function of time in the neighbourhood of an airport. Although the movements A and B are different, they give the same $L_A = 94$ dB(A) and the same $L_{AE} = 101.8$ dB(A)

4. Percentile levels L_N

These levels deduced from the cumulative distribution of a noise give the percentage of the duration then the level exceeds a certain value. In this respect L_1 can serve as an approximation of the average maximum level and L_{95} as an indication of the masking effect of a specific noise under consideration by the residual noise. Both levels can be a useful complement to the L_{eq} value in order to give a better description of the annoyance.

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VERIFICATION TESTS OF A METHOD OF DETERMINING SOUND INTENSITY WITH THE APPLICATION OF A TWO MICROPHONE TECHNIQUE

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The paper presents several verification results of experimental tests. These tests led to the development of software and instrumentation for the method of measuring the acoustical intensity with the application of a two microphone technique. Results of a quantitative evaluation of chosen method errors occuring for the accepted for investigation pair of microphones were presented, as well as results of investigations leading to the evaluation of the accuracy of intensity measurements done with the developed measuring system.

1. Introduction

In the past few years we can observe a development of methods leading to a quantitative evaluation of acoustical energy flow from the source to various zones of the acoustical field of industrial systems. Energetic and statistical methods [1,7] are of special significance, because they allow the quantitative evaluation of the source radiation and the analysis of the probable transfer paths of the acoustical energy. One of the most effective methods are two- and multi-parameter methods, where the mutual correlation or power spectrum density functions are applied. The two microphone method (*acoustical intensity*) is recognized among the simplest. It is based on the acoustical intensity determination from two time functions of the acoustical pressure, measured in two near lying points of the acoustical field. Lastly the possibilities of applying these methods in determining the acoustical power of machines, locating noise sources, measuring the sound attenuation coefficient and acoustical isolation, were investigated. Parallely to the development of the universal sound intensity meters, research is conducted on the improvement of sound intensity estimation. Several research centres are engaged into its implementation in their own laboratories with the application of their computing systems. This paper presents the results of theoretical calculations and experimental research on the development of an acoustical intensity measurement method, applying two microphone probes produced in Poland. An analysis of chosen errors of the method is also presented, as well as results of research on the accuracy evaluation of the intensity measurements done with the developed measuring system.

2. Theoretical foundations of the two microphone method

In currently applied methods of measuring acoustical intensity a principle of simultaneous acoustical pressure measurements in two near lying spots, with a two microphone probe, is put into use. At the same time the two microphone probe is applied for simultaneous measurements of the acoustical pressure and acoustical velocity. The acoustical velocity can be theoretically defined from the Euler equation. For a linear medium without a considerable flow this equation will be

$$\overrightarrow{\operatorname{grad} p} = -\varrho \frac{d\vec{v}}{dt}, \tag{1}$$

where p – acoustical pressure [Pa], $\vec{v}$ – acoustical velocity [m/s], ϱ – medium density [kg/m³].

For a chosen direction, x, equation (1) will be

$$v_x = -\frac{1}{\varrho} \int \frac{\partial p}{\partial x} dt, \tag{2}$$

where v_x – coordinate of vector $\vec{v}$ in direction x.

Assuming that the distance between the microphones is small in comparison to the wave length, we can replace the acoustical pressure gradient by a finite difference

$$\frac{\partial p}{\partial x} \approx \frac{p_2 - p_1}{\Delta x}, \tag{3}$$

where p_1, p_2 – acoustical pressures in the first and second measuring point respectively [Pa], Δx – distance between the microphone membranes [m].

Taking advantage of the acoustical intensity definition and of equation (3), we can determine the approximate value of the acoustical intensity component in direction x

$$I_x = \frac{1}{2\varrho \Delta x} \left\langle (p_1 + p_2) \int (p_2 - p_1) dt \right\rangle, \tag{4}$$

where symbol $\langle \cdot \rangle$ means a mean in time.

There are two main methods of processing time signals of acoustical pressure:

a) analogue,

b) numerical.

A. Analogue method

This method, called also the *sum-difference method*, directly applies equation (4). The practical construction of analogue meters is difficult considering the significant influence of phase shifts occuring during signal filtration, on the measurement accuracy. To avoid this problem certain method modifications are applied; such as using one filtering system and switching measuring channels.

B. Numerical method

The numerical method of determining acoustical intensity applies the Fourier transform of microphone signals. The theoretical analysis shows that the acoustical intensity can be determined from the imaginary part of the cross spectrum of microphone signals

$$I_x(f) = \frac{\operatorname{Im}\{G_{12}(f)\}}{2\pi f \varrho \Delta x}, \tag{5}$$

where $I_x(f)$ — acoustical intensity in direction x [Wm^{-2}], f — middle frequency of measuring band [Hz], ϱ — medium density [$\mathrm{kg/m^3}$], Δx — distance between microphones [m], $G_{12}(f)$ — cross spectrum of microphone signals [Pa^2].

The numerical method of calculating acoustical intensity is based on the determination of the Fast Fourier Transform of the pressure time function and then the cross spectrum function. The constant coefficients in equation (5) differ in dependence of the individual frequency bands. Calculations are done numerically with the application of an analogue-to-digital converter and a computer.

3. Errors of the two microphone method

The accuracy of the acoustical intensity measurements done with the application of the two microphone method depends mainly on the following factors:

distance between the microphones (distance between the membranes or axis of the microphones),

phase shifts developed in the measuring system,

ratio of the microphone signal amplitudes,

linear dimensions of the microphones,

random errors occuring during signal processing.

Among the above mentioned factors the phase shifts have a decisive influence on the measuring accuracy in the range of low and medium frequencies. Numerical methods allow the correction of the error due to phase shifts through introducing constant corrections. The value of an error due to a finite distance between the microphones and the occurrence of the phase shifts can be calculated theoretically. As an example, an equation is given for a monopole source

$$I_t = 10\log_{10}\frac{I_p}{I_d} = 10\log_{10}\left[\frac{\sin(k\Delta x+\varphi)x^2}{k\Delta x\, x_1\, x_2}\right], \tag{6}$$

where I_d — accurate value of the acoustical intensity [$\mathrm{Wm^{-2}}$], I_p — approximate value of the acoustical intensity [$\mathrm{Wm^{-2}}$], x_1, x_2 — distance of the microphones, 1 and 2, from the centre of the source [m], x — distance of the midpoint between the microphones from the centre of the source [m], k — wave number [rd/m], φ — phase shift angle between the microphone signals [rd].

In the case of the numerical method the influence of the phase shifts on the measuring accuracy is limited by a very careful selection of the pair of microphones.

4. Testing microphone probes

In order to avoid or eliminate the effects of errors due to phase shifts research has been undertaken on the phase characteristics of a two microphone probe. The following methods were applied:

measuring method in the far field of the loudspeaker,
measuring method in the near field of a vibrating piston,
electrostatic actuator method,
measurement of the phase shift of the microphone preamplifiers.

The measuring method in the far field of the loudspeaker was based on the measurement of the phase shift between the acoustical pressures measured by two microphones placed beside each other in an anechoic chamber at a considera-

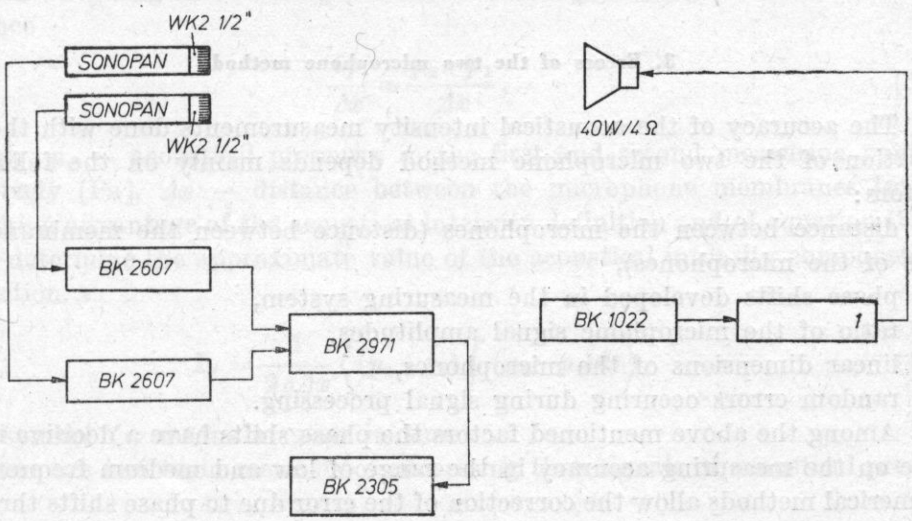

Fig. 1. Diagram of measurements of phase shifts between microphone signals. Measurements in the far field of the loudspeaker; *1* — power amplifier

ble distance from the loudspeaker (about 3 m). The microphones were located exactly in a plane perpendicular to a line connecting the microphones and the loudspeaker being turned toward the loudspeaker.

Investigations were conducted for harmonic signals of the frequency of the middle octave bands, ranging from 125 to 2000 Hz. Fig. 1 presents the measuring system diagram. The obtained phase measurement results were not stationary. The method proved itself useless for this kind of tests.

The measuring method in the near field of a vibrating piston applied a rigid piston as a plane wave source. Microphones were placed on a rigid support in such a mannerr so their membranes were in a plane parallel to the piston surface. A constant value of the acoustical pressure in the surroundings of the microphones was maintained, owing to the application of a feedback with a vibration exciter. The measuring diagram is shown in Fig. 2. Measurements

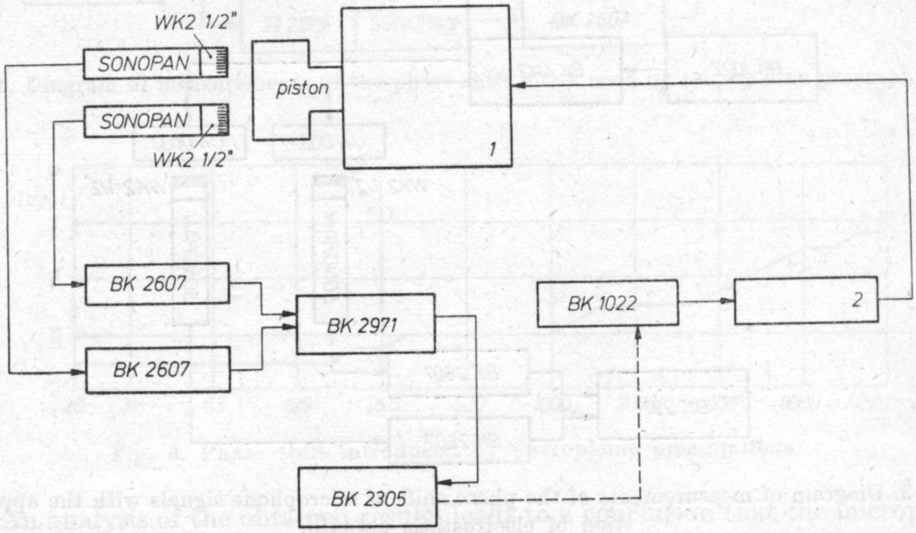

Fig. 2. Diagram of measurements of the phase shift between microphone signals. Measurement in the near field of a vibrating piston; *1* — exciter of vibrations RFT; *2* — power amplifier

were conducted in a frequency range from 20 to 5000 Hz. Obtained frequency characteristics can be recognized as consistent with reality only in the range from 20 Hz to about 2000 Hz. In higher frequency ranges significant phase changes, amounting to 360°, were observed. They were due to the non-homogenity of the acoustical field of the piston and the vibrations of the console supporting the microphones. The vibrations of the console were registered by a BK 4332 type accelerometer placed between the microphones. Significant phase changes occurred a ta frequency of about 2000 Hz. This corresponds to a frequency at which resonance vibrations of the support have been observed.

The electrostatic actuator method made use of a BK 4142 standard reciprocity calibration apparatus. The electrostatic actuator was used in the measurement of the amplitude characteristic of the microphone. After putting it on a microphone a condenser is formed, where the microphone membrane and the electrostatic actuator are the plates. The actuator is supplied with a current with an altering voltage, which is a sum of a constant voltage (800 V) and a variable voltage of an amplitude of $30 \div 40$ V. This causes the formation of an electrostatic force acting on the microphone membrane in a similar manner as the acoustical pressure which achieves accordingly a level of 88–100 dB. Thanks to supply voltage retuning and microphone signal registration an amplitude characteristic can be obtained. After placing two actuators supplied from one power source on two microphones, the measurement of a phase shift between the microphone signals is possible. Because the acoustical field surrounding the microphone may have an influence on the accuracy of the measurement, it is advised to make sure that the background pressure level does not exceed 30 dB.

The measuring diagram is shown in Fig. 3. The obtained phase characte-

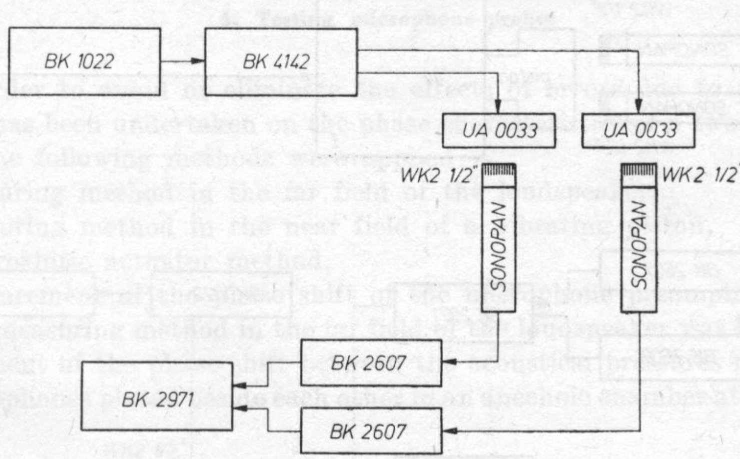

Fig. 3. Diagram of measurements of the phase shift of microphone signals with the application of electrostatic actuator

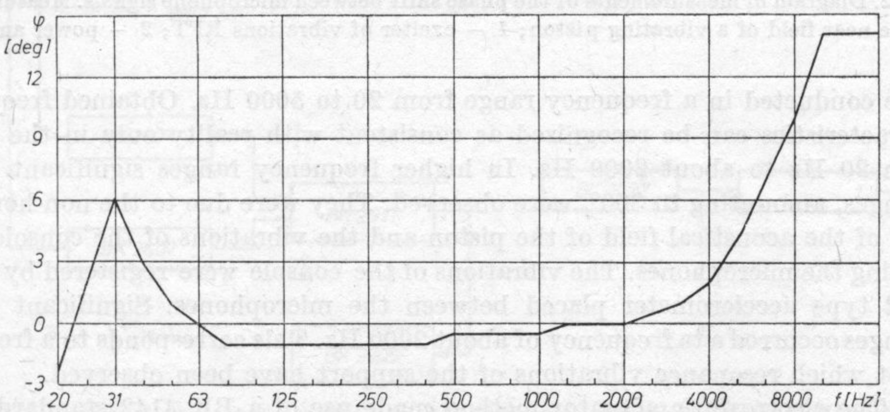

Fig. 4. Phase shift between microphone signals

ristic for a chosen pair of microphones, presented in Fig. 4, is a total phase characteristic of microphones with microphone preamplifiers.

Investigations of the phase shift introduced by the microphone preamplifiers were conducted for a pair of preamplifiers applied in the two microphone probe. In order to supply identical signals to the preamplifiers, JJ 2615 adaptors were applied in place of microphone cartridges. Frequencies of signals supplied to the adaptors were changed in a continuous way in a range from 20 to 20 000 Hz. Phase shifts were read for frequencies of the middle octave bands. The measurement diagram and the test results are shown in Fig. 5 and 6, respectively.

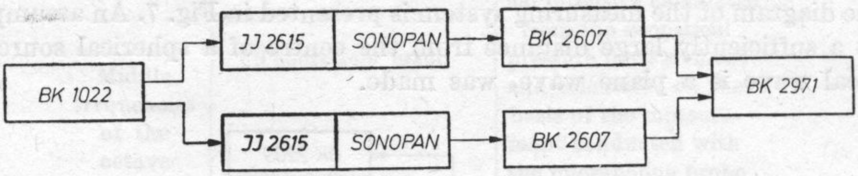

Fig. 5. Diagram of measurements of the phase shift introduced by microphone preamplifiers

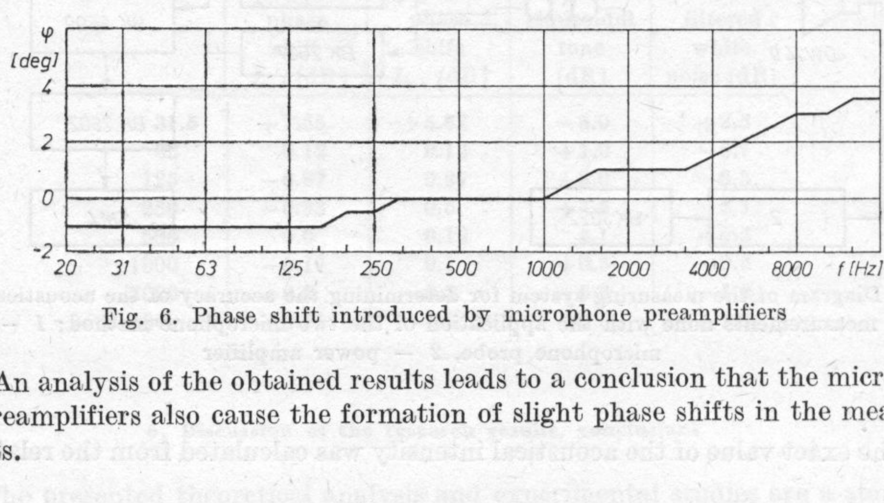

Fig. 6. Phase shift introduced by microphone preamplifiers

An analysis of the obtained results leads to a conclusion that the microphone preamplifiers also cause the formation of slight phase shifts in the measurements.

5. Verification tests of the two microphone method

A. Measuring system

A system consisting of the elements mentioned below was applied for the measurements done according to the two microphone method:

two microphone probe, made from microphones and preamplifiers produced by "SONOPAN" in Białystok,

microphone amplifiers, type BK 2607,

digital recorder BK 7502,

multiplexer BK 5699,
digital computer SM 4.

According to equation (5) a program was worked out in order to calculate numerically acoustical intensity.

B. Acoustical intensity measurements

The verification of the accuracy of the acoustical intensity measurements with the application of the developed measuring system, was done by measuring acoustical intensity in the far field of a loudspeaker placed in an anechoic chamber. The diagram of the measuring system is presented in Fig. 7. An assumption, that at a sufficiently large distance from the centre of a spherical source the acoustical wave is a plane wave, was made.

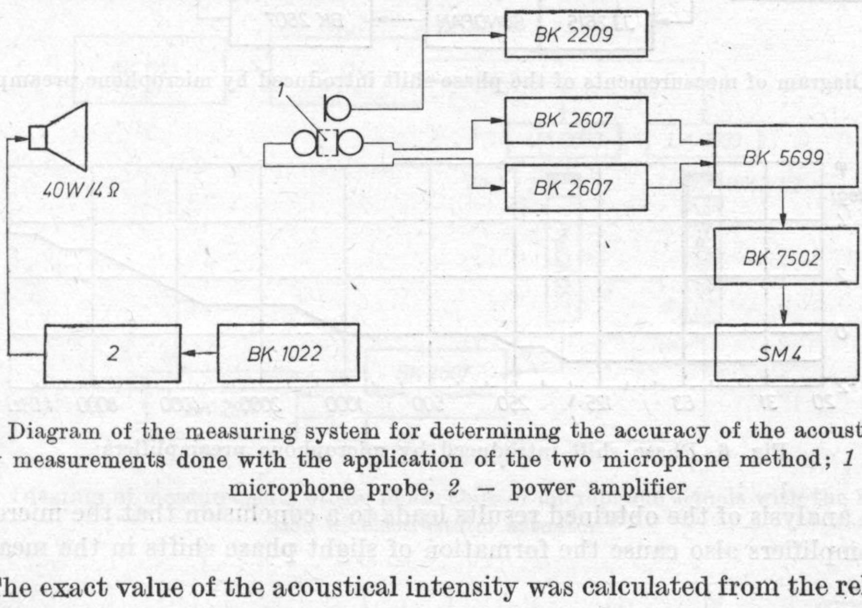

Fig. 7. Diagram of the measuring system for determining the accuracy of the acoustical intensity measurements done with the application of the two microphone method; *1* – two microphone probe, *2* – power amplifier

The exact value of the acoustical intensity was calculated from the relationship

$$I_x = \frac{p_{su}^2}{\varrho c} \quad [\mathrm{Wm}^{-2}], \tag{7}$$

where p_{su} – rms value of the acoustical pressure [Pa], ϱ – air density [kgm^{-3}], c – sound velocity in air [m/s].

The measurement was conducted through placing an additional microphone near the probe. A loudspeaker and a system of loudspeakers with a circular directional characteristic in the investigated frequency range were the sound source. Acoustical signals generated by the source had the character of white noise filtered in third-octave bands and sinusoidal tones with frequencies of middle octave bands.

Results achieved by the discussed methods are presented in Tab. 1.

Table 1 gives also the results of numerical calculations of the theoretical error of the method for a monopole source, taking into account real phase shifts occuring in the applied two microphone probe.

Table 1. Comparison of the results of the acoustical intensity measurements done with the application of different methods, and list of method errors determined theoretically

Middle frequencies of the octave bands	Theoretical error		Level difference between the acoustical intensity calculated from the acoustical pressure measurement and calculated on the basis of the measurement conducted with the microphone probe	
	for a positive phase shift $L_{\varepsilon+}$ [dB]	for a negative phase shift $L_{\varepsilon-}$ [dB]	forcing with sinusoidal tone [dB]	forcing with filtered white noise [dB]
31.5	+7.55	+5.62	−8.0	+3.5
63	0.12	0.12	+1.0	−6.7
125	−0.87	0.87	+3.0	−3.5
250	−0.33	0.5	+4.5	−2.7
500	0.0	0.12	4.1	−0.5
1000	−0.17	0.1	+0.3	−0.5
2000	−0.2	−0.2	+5.3	−1.0
4000	−0.81	−1.0	+2.0	−2.3

6. Discussion of the research results, conclusions

The presented theoretical analysis and experimental studies are a stage of a work of implementing the two microphone method of determining acoustical intensity.

The theoretical analysis of the method errors and measurements of the phase shifts between microphone signals enable the theoretical evaluation of the corrections introduced into the calculating program, in order to obtain the exact value of the measured acoustical intensity.

On the basis of literature studies, theoretical analysis and conducted tests the following conclusions can be reached:

a good conformity between the acoustical intensity measured with the two microphone probe and the results obtained from the measurements of the acoustical pressure in the far field, has been reached in the frequency range from

125 Hz to 2000 Hz. This concerns noise signals;

investigations on phase shifts between microphone signals for a chosen pair of microphones have proved that their values vary in a range of $\pm 1°$ in the frequency range from 63 to 2000 Hz;

there is a possibility of implementing the two microphone method of determining acoustical intensity with the application of the imaginary part of the cross spectrum function, on the basis of possesed measuring and calculating instrumentation (system SM 4);

the accuracy of the acoustical intensity evaluation depends upon the source radiating character (e.g. order of source power) and the distance of the microphones;

further verification studies should include other types of signals and the cases of signals generated by industrial sources.

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REVERBERATION TIMES AND REVERBERATION LEVELS

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Reverberation times and reverberation levels are the statistical parameters of the stochastical sound field in a room. Their theoretically derived and practically measured properties are discussed. Reverberation times and reverberation levels are place dependent, they are also to a certain extent influenced by the shape of the hall and the distribution of the absorption in it. This influence is however limited as regards the reverberation times.

1. Introduction

On 29 October 1898 W. C. SABINE observed that the product of total absorption and the duration of residual sound was a constant. The well known SABINE reverberation formula had been discovered.

There have been times that the reverberation time (*RT*) was highly valued, even overvalued, as the determining factor in roomacoustics; there have also been times that it was undervalued as not being of real importance for the subjective quality of the acoustics of a room.

We now know that the value of the *RT* is important, even very important, but certainly not the only important factor.

The sound energy density at different places in a room — mostly refered to in the form of the sound level — in absolute values, as well as relative to the sound power of the source, can be considered even more determinant for the acoustics of a room.

Reverberation times and the sound energy densities are related to each other, be it in a complicated way. We must always study them in their mutual relationship.

2. The sound field in a room

All known deductions of reverberation time formulae are based on models, on certain conceptions of what happens in a room. Most models are raymodels, mostly ending up in EYRING type of formulae; also well known is the "eigen" modes model etc.

All these models try to approximate reality, none of them is convincing and all are based on assumptions that can not be completely verified.

A rigorously valid reverberation formula can be found in differential form, but does not give an as simple and straightforward solution as the SABINE'S formula. The SABINE formula as well as the EYRING formula prove to be special cases or solutions of this more general formula. One may however, using this differential form deduce under what conditions the SABINE formula may be valid, and what may be expected, in general, if other conditions prevail.

The sound energy at a certain point in a room consists of kinetic energy and potential energy

$$E = \frac{1}{2}\frac{P^2}{K} + \frac{1}{2}\varrho v^{-2} \tag{1}$$

after differentiation

$$+\frac{dE}{dt} = \varrho \vec{v}\frac{\partial \vec{v}}{\partial t} + \frac{1}{K}\,p\,\frac{\partial p}{\partial t}. \tag{2}$$

With the use of the equations of continuity and inertia one finds

$$-\frac{dE}{dt} = \vec{v}\operatorname{grad} p + p\operatorname{div}\vec{v} = \operatorname{div} p\vec{v}. \tag{3}$$

Using GAUSS's theorem one can now write

$$-\frac{dE}{dt} = \int\!\!\int_S p\cdot\vec{v}\vec{n}ds. \tag{4}$$

This is, the total sound energy E in a certain volume will decay as the scalar product of the normal component of the particle velocity on the surface times the pressure integrated over the boundary of the volume.

One may readily assume that the on the sufraces mean incident intensities (over time and total boundaries taken) will be proportional to the sound energy in the volume. Thus, in the mean again, an exponential decay may be expected.

From the formula (4) one learns, that if there is an ideal diffuse, that is a completely homogeneous and isotropic sound field, that will stay during the whole decay process ideally diffuse, the SABINE formula can be deduced. This can however only be the case if everywhere and in every direction in a room, sound will be equally absorbed. This may be found if air absorption is the only sound absorbing machanism present in the room — not a very interesting case from the practical point of view.

Normally the sound absorption will take place at the room boundaries, and therefore an ideal diffuse sound field will no longer be possible, because of the finite sound velocity. It will take some time before the depletion of the sound field, taking place at the boundaries will have been equally divided over the whole room and will be felt in the incoming sound at the absorbing boundary. The sound energy falling on the boundary will, in the mean being older and thus stronger than the mean, take over the whole hall of the in the hall present sound energy. Taking this effect into account results in the EYRING formula if one assumes, according to EYRING that one may substitute the "mean free path" for the distance sound will transverse between two reflections and that all boundaries have the same absorption coefficients under all conditions. If one calculates the influence of what we will call the EYRING effect one finds, as long as the mean absorption coefficient is smaller then ca. 0.7, that in a very good approximation we can write

$$T_r = T_S - 0.1\,c\,\frac{V}{S_{\text{tot}}}, \tag{5}$$

wherein T_r is the expected reverberation time and T_S the SABINE reverberation time; V the volume; S_{tot} the total surface of the boundaries and c a constant that in the derivation of EYRING becomes (1). This constant proves to depend also on the shape of the room and the absorption distribution over the boundaries. If one takes also into account that the real distances between two reflections in a room vary from zero to much larger then the mean free path, then c proves to be in most cases around 0.5 to 0.6.

If the value for c is found to be smaller than ca 0.5, then we must question the validity of the geometrical model on which the EYRING reverberation time formula is based. The EYRING effect is certainly not the only example of an uneven sound energy density distribution in a hall.

The uneven distribution throughout the hall of the sound sources, the uneven distribution of the absorption etc. etc., all this results in an uneven sound energy distribution that in general will result in an reverberation time formula in the form

$$T_r = T_S\left(1 + \frac{a_1}{T_S} + \frac{a_2}{T_S^2} \ldots\right). \tag{6}$$

Wherein $a_1, a_2, \ldots$ are coefficients (positive or negative) that depend on the unevenes of the sound distribution.

Until now we have assumed that the sound energy density distribution stays during whole the decay process the same. Only in this case one may expect a logarithmic decay.

As regards the instantaneous sound pressure distribution, large variations in time and in space are seen. This alone does however not mean that the sound energy density varies in the same way, for the sound pressure represents in general only the potential energy, the kinetic part of the total energy may and will in most cases vary as a complement to the potential energy. Nevertheless the sound energy density may also vary in the short term, as a consequence of the stochastic character of the sound field, as it will be discussed in the following section. If however, in the long run the sound energy distribution stays still the same, then the short time variations have for the reverberation time no other meaning then that they influence the accuracy with which the reverberation time can be measured. If on the other hand the sound energy density changes with time, then the decay will no longer be exponential although in most practical cases it will still stay nearly exponential.

3. Accuracy of reverberation times and reverberated sound level measurements

The sound field in a room may almost always — that is if the dimensions of the room are not small with regard to the wavelength and/or the walls etc. are not nearly 100 % absorbing — be considered to have a stochastic nature [1]. The sound pressure at any point in such a room will be a variable that is specified by the so called chi-square probability distribution function. Using this probability distribution it is possible to calculate the accuracy with which sound levels and reverberation times in a room can be determined [2].

The sound level may be determined with a standard deviation

$$\sigma_L \approx \frac{12.5}{\sqrt{2ft}} \tag{7}$$

and the standard deviation with which the reverberation time can be determined is

$$\frac{\sigma_{T_{60}}}{T_{60}} \approx \frac{12.5}{S} \sqrt{\frac{3}{2ft_0}}, \tag{8}$$

wherein σ_L — the standard deviation in the sound level, $\sigma_{T_{60}}$ — the standard deviation in the reverberation time, f — the frequency bandwidth, t — the measuring time (in case of sound level), t_0 — the measuring time (in case of reverberation time), s — the so called dynamic span, T_{60} — the reverberation time.

Between t_0, S and T_{60} exists the relation $t_0/T_{60} = S/60$. In Fig. 1 the meaning of these quantities is illustrated.

Using regression techniques, we can now measure reverberation times much more precisely than hitherto was possible.

The standard deviation that can then be determined will in every case be different but must in the mean agree with the above given values. This has been confirmed in practice.

It is thus possible to determine the reverberation times at a certain place in a room with standard deviations of ca. 1 to 7 %, depending on frequency etc. In the mid frequencies in general an accuracy better than 2 % can easily be reached.

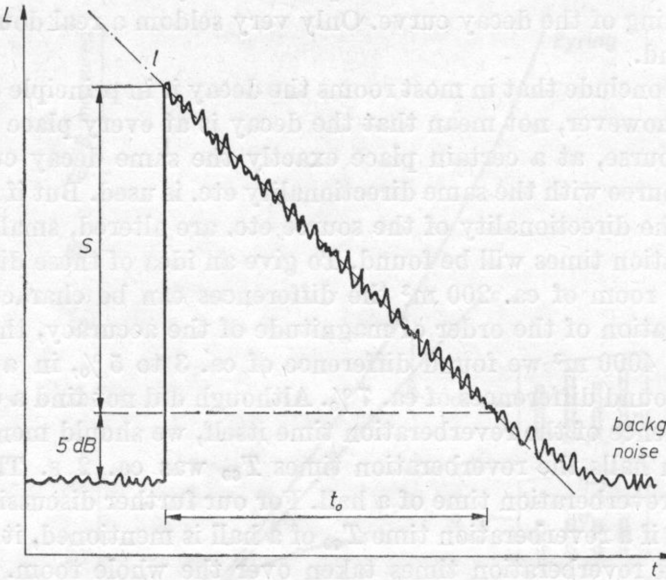

Fig. 1. Determination of the time t dependence of the decay of level L, by the regression technique. The figure illustrates the meaning of the dynamic span S and the measuring time t_0

4. Measured reverberation times

A precondition for the validity of the before given formulae is the assumption of an exponential decay.

Of course one may define the reverberation decay as an essentially exponential decay and consider every departure from it as an inaccuracy, but it is difficult to agree with such a point of view.

In general one considers a non-exponential decay as a consequence of many coupled system giving rise to a combination of exponential decays.

We have developed a procedure based on repeated integration to determine the different exponential components that may be found in a reverberation decay.

Beside real components (real reverberation times) we also had to take into account complex components (complex reverberation times). Moreover, as a set of exponential decays does not form a orthogonal system one should take a certain arbitrariness in the determination of the different decay times as unavoidable.

The results of the application of these procedures were nevertheless very interesting. In most of a very large number of cases that the reverberation times was determined this way, one important component was found that proved to agree reasonably well with the reverberation time as determined by regression techniques. And if there were more components they related in most cases to the very beginning of the decay curve. Only very seldom a real double or triple decay was found.

One could conclude that in most rooms the decay is in principle exponential.

That does however, not mean that the decay is at every place in the room the same. Of course, at a certain place exactly the same decay curve will be found when a source with the same directionality etc. is used. But if the distance to the source, the directionality of the source etc. are altered, small differences in the reverberation times will be found. To give an idea of these differences, in a reverberation room of ca. 200 m^3 the differences can be characterised with a standard deviation of the order of magnitude of the accuracy, thus 1 to 2 %. In a hall of ca. 4000 m^3 we found difference of ca. 3 to 5 %, in a hall of ca. 100.000 m^3 we found differences of ca. 7 %. Although did not find a clear indication of the influence of the reverberation time itself, we should mention that in the above given halls the reverberation times T_{60} was ca. 2 s. There is thus clearly not one reverberation time of a hall. For our further discussions, it must be stressed that if a reverberation time T_{60} of a hall is mentioned, it is the mean of the measured reverberation times taken over the whole room.

As we will discuss in the next section, one finds never or almost never an even sound distribution in a room. We may thus expect to find the SABINE formula not completely validated by experiments. The absorption coefficients of materials, measured using the SABINE formula seldomly agree with those measured using other techniques.

But on the other hand, no indication could be found of the preferation of the EYRING formula.

We tried to verify the by EYRING predicted more than proportional shortening of the reverberation time if the amount of absorbing material in the room is increased.

At first we did our experiments in two reverberation rooms of different size. Although the accuracy was high enough to detect the EYRING effect no such effect could be found.

The results of the measurements (at 1000 Hz) in one of the reverberation rooms are given in Fig. 2. Vertically the apparent (calculated using the SABINE formula) mean absorption coefficient is given, if a part of the total material as indicated horizontally was placed in the reverberation room. The absorbing material was equally divided on walls and floor. The different patches of material were placed at a distance from each other in such a way that no influence of the edge effect was to be expected. To be sure that the absorption was halved etc. a rotation system was used.

We did a large number of reverberation measurements under all kinds of circumstances in the well known *Espace de Projection* in the Centre POMPIDOU in Paris. This hall has walls and a ceiling that is made of rotatable panels that have an absorbing, reflecting and diffuse reflecting side. Moreover the ceiling

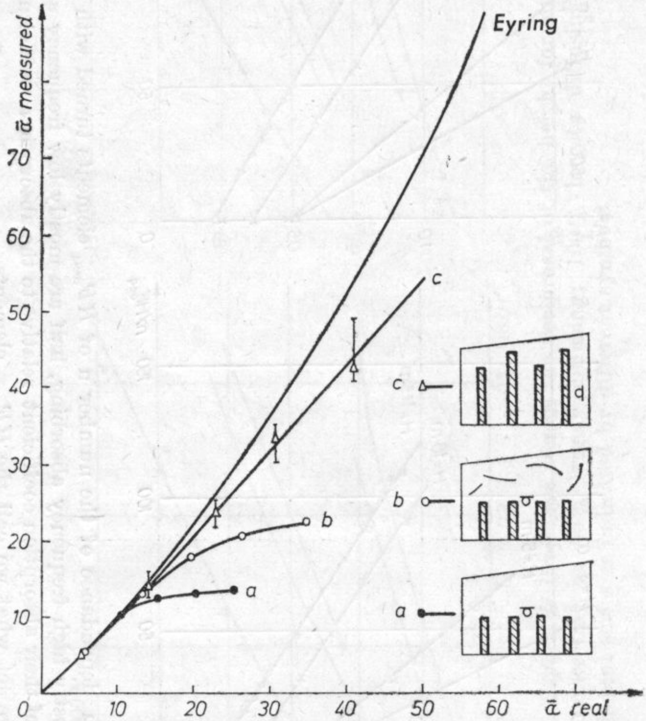

Fig. 2. The apparent (measured) mean absorption coefficient in dependence of the "real" absorption coefficient, as determinded according to the standardized reverberation method. To be sure that the relative points on the horizontal scale are correct, a method of halving and rotating of the material was used. In the first experiment (a) the walls were only partially covered. Bringing in some more material and diffusing elements (b) resulted in a better agreement with what was expected on basis of the SABINE's reverberation law. Covering all walls and the floor with material gave curve (c). Volume of the room 200 m^3. Impulse type source, filtered octave of 1000 Hz

consists of three parts that all three can independently be lowered from maximum height (11.5 m) to ca. 2 m (lowest height permitted for safety reasons). The floor area is 24×15.5 m^2. Between the three ceiling parts heavy iron curtains can be lowered. Thus absorption and shape of the room can be varied. It is also possible to divide the hall in two or three coupled rooms.

Earlier measurements in this room, did not show an EYRING effect. We tried again, measuring at different heights and for a large number of situations. The results are given in Fig. 3 to 5 for the frequencies 500, 1000 and 2000 Hz.

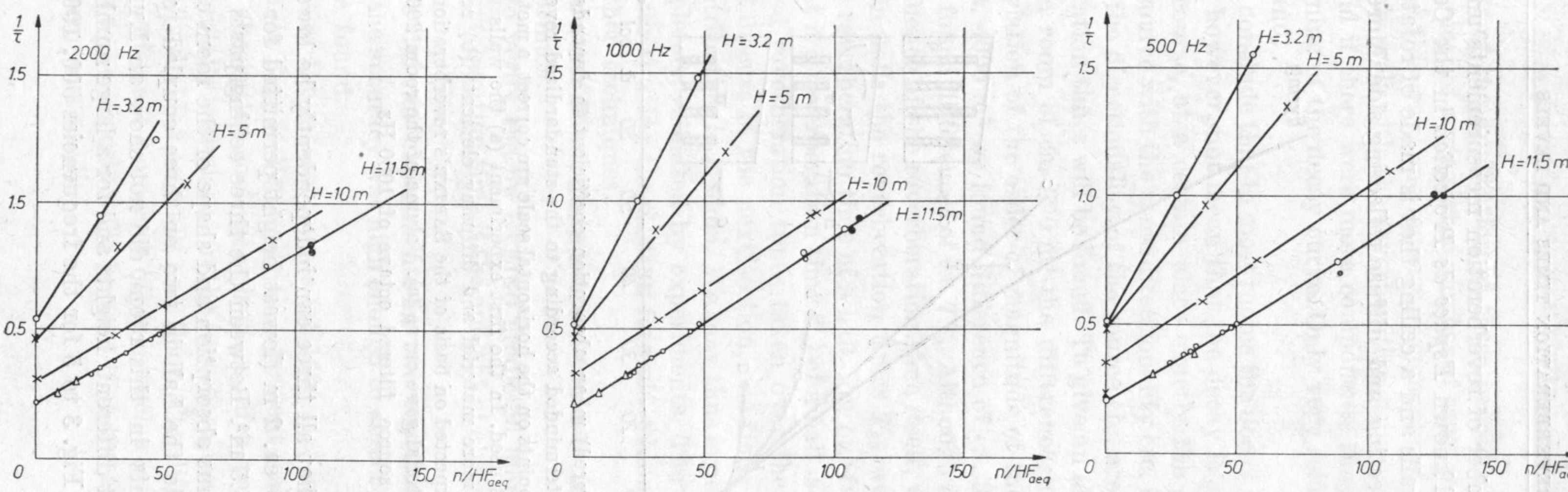

Fig. 3–5. The reciprocal $1/\tau$ of the reverberation time in dependance of the number n of HF_{aeq} elements turned with their absorbing sides to the hall. Half of the turnable elements (periactes) are mostly high frequency absorbing, half are mostly low frequency absorbing. If one counts the low frequency absorbing periactes as in proportion of their absorption coefficient relative to the absorption coefficient of the high frequency elements we get what we call the HF_{aeq} elements

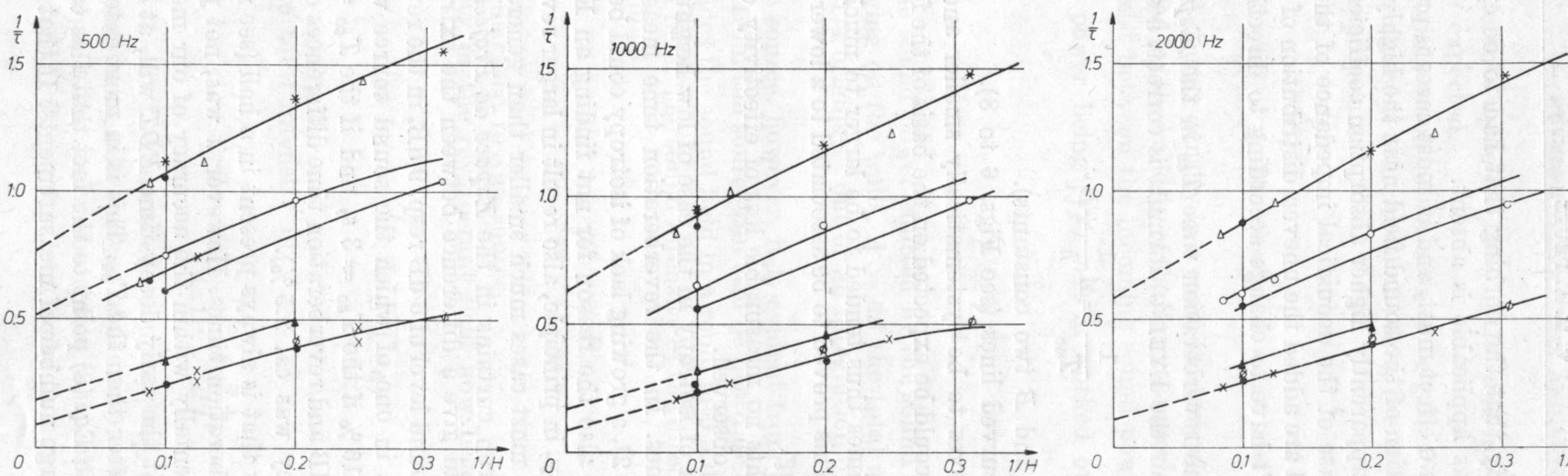

Fig. 6–8. The reciprocal of the reverberation time of the *Espace de Projection* versus the reciprocal of the height for different wall and ceiling configuration (100% absorbing; 50% absorbing and 50% reflecting or 50% largely diffusing; 100% largely diffusing c.q. 100% reflecting). Measured in different series as indicated by different symbols

One can easily verify that the curbing that had to be expected if indeed the EYRING formula was applicable is absent.

Still a certain EYRING effect must, and in most cases is present. In a reverberation room the depletion of the sound field near the highly absorbant 10 m^2 surface can result in an apparently higher absorption coefficient of the material then expected on the base of the acoustical impedance of the material, but if more patches of material are added the uneven distribution of the sound energy density evidently doesn't have to change according to direction expected from EYRING.

The reciprocal of the reverberation time T_{60} in the *Espace de Projection* must be, if the SABINE (or the EYRING) formula is correct, a straight line

$$\frac{1}{T_{60}} = A\frac{1}{h} + B$$

(h being the height, A and B two constants).

Instead we found curved lines (see Figs. 6 to 8).

The value of A proves to be systematically smaller and the value of B proves to be larger then could be expected on the basis of the formula and other measured data.

The reverberation times thus tended to be larger to much larger for lower heights, than expected. This proved to be associated to a lower effectivity of the absorption on the ceiling.

For the higher heights no measurable less of effectivity of the absorption on the ceiling could be observed.

This indicates a lack of isotropy in the case of low heights as the origin of longer reverberation times. In the reverberation time measurements in the reverberation room (Fig. 2), a growing lack of isotropy could be considered to be at least for a large part also the reason for not finding an EYRING effect.

Non-homogenity may, in principle, also result in larger reverberation times, however this effect is in most cases much smaller than generally expected.

Lowering the two iron curtains in the *Espace de Projection* to 5 m resp. 2.80 m from the floor did give a difference between the extreme parts of the hall in the steady state sound level of 6 dB resp. 9 dB, in the reverberation times for these extreme parts, in one of which the sound source was located, only a difference of 7 % resp. 10 % if the $T_{60} = 3$ s, and if the $T_{60} = 1$ s level differences of 10 dB resp. 17 dB and reverberation time differences of 10 % resp. 12 % were found (the accuracy was ca. 2.5 %).

The non-homogenity that is always present in a hall (see next section) will certainly influence reverberation times. However it was, not possible to determine this effect experimentally within the accuracy of our measurements.

One may expect that the early decay time *EDT* will, at least near to the source be somewhat shorter then the T_{60}. This is in most cases also found, but we must with reference to Eq. (8) point to the fact that this can only be found as the mean of a very large number of measurements. If that is taken into account then a difference up to ca. 10 % in the area near the source (see also next section) may be expected.

There is certainly much information to be gained out of the first part of the decay curve; to extract it other methods then the statistical reverberation analysis are then appropriate.

5. The reverberant sound level

If the sound energy density in a room is everywhere homogeneous and isotropic, as is needed for the validity of the SABINE reverberation formula, then equilibrium must exist between the incoming sound power P ($= -dE/dt$) and the outgoing sound power being $\iint p\vec{v}\cdot\vec{n}\,ds$ or worked out in case of homogenity and isotropy

$$P = I\frac{A}{4}, \tag{9}$$

wherein P — the sound power emitted by the sound source (s), I — the sound intensity impinging on the walls, $A = \bar{a}S$ the total absorption in the hall.

The factor 1/4 originates out of an integration over all angles of incidence on the walls.

Near to the source, however, it is evident from experience that the sound coming directly from the source can still be heard distinctly. As a logical consequence, one divides the sound field in a room into a direct sound field and a reverberant sound field. The sound intensity of the direct field can be given by the well known formula

$$I_{dir} = \frac{P}{4\pi D^2}, \tag{10}$$

wherein D the distance to the source.

The intensity in the reverberant field now becomes

$$I_R = \frac{P}{\dfrac{A(1-\bar{a})}{4}} \tag{11}$$

resulting for the total intensity in

$$I = P\left(\frac{1}{4\pi D^2} + \frac{4}{R}\right), \tag{12}$$

wherein $R = A(1-\bar{a})$ the so called roomconstant, $\bar{a}$ is the mean absorption coefficient. It is better to take instead of the mean absorption coefficient $\bar{a}$, the so called a_{proj}, that is the mean absorption when all absorbing surfaces in the room are projected on the surface of a body that results if one projects out of the sound source in all directions rays that are proportional to the square of the directivity factor Q in that direction.

In case of a real point source, the body is a sphere. Now one sees directly that absorption found near to the source has more influence on the level of the reverberant sound, then the same amount of absorption further away.

The second term in the expression at the right side in Eq. (12) proves in practice to be dependent on the distance to the source also, that is it decreases with distance. In most cases a power function is found experimentally, see Fig. 9 and 10.

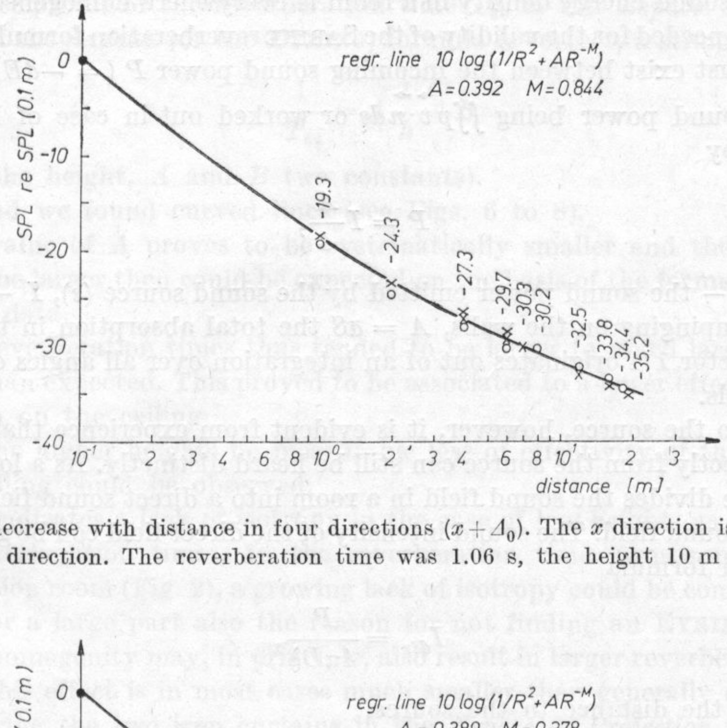

Fig. 9. The decrease with distance in four directions ($x + \Delta_0$). The x direction is the vertica direction. The reverberation time was 1.06 s, the height 10 m

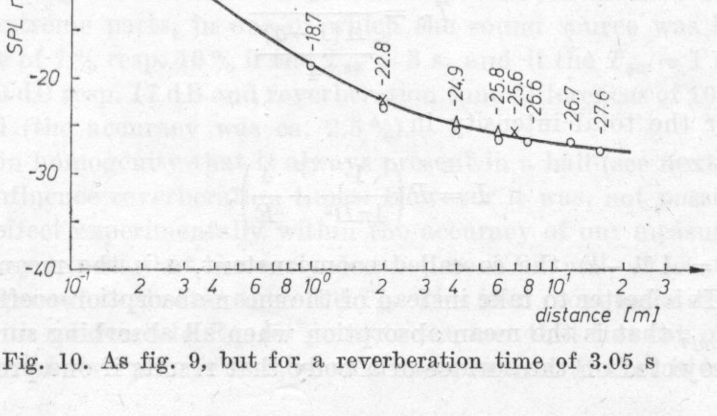

Fig. 10. As fig. 9, but for a reverberation time of 3.05 s

More precise observation shows that the basic mechanism is an exponentially decreasing propagation superimposed on a power function originating out of sound expansion. At the wall boundaries a reflection takes place, as can be seen in Fig. 12. The results in the overall decrease what gives the impression of a power function.

In a very large low room the exponential decrease is still very clear in the so called "plateau" (see Fig. 11). As can be seen from Fig. 9, the decrease with dis-

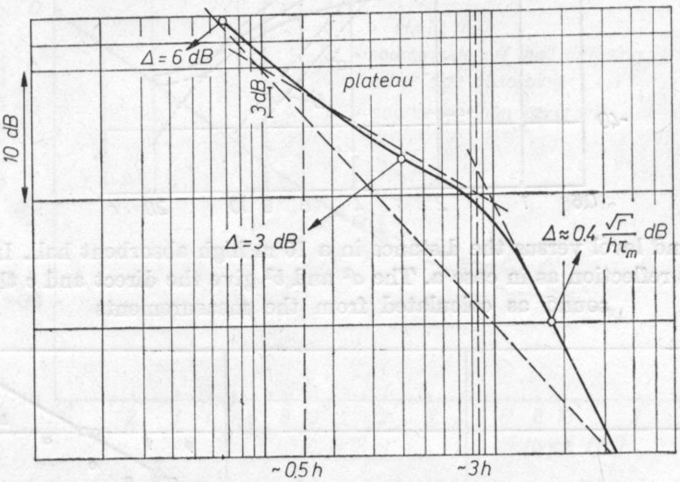

Fig. 11. Schematic presentation of the decrease with the distance to the source of the sound level in a large low space. Δ is the decrease in dB for each doubling of distance. V the volume of the room in m³, h the height in m and τ_m the measured reverberation time ($r: = V$)

ance is essentially the same in all directions. This has also been found in all kinds of rooms. An experimental relation between the decrease when doubling the distance ΔL in rectangular rooms and different roomconstants has been found (see Fig. 12)

$$\Delta L \cong 0.4 \frac{\sqrt{V}}{HT_{60}}. \tag{13}$$

From Fig. 12 and Fig. 13 we learn that this Eq. (13) is the best fit to experimental data only; it is however, an useful equation for practical calculations.

In any case, we learn that the reverberant sound field in a room in a steady state situation is far from homogeneity and isotropy.

The distribution of the absorption in a room with normal dimensions has almost no measurable influence on the reverberation time, absorption in corners is only a very little bit more efficient than in the middle of a wall. We also studied the case when one half of the *Espace de Projection* was absorbant, the other half diffusely reflecting. There was no significant difference in reverberation times observed. The reverberant sound levels in the room, especially the

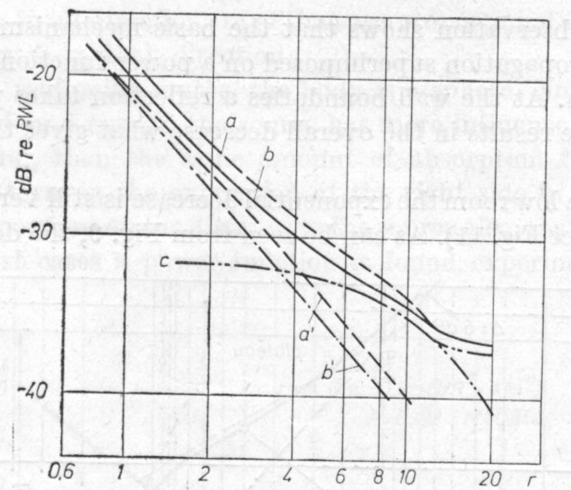

Fig. 12. The sound level versus the distance in a 10 m high absorbent hall. In case b there is no strong first reflection as in case a. The a^1 and b^1 give the direct and c the reverberant sound as calculated from the measurements

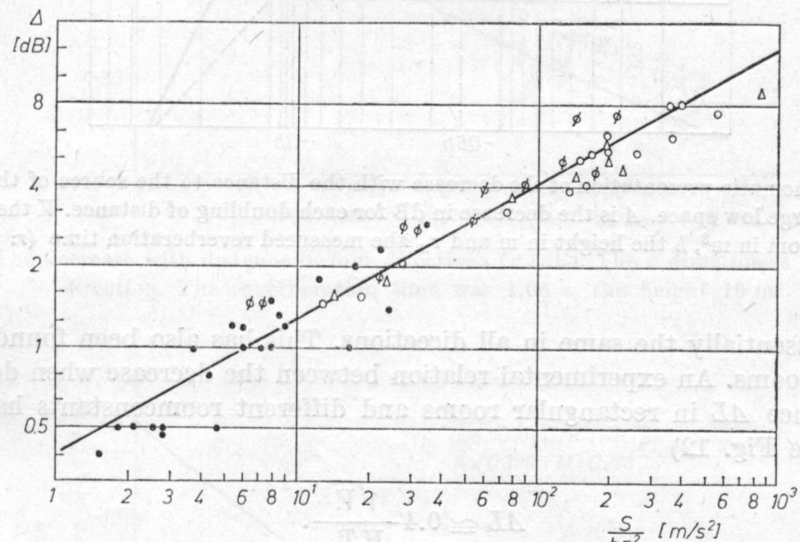

Fig. 13. The decrease per doubling of distance Δ vertically versus S/hT_m^2 the floor area S over the height h, times the square of the measured reverberation time T_m, as measured in a large number of rectangular rooms, halls etc.

decrease with distance showed a typical, but not very large difference (see Fig. 14).

The decaying sound field has in general the tendency to become more and more homogeneous, if not also isotropic. This can be deduced from i.a. Fig. 15. This figure shows the decay of a sound field at different places in the same room. The different decay curves at different distances of the sound sources (a pistol shot) have, as regards the starting points of the different curves, time lags as can be seen in the figure. The curves fall after a short time almost perfectly on each other, indicating that the decaying sound field becomes spatially equal in intensity. The same has been experienced in a large quantity of rooms of simple shape. In rooms having, a more complicated shape larger, to much larger, level differences have been found. It would, however, be outside the scope of this paper to discuss this.

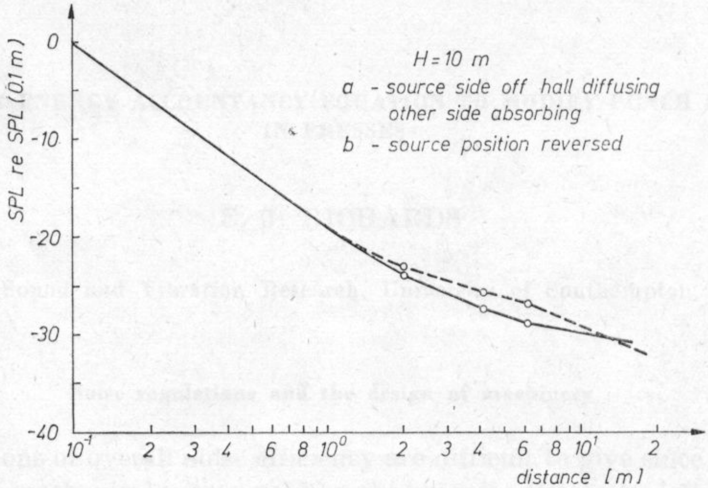

Fig. 14. The sound level versus distance for two loudspeaker position in the *Espace de Projection* of IRCAM Paris. One side of the hall is absorbent the other side diffusely reflecting ($T_{60} = 1.6$ s)

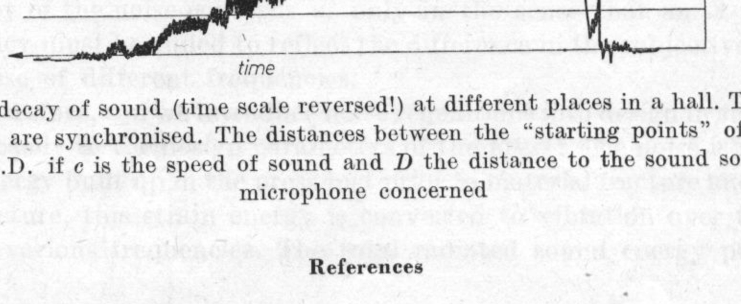

Fig. 15. The decay of sound (time scale reversed!) at different places in a hall. The different decay curves are synchronised. The distances between the "starting points" of the curves agree with c.D. if c is the speed of sound and D the distance to the sound source of the microphone concerned

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USING THE ENERGY ACCOUNTANCY EQUATION TO MODIFY PUNCH DESIGN IN PRESSES

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Noise regulations and the design of machinery

Estimations of overall noise efficiency are difficult to give since the measurement of the mechanical energy used by the press is not easy to define, let alone to measure. Flywheel machines take the power from the flywheel, which slows down, but much of this is absorbed in accelerating the mechanism and in other ways. However, we can estimate that, of the flywheel energy which is available, some 0.0001 goes into noise. This is a small fraction and far too low to consider using for any other purpose. Thus, once again, the argument that noise is wasted energy cannot be a valid reason for noise reduction. The reason for reducing noise must hinge on the need to meet deafness limits, and must be related to the 85 or 90 dB(A) limit on $L_{Aeq,8L}$. This is an average radiated acoustical energy reaching the ear of an operator and must be related to the average energy radiated from the whole machine during the same period of time. Knowledge of the frequency of the noise concerns us only in the sense that an A-weighting with frequency must be added to reflect the difference in the subjective acceptability of noise of different frequencies.

How, therefore, can we introduce noise regualtions into design in such a form as to relate easily to the design parameters of the press? The noise is related to the strain energy built up in the press just prior to material fracture and the way that, on fracture, this strain energy is converted to vibration over the whole machine at various frequencies. The total radiated sound energy per impact

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will be given by

$$\omega_{\mathrm{rad}} = \varrho_0 c \sum S \tau_{\mathrm{rad}} \int_0^T < v^2 > dt \tag{1}$$

over the surface of each member.

As seen in Fig. 1, the measurement or calculation of vibration levels in parts of the machine well away from the workplace lead to a poor diagnostic method

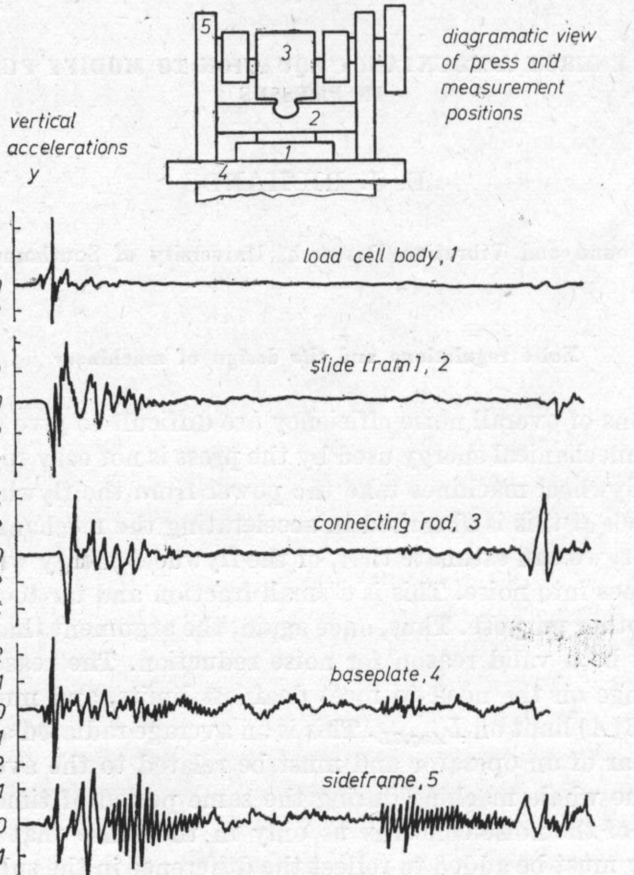

Fig. 1. Progress of vibration through a press

and the use of the energy accountancy equation in its most convenient form carries more promise [1], the impulsive force concerned being the one most easily measured, i.e. the one exerted by the punch itself.

Alternative forms of the energy accountancy equation

The Energy Accountancy Equation [1] is usually formulated in terms of the L_{Aeq} created in a frequency band Δf centred around a frequency f_0 but this is largely because of the need to make allowance for A-weighting of the sound with frequency. There are several other alternative forms which are more convenient to use in practical applications. This is particularly so if the structural modifications being made to the whole of a machine are of a minor character and the noise control measure being taken can be considered as simply an alteration to the force pulse shape $f(t)$ with no alteration to the structural response term nor the acoustic terms.

When this force pulse $f(t)$ is transferred to the frequency (f_0) domain by the Fourier transformation

$$F(f_0) = \int_{-\infty}^{\infty} f(t)\, e^{-j2\pi f_0 t}\, dt \qquad (2)$$

the force pulse $F(f_0)$ becomes a frequency weighted time average of the time average of the time pulse force centred around what we call a frequency (f_0). The term in the Energy Accountancy Equation, $10 \log |F(f_0)|^2$ is not as easy to visualise physically in terms of the operation of the mechine as would a pulse term indicating the variation of the force with time. It is therefore good sense to rewrite the Energy Accountancy Equation in the time domain if this can be achieved.

The noise regulations are drawn up in terms of the total radiated noise energy emitted per day and we can relate this to the noise energy per machine operation by dividing by N, the number of operations per day.

In its original form prior to taking logarithms and expressing it in decibel terms, the Energy Accountancy Equation is written

$$\omega_{\text{rad}}(A, f_0, \Delta f) = \sum_{f_0}^{f_0+\Delta f} \frac{\varrho_0 c \Delta f}{2\pi^2 \varrho_m f_0} N \frac{A\, \tau_{\text{rad}}}{f_0} \frac{1}{\eta_s d} |\dot{F}(f_0)|^2 \operatorname{Im} H(f_0) \qquad (3)$$

and the term $10 \log |F(f_0)|^2$ can be taken out of the equation and returned to its time integrating form only if it is constant with frequency, at least in the range of frequencies in which the noise energy is greatest.

This may not be possible, but there will always be some derivative of $|\dot{F}(f_0)|^2$ which will be effectively flat with frequency, and the secret is to examine the impulsivity of the motion and to find a force derivative which is flat.

Looking at the expression for the Fourier transform of $f(t)$ above, we see that we have the freedom of choosing whichever derivative we use by recognizing that

$$\dot{F}(f_0) = 2\pi j f_0 F(f_0) \quad \text{and} \quad \ddot{F}(f_0) = -4\pi^2 f_0^2 F(f_0), \quad \dddot{F}(f_0) = 2\pi j f_0 \ddot{F}(f_0) \text{ etc.}$$

It follows that we can replace $10 \log |\dot{F}(f_0)|^2$ by $10 \log |\ddot{F}(f_0)|^2 - 10 \log (4\pi^2 f_0^2)$ and we can write the Energy Accountancy Equation in the alternative noise energy form (not logarithmic)

$$\omega_{\mathrm{rad}} = \frac{\varrho_0 c N}{8\pi^4 \varrho_m d} |\ddot{F}(f_0)|^2 \sum_{f_0=0}^{\infty} \mathrm{Im} \frac{H(f_0)}{f_0} \frac{A\,\tau_{\mathrm{rad}}}{f_0} \frac{\Delta f}{f_0} \frac{1}{\eta_s} \tag{4}$$

and if $|\ddot{F}(f_0)|^2$ is constant with frequency, we can write $10 \log \omega_{\mathrm{rad}}(A) = 10 \log |\ddot{F}(f_0)|^2 +$ other terms which do not depend upon pulse shape but are dependent upon the structural response, radiation efficiency and A-weighting.

The same would apply if we were to use the third or any derivative of $F(f_0)$ except that the other non-pulse-dependent term would be modified to accentuate the lower frequency terms.

Which form of equation to use depends upon the degree of physical insight required. Transferring back to the time domain we have

$$|\ddot{F}(f_0)|^2 = \ddot{F}(f_0)\ddot{F}(f_0)^* = \int_0^{\infty} \ddot{f}(t)\, e^{-2\pi j f_0 t}\, dt \int_0^{\infty} \ddot{f}(t)\, e^{2\pi j f_0 t}\, dt$$

and if $f(t)$ is only significant for a series of short times of length Δt this equals

$$\sum [\ddot{f}(t)_{\max}]^2 (\Delta t)^2 = \sum [\dot{f}(t)_{\max}]^2$$

while

$$|\ddot{F}(f_0)|^2 = \sum [\dot{f}(t)_{\max}]^2 \quad \text{and} \quad |\dot{F}|^2 = \sum f^2(t)_{\max},$$

where Σ implies a summation of the peaks in the values of the relevant function with time.

Since noise is often greatest in the medium frequency range because the modified radiation efficiency is greatest there on machinery type structures, a good compromise is to take the form $|\ddot{F}(f_0)|^2$ or $\Sigma[\dot{f}(t)_{\max}]^2$.

Thus, a useful form of the Energy Accountancy Equation when applied to tool design is

$$L_{A\mathrm{eq}} = 10 \log \sum [\dot{f}(t)_{\max}]^2 + \mathrm{const} = L_{\dot{f}} + \mathrm{const}. \tag{5}$$

More important is the fact that rates of change of force are the highest derivative of the force which can be visualized diagnostically, and that errors arising from its non-constancy in the frequency domain are less important than the loss of conceptuality arising from using the wrong derivative.

Needless to say, the derivative to be used must also depend upon the variation of the structural response term. The higher the derivative used, the greater the fall-off with frequency in the relevant structural response factor, $\mathrm{Im}\,[H(f_0)/f_0^n]$ and the more the constant term in equation (5) will reflect the low frequency behaviour of the structure: each form must be taken on its merit in the particular case and the instrumentation which is available.

In all cases, the original form of the Energy Accountancy Equation in which everything is related to the frequency domain is the most reliable, though not necessarily the most useful. In practice, as frequency can be related to $(1/2)t_0$, we have found that good diagnostic skill can still be developed in the frequency form of the equation by recognizing that high frequency content can be equated to sharp temporal changes, except possibly in interpreting acceleration noise.

Tool design to reduce noise

While it may be possible to add internal damping (2) to a press at the design stage, the opportunities for doing so on existing presses are not great, especially as the damping has to be increased greatly in the vicinity of the punch as well as in the columns, a region whose design depends upon so many other parameters.

Increasing the stiffness by a useful amount is not very practical either, since the radiated noise level will vary as 10 log k_1/k_2, i.e., the stiffness will need to be doubled for even a 3 dB noise reduction.

Changes int ooling design to reduce the rate of change of force both in the time and the frequency domain bears the greatest promise of practical improvement as such modifications are simple and relatively easy to implement. Tool sets are compact and are not in themselves high noise radiators unless they are badly designed and give rise to ancillary impacts. They can be looked upon therefore as a means of modifying the pulse shapes without alteration to the other characteristics of the machine.

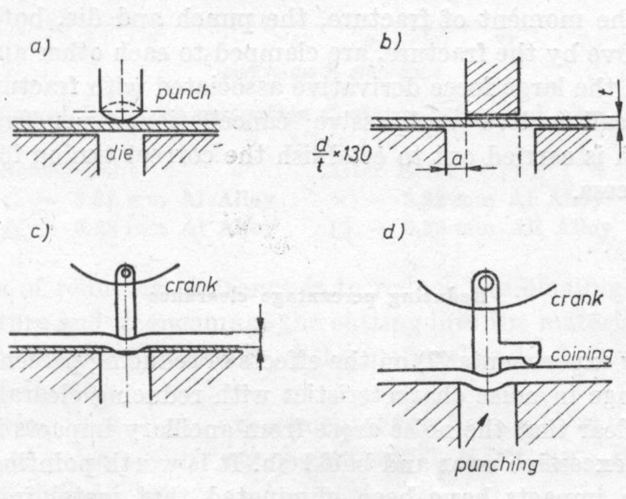

Fig. 2. Methods of reducing $\Sigma[\dot{f}(t)_{\max}]^2$: (a) using sheared cutters to provide longer fracture time, (b) reducing percentage clearance between punch and die, (c) adjusting maximum cutting depth to reduce inertia forces in press at fracture, (d) combining punching with coining

Figure 2 shows some of the many tool modifications that can be made to reduce the force derivatives and therefore the noise radiated. These may be listed as follows:

(1) Reducing the percentage clearance between the punch and die sections of the punch set so as to increase the cutting into the workpiece prior to fracture and thus reducing the size of the fracture force derivative. The disadvantage lies in the necessity for more accurate tooling and increased heating and wear of the punch.

(2) Providing a small degree of shear in the punch or die in order to encourage a progressive fracture around the periphery of the cutter. This provides a longer cutting pulse and a smaller force derivative. The objection to this is the poor finish and the possible need for a further operation to eliminate distortion of the component.

(3) Raising the tool bottom dead centre to give a reduced inertia force following fracture. The ideal would be for the punch to begin to lift off the workpiece at the instant of fracture so that the strain energy which needs to readjust itself in the frame is kept to a minimum. The disadvantage lies in the greater risk of incomplete fracture and the problem of ejecting the blanks.

(4) Coupling the fracture to some amount of coining, e.g., to halt the punch via a relatively slow surface forming (coining) stage. This can be advantageous in certain instances where forming is needed, but there are obvious limitations on when this can be used.

(5) Combining several operations so that the overall rate of change of force is minimized. This needs very careful construction of multiple punch tool sets to optimize the fracture sequences.

(6) If, at the moment of fracture, the punch and die, both of which are made free to move by the fracture, are clamped to each other and then allowed to move slowly, the large force derivative associated with fracture can be minimized. This we call "active" or "passive" cancellation, depending on the degree of sensing which is carried out to establish the correct timing for the clamping or freezing process.

Reducing percentage clearance

In our early experiments (2) on the effects of reducing percentage clearance, little or no change in noise characteristics with reducing clearance was found, and it became clear that the noise arose from ancillary impacts in the bearings associated with excessive wear and backlash. It is worth pointing out that it is only when such impacts have been eliminated that metal fracture and the associated redistribution of strain energy in the whole machine becomes the basic noise mechanism. Fig. 3 shows the measured noise output from the ISVR C-frame open press with increasing percentage clearance (defined as the average punch gap, divided by the material thickness and multiplied by one hundred) before and after the bearings had been replaced. It may be seen that with poor bearings, the noise reduction achieved by reducing the clearance is minimal. What follows describes noise reductions on well-maintained punch presses.

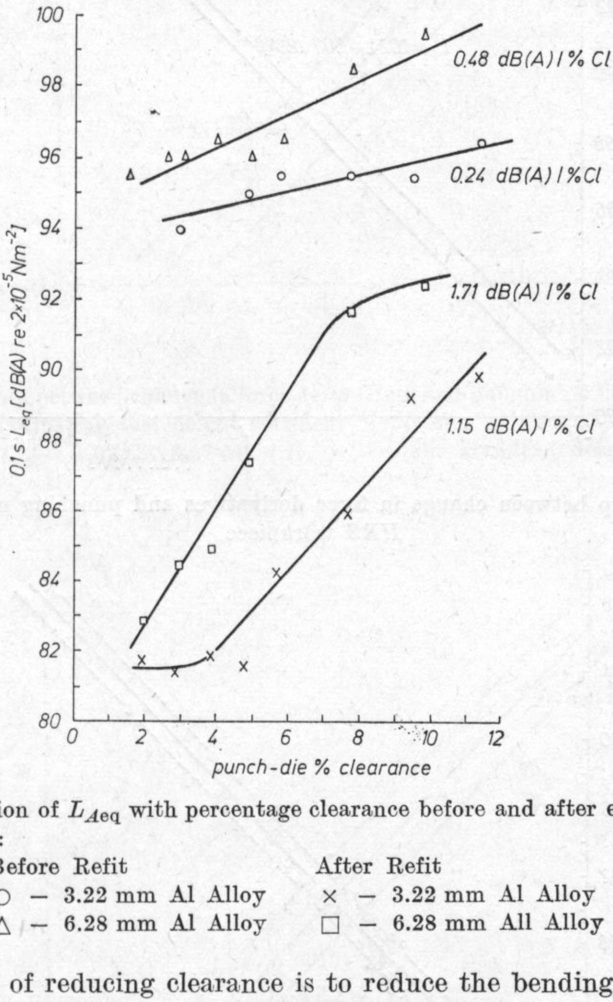

Fig. 3. The variation of $L_{A\mathrm{eq}}$ with percentage clearance before and after elimination of backlash in bearings:

Before Refit	After Refit
○ – 3.22 mm Al Alloy	× – 3.22 mm Al Alloy
△ – 6.28 mm Al Alloy	□ – 6.28 mm All Alloy

The effects of reducing clearance is to reduce the bending moment in the vicinity of fracture and to encourage the cutting into the material by the punch. This reduces the effective thickness of the material and a lower punch force at the moment of fracture.

It is gratifying that all the experimental results can be collapsed on a straight line curve of noise against our parameter $L_{\dot{f}}$ this time expressed in terms of a force level defined by

$$L_{\dot{f}} = 10 \log_{10}\left[\frac{\sum\left(\dot{f}(t)_{\max}\right)^2}{f_R^2}\right],$$

where f_R is the reference force derivative of 1 Mn·s^{-1}.

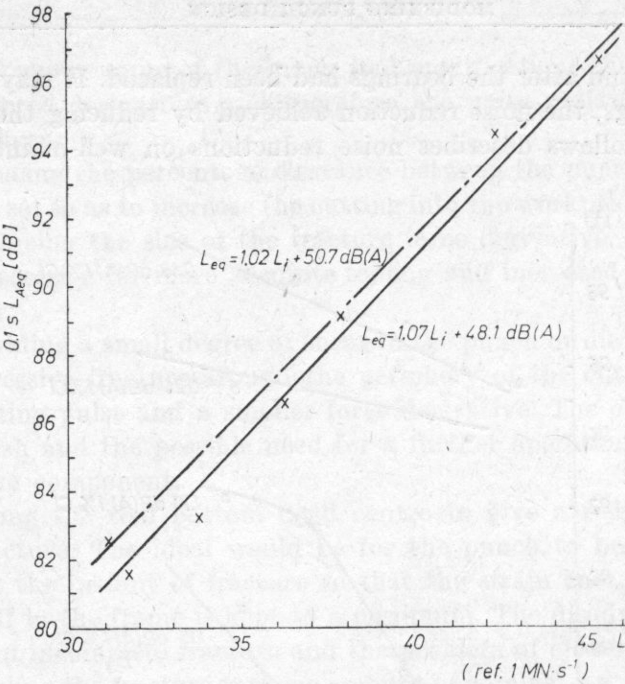

Fig. 4. Relationship between change in force derivatives and punching noise for 3.01 mm *HRS* workpiece

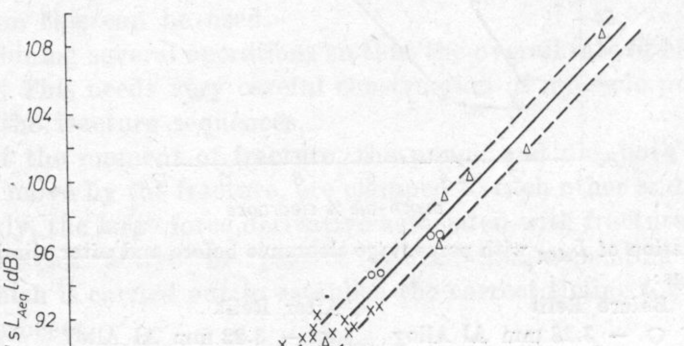

Fig. 5. Relationship between change in force derivatives L_f and punching noise L_{eq} for all measurement conditions. Workpiece material: × — aluminium alloy, ○ — hot rolled steel, △ — bright drawn steel, ——— $L_{eq} = 1.02\ L_f + 50/\mathrm{dB}(A)$, – – – one standard derivation limits. Best straight line fit to all measurement points (except those correspnding to $L_f < 28$ dB) $L_{eq} = (1.02 \pm 0.025) L_f + (50.7 \pm 1.0)\,\mathrm{dB}(A)$

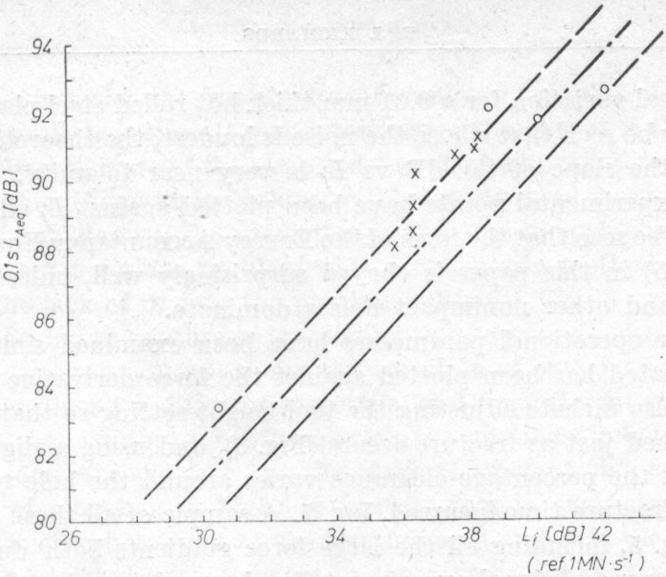

Fig. 6. Relationship between change in force derivatives and punching noise for measurements on the effect of adjusting tool height settings: × 3.2 mm Al alloy; ○ 6.28 mm Al alloy; $-\cdot-\cdot-L_{eq} = 1.02 L_{\dot{f}} + 50.7$ dB (A), — — — one standard derivation limits

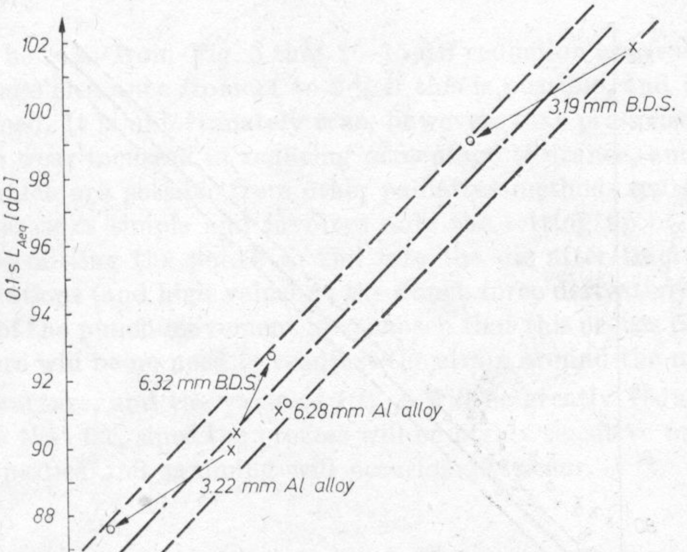

Fig. 7. Relationship between change in force derivatives and punching noise for measurements on the effect of an eccentric die × concetric die; ○ eccentric die; $-\cdot-\cdot- L_{eq} = 1.02 L_{\dot{f}} +$ $+ 50.7$ dB (A); — — — one standard derivation limits

One typical variation for a 3.01 mm thick hot rolled steel plate is shown in Fig. 4. It may be seen that where the noise is loudest, the theoretical prediction is good and the slope of the dB vs $L_{\dot{f}}$ is very near to unity.

All the experimental points have been plotted against $L_{\dot{f}}$ on one graph in Fig. 5. It may be seen that the form of the Energy Accountancy Equation derived as equation (5) in this paper is obeyed surprisingly well, unless the fracture noise is low and other nonimpact noises dominate.

The other operational parameters have been examined similarly (3) and the noise radiated has been plotted against the force derivative factor. These experiments also include adjusting the tool height setting so that bottom dead centre is reached just as fracture occurs (Fig. 6), and using a slightly eccentric punch so that the percentage clearance varies around the hole to be punched and a longer fracture time incurred, Fig. 7. A sample of all these results is also plotted in Fig. 5, including all the large force gradients both during fracture build-up and any additional experimentally observed transients occurring due to backlash. It may be seen that there is good agreement between the values of L_{Aeq} averaged over a 0.1 s period, and $L_{\dot{f}}$ the standard deviation of one decible from the best fit line which in turn implies a 2 per cent greater slope than the theoretically derived unity, and a standard deviation about this line of 0.025 $L_{\dot{f}}$.

Needless to say, the agreement is less acceptable, particularly for thin materials and low values of $L_{\dot{f}}$. It may be seen that while agreement is good for

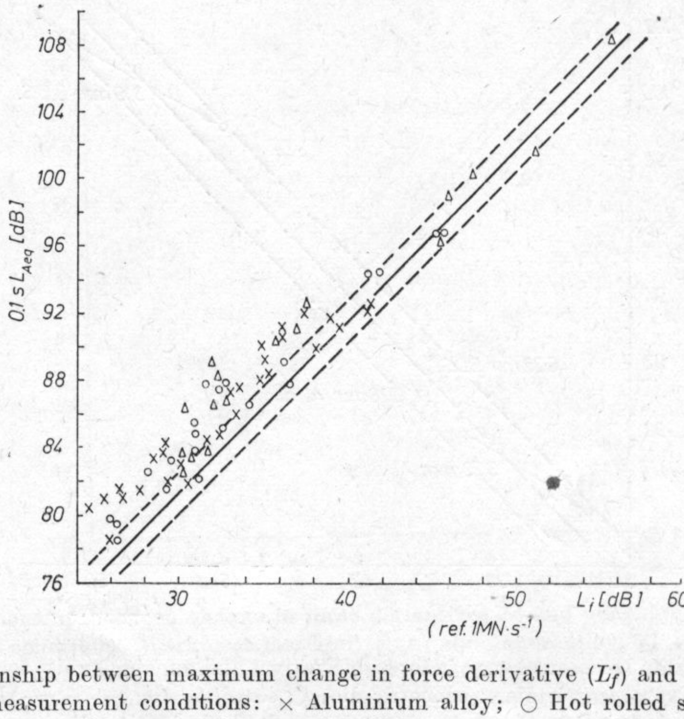

Fig. 8. Relationship between maximum change in force derivative ($L_{\dot{f}}$) and punching noise (L_{eq}) for all measurement conditions: × Aluminium alloy; ○ Hot rolled steel; △ Bright drawn steel, ——— $L_{eq} = 1.02\ L_{\dot{f}} + 50.7$ dB (A), – – – – one standard derivation limits

noisy processes in which the fracture forces dominate the noise output, the noise prediction formula (equation (5)) underestimates the output if the punch force build-up and backlash are of the same order as the fracture forces. This situation often occurs when the press is working well below its rated maximum force, and under such circumstances, the prediction is poor. As in such circumstances the total noise is not a serious problem, this deficiency in method is not serious.

In fact the lack of fit into the prediction line of Fig. 8 provides a useful method of checking the nature of the dominating source of noise for that condition. Table 4 (not reproduced) shows the variation of both best fit gradients of such prediction lines for different materials, and for different percentage clearances. It may be seen that for soft materials the relationship between noise and fracture force derivative is poor but the residual constant "b", representing the general level of vibration, is high. It is clear that in such circumstances the impact point is elsewhere than at the fracture area and that the structural response terms associated with impacts in these areas are greater.

Predictions are still possible, if the full strength of the energy accountancy equation is incurred, just as they have been using the transfer mobility methods used in (4) to deal with the noise radiated from the surface of a diesel engine frame subject to combustion forces on the piston.

Altering punch settings, the use of eccentric punches and using sheared cutters

It may be seen from Fig. 5 that 10–15 dB reduction are feasible by reducing percentage clearance from 11 to 2 % if this is possible, and if the press is well-maintained. It is unfortunately true, however, that press users to not like the excessive wear incurred in reducing percentage clearance, and the scope of reductions which are possible from other palliative methods must be explored.

One of these is simple and involves only the setting up of the punch; if instead of permitting the punch to run into the die after fracture, incurring sharp accelerations (and high values of the punch force derivatives), the bottom dead centre of the punch movement is so chosen that this occurs just as fracture happens, there will be no need to readjust the strain around the machine at the moment of fracture, and the value of $\dot{f}(t)_{\max}$ will be greatly reduced. It is not feasible to go that far, since the process will be highly sensitive to variations in material properties and jamming will occasionally occur.

Sheared or progressive punch design

Having established firmly that in their most noisy conditions punch press noise is directly related to the time rates of change of punch force and that once such large rates have been designed into tooling, reduction is extremely difficult to achieve, consideration needs to be given to minimizing such force derivatives at the design stage by deliberately lengthening the fracture process by encouraging progressive fracture around the hole to be punched or by overlapping sensibly and integrating the punching of several holes and coining operations.

This can be done (a) by making less uniform the percentage clearance between the punch and its cavity around the hole by fitting the punch eccentrically: (b) by shearing the cutter by a small amount so that the fracture travels progressively around the hole: or (c) by selecting the progression of cutting times of different holes in such a way that the total load is less impulsive.

Experiments on all these possibilities have been carried out at ISVR and are reported here, not so much to provide a detailed record, as much as to indicate the promise to be obtained using such pallative methods, and to raise suggestions regarding tool design.

Experiments on the effects of sheared punches have been more successful than any of the previous methods. Shear cutting is often used as a means of reducing the maximum punch load both on a single cutter and on a multi-punch tool involving progressive cutting of a series of single holes in the same action.

It is interesting to examine the theoretical design associated with shear, since the need is for a uniform reduction of the punch force derivative rather than to space the fractures and individual events. Such calculations have been made

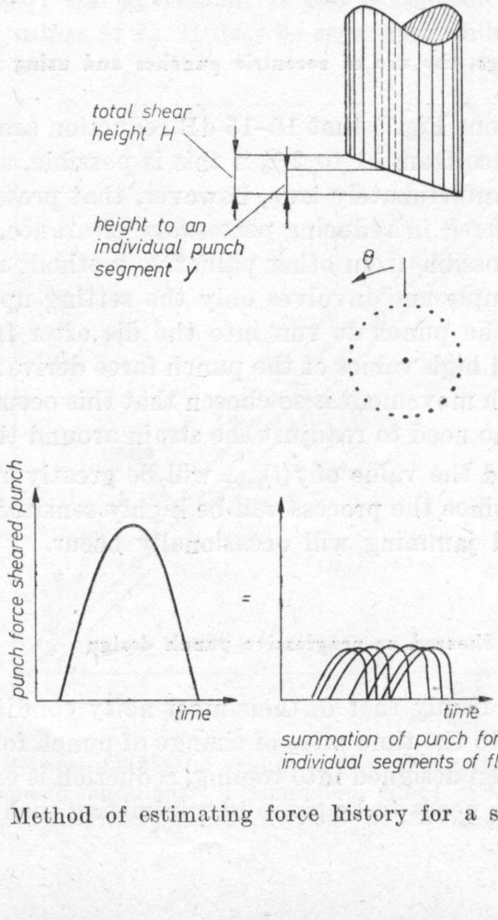

Fig. 9. Method of estimating force history for a sheared punch

for the sheared cutter shown in Fig. 9, each fracture around its periphery being assumed to occur at delayed times determined by the displacement. Fig. 10 illustrates the resultant calculated force history for varying shear angles. It may be seen that there is little point in going beyond three or four degrees of shear for the case being considered. This is illustrated more clearly in a study of the estimated values of the force derivative parameter L_j shown in Figs 11(a)-(d).

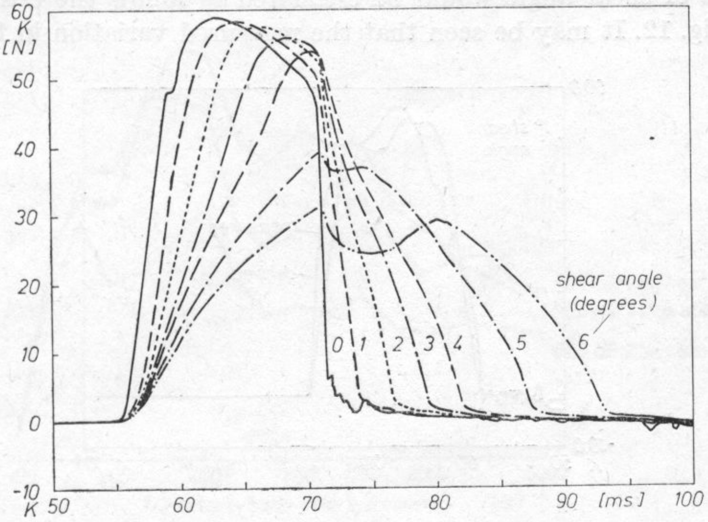

Fig. 10. Derived force histories for various punch shear angles

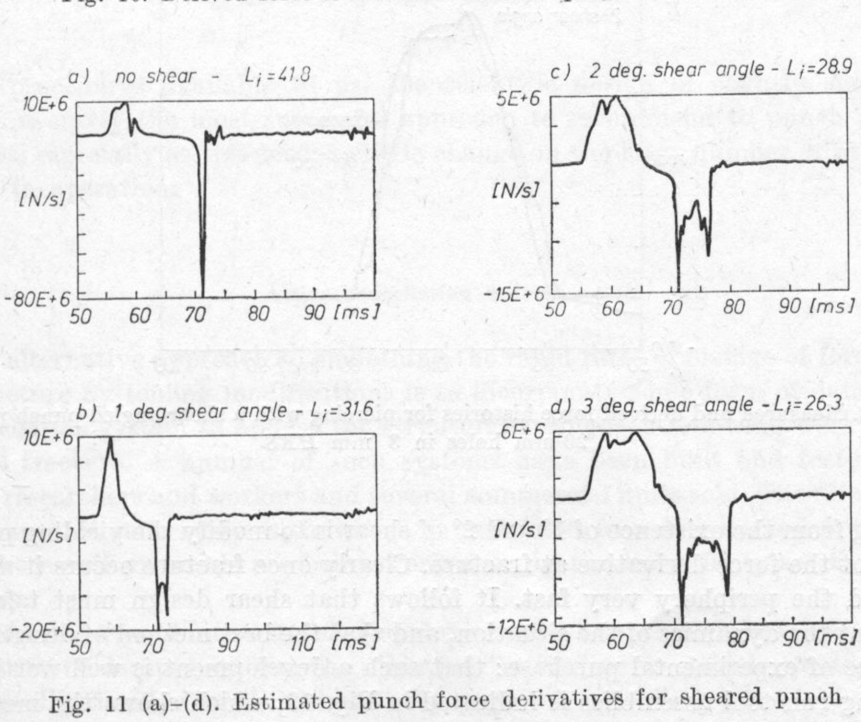

Fig. 11 (a)–(d). Estimated punch force derivatives for sheared punch

The estimated value of L_f falls from 41.8 to 31.6 for one degree of shear angle with very little worthwhile additional reduction below three degrees of shear. As the use of such small angles gives no great workpiece distortion it is thought that small shear cutting tools may be used with no further work needed.

If fracture were to occur according to the calculated local displacement of the tool around its periphery, the expected force-time diagrams with one and two degrees of shear angle would be estimated to follow the estimated paths shown in Fig. 12. It may be seen that the measured variation in force pattern

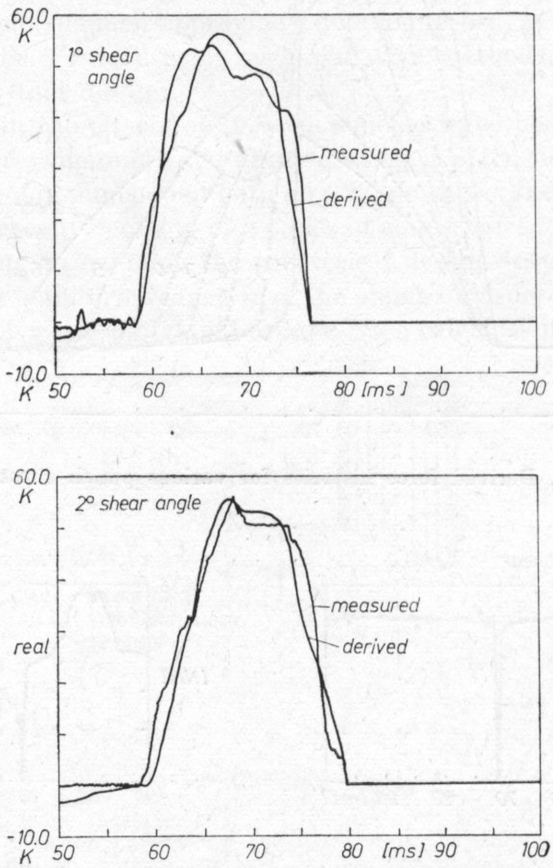

Fig. 12. Measured and derived force histories for piercing with a shear angled punch piercing 20 mm holes in 3 mm *HRS*

arising from the existence of 1° and 2° of shear is to modify the yielding pattern but not the force derivative at fracture. Clearly once fracture occurs it spreads around the periphery very fast. It follows that shear design must take into account the dynamics of the situation, and that the best method of design is via the use of experimental purchase: that such a development is well worthwhile on long run tool assemblies is indicated in Fig. 13, which shows the measured noise spectra (L_{eq}) for 0.1 and 2 degrees of shear. The reduction in the linearly weighted noise level is 12 dB: since so much of the noise energy is around frequencies 200–500 Hz, an additional 2 dB reduction would be recorded if A-weighting of the spectra had been carried out.

To conclude this section, it is probably true to say that of all the noise

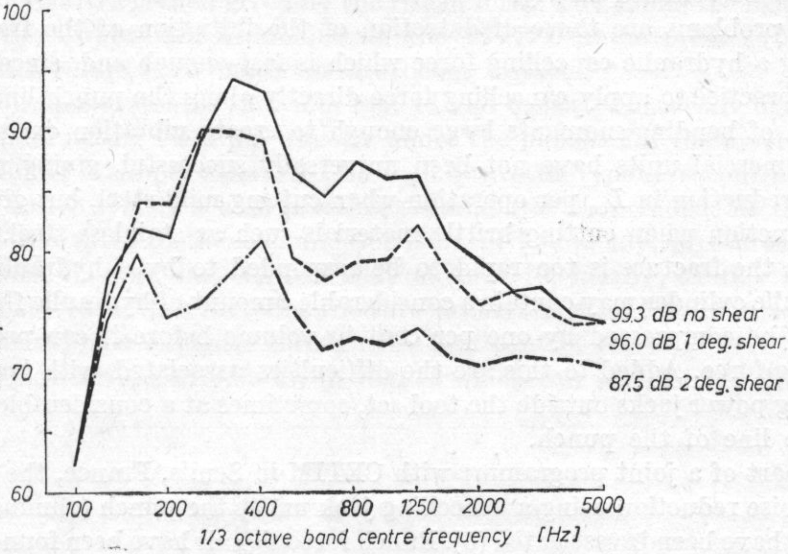

Fig. 13. *SPL* spectra — sheared punch

control procedures available to us, the scientific design of slightly sheared punches is surely the most successful approach to recommend to punch press designers, especially as this needs so little change on the large number of presses already in operation.

Active cancellation devices

An alternative approach to smoothing the rapid rates of change of force on the structure by tooling modifications is to incorporate some form of damping or cancellation system to oppose the structural springback following workpiece material fracture. A number of such systems have been built and tested by various researchers and workers and several commercial units sold. The effective performance of such systems, however, is restricted to a few limited applications where soft materials are being worked or on the less stiff types of press structure.

Two categories of force cancellation device exist, one which is best described as a passive or damper system, the other an "active" device which provides an electronically timed, though hydraulically powered, cancelling force to replace the tooling forces at the instant of material fracture. Both aim at reducing the sudden springback of the structure following fracture, thus smoothing the sudden release of strain energy around the whole machine. It is the sudden release which gives rise to the high frequency vibrations replacing it by a subsequent "leak" system, which cuts down these high frequencies and allows the strain energy to be released slowly.

The problems are those of detection of the initiation of the fracture, of obtaining a hydraulic cancelling force which is fast enough and, since it is difficult in practice to apply cancelling force directly along the punch line, the elimination of bending moments large enough to create vibration excitation.

Commercial units have not been universally successful, giving up to ten decibels reduction in L_{eq} per operation when cutting mild steel, but giving very little reduction when cutting brittle materials such as stainless steel. In these instances the fracture is too rapid to be responded to by a hydraulic device. A hydraulic cylinder may contain a considerable amount of hydraulic fluid which needs to be compressed by one per cent in volume before it can provide the concelling force. Added to this are the difficulties associated with housing the cancelling power jacks outside the tool set, sometimes at a considerable distance from the line of the punch.

As part of a joint programme with CETIM in Senlis, France, the possibilities of noise reduction using a cancelling jack under the punch (eliminating bed bending) have been investigated (5). Punch force signals have been found in practice to be surprisingly repetitive and similar to each other and consistent enough to use a simple time delayed pulse triggered at a fixed point in the press cycle to operate the cancellation device.

The work was aimed basically at investigating the performance and noise

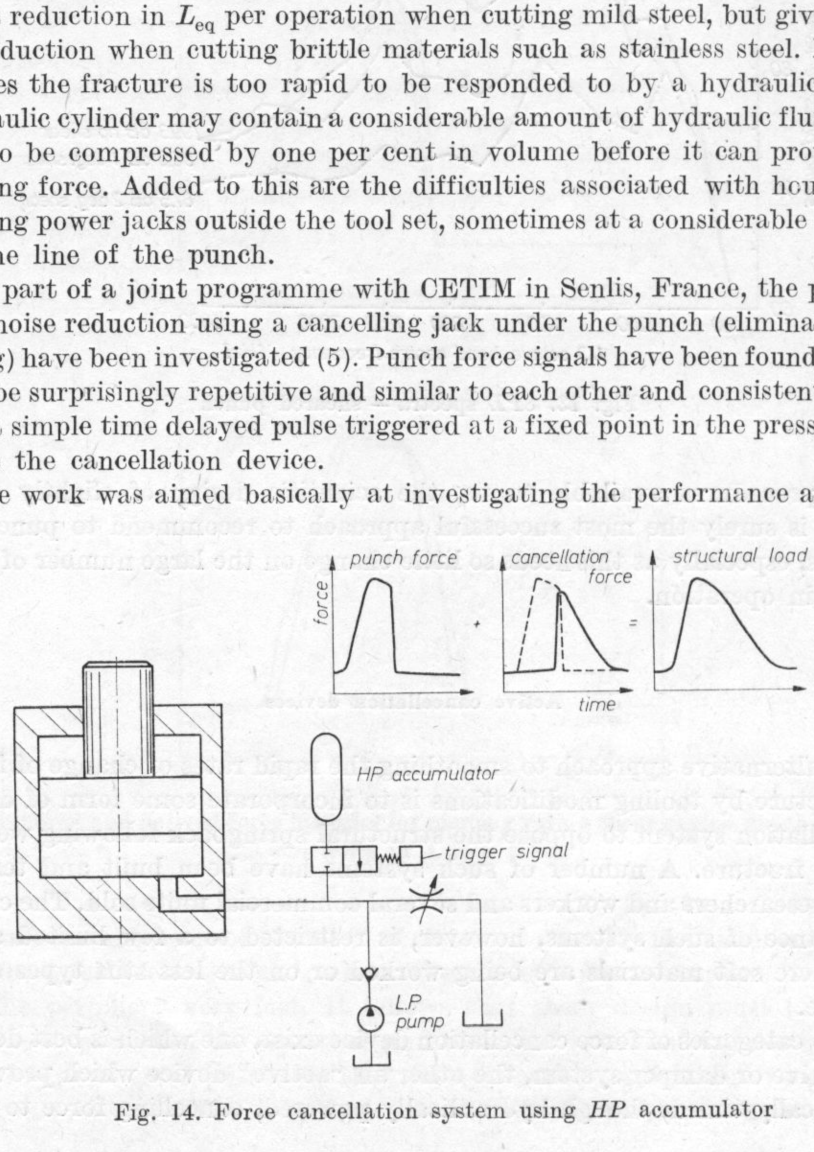

Fig. 14. Force cancellation system using HP accumulator

reductions possible with such systems. There is of course a limit to the noise reductions possible by smoothing the release of strain energy from the structure. This is related to the rate of rise of force. This in turn is determined by the punch velocity on contact with the workpiece. Smoothing the release of strain energy below that of the rate of rise will not give any further noise reduction as the peak force derivative is then given by the rise of force. This limits the maximum noise reductions which are attainable on the ISVR C-frame press to around 10 dB when medium hard materials are being worked.

The hydraulic system is shown in Fig. 14 and consisted basically of an hydraulic cylinder acting via a pin directly under the punch. For the passive system this cylinder is simply exhausted through a throttle-type flow control valve but for the active system a high pressure accumulator is included. At the correct instant the valve to the accumulator is opened suddenly, pressurising the cylinder which then applies the cancellation force to, ideally, exactly replace the force removed as the material fracture occurs. The load is then released slowly as the cylinder exhausts through the flow control valve.

Two typical results are shown in Fig. 15 which can be taken to illustrate

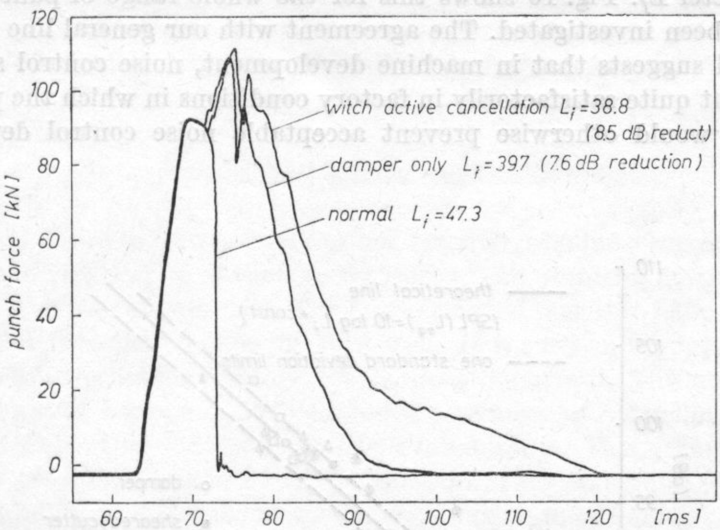

Fig. 15. Effect of active cancellation – 3 mm *BDS*

the end of this particular investigation. In this work, noise was not measured as such, since the measurement of

$$10 \log \frac{\sum [f(t)_{\max}]^2}{f_{\text{ref}}^2}$$

was found to be simpler and a reliable measure of machine noise output. It may be seen that ultimately a noise reduction of 8.5 dB was estimated to occur with active cancellation, 7.6 dB reduction with the passive damper. On the other hand, at the beignning, the introduction of the damper gave only 1.7 dB reduction from that from a normal system. The decrease can therefore be traced more to the timing of the punch "freezing" process rather than to the details of the pulse cancellation.

This is generally true in all our work, and provides a clear warning in the use of such methods. Fig. 15 shows clearly the sharp succession of high force gradients which occur if timing is not right. If this is not kept in mind, the total effect of the cancellation device will be to add additional vibrational energy, risking distortion and possibily damage to the press, and a possible increase in the total noise radiated.

Noise measurements against values of L_f

Throughout the latter part of the work described in this paper, spot checks have been made of the measured noise levels L_{eq} (0.1 s) against our punch design noise parameter L_f. Fig. 16 shows this for the whole range of punch variables which have been investigated. The agreement with our general line is very satisfying, and suggests that in machine development, noise control studies can be carried out quite satisfactorily in factory conditions in which the presence of other noises would otherwise prevent acceptable noise control development.

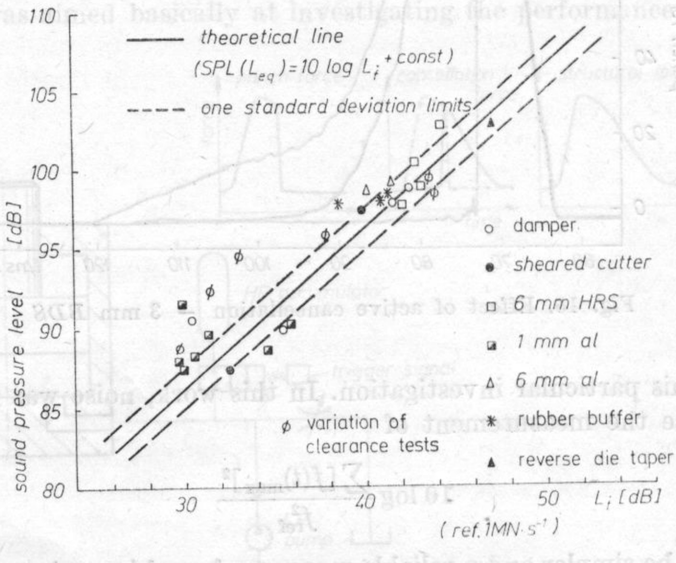

Fig. 16. Sound pressure level versus punch force derivative parameter for various piercing conditions

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ACOUSTIC PROBLEMS IN MULTI-STOREY RESIDENTIAL BUILDINGS RAISED BY MEANS OF INDUSTRIALIZED TECHNOLOGY METHODS

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The paper comprises results of subjective (social survey) and objective tests (measurements) of the acoustic climate in multi-storey buildings raised by industrialized methods. On the basis of performed tests the paper also discusses factors influencing the sound insulation in dwellings, as well as opinions concerning designing.

1. Introduction

People's desire to own a dwelling has recently become very common in almost all countries despite of their social status. The problem, however, is solved differently in different countries. The most essential task is to build a great number of dwellings in the possibly shortest time. It is such an urgent problem that sometimes even economic factors become less important. The number of new dwellings built in Europe in 1975 exceeded a number of 8 dwellings/1000 inhabitants. In Poland — 5.8 dwellings/1000 inhabitants. This certainly does not satisfy the growing demand of the population. That is why in Poland it was planned to build in 1979 340000 dwellings, index — 9.5 dwellings (1000 inhabitants). The realization — 274000. Population's demands are however greater, about 400000 dwellings every year. It requires further speeding-up of the tempo of rising buildings. According to specialists' opinions it is possible only in such a case, when buildings are built by means of industrialized methods as multi-storey residential buildings. At this point it is essential not to worsen the quality of buildings with such a big tempo of building rising.

By quality we also mean a proper acoustic climate which depends on many factors discussed further in this paper.

All multi-storey residential buildings raised by industrialized technology can be divided into five groups:

a) multi-block buildings built from blocks having with from 90 to 120 cm (or wider with a mode of 30 cm) and with length such as the length of a storey and in case of partitions — with of a bearing tract of a building;

b) framework buildings built from concrete frames — prefabricated or poured, or steel frames filled with light block walles or partitions;

c) monolithic, performed from poured concrete in climbing shuttering;

d) multi-block buildings performed from concrete large blocks having the dimensions of the whole room;

e) multi-spatial element buildings formed from spatial elements comprising one or two rooms (walls and partitions), completely equipped.

The above specification does not include buildings raised according to traditional methods which are not a subject of this description.

2. Subjective and objective tests on the acoustic quality of dwellings

As it is well known there are large discrepancies between the walls and partitions, insulation values measured in a laboratory and in a building. These discrepancies are caused by flanking transmission and very often by bad quality of assembling leakages at joints, bad fillings of assembly apertures partitions, small damages of elements. The value of acoustic insulation of partitions in buildings is always smaller than measured in a laboratory. Considering a common construction of buildings by means of industrialized methods it is necessary to have not only concerning measured transmission loss of these partitions but also a subjective evaluation of the insulation by inhabitants. Also, it is necessary to know to what extent a given partition meets the standard requirements and how far this is accepted by the inhabitants. In order to explain all these questions the Department of Acoustics, ITB [1] has performed tests on objective and subjective acoustical insulation of partitions in residential buildings which are raised according to the most common industrialized systems (Figs. 1–4), namely:

a) multi-block systems: "Szczeciński" and W-70,

b) monolithic system,

c) ferroconcrete frame system.

Objective tests (measurements) performed according to PN-68/B-02154 and ISO R 140 in many buildings of the above mentioned systems have proved that partitions and interdwelling walls satisfy the standard requirements ($E_L \geqslant -1, E_T \geqslant 0$). Solely, in the frame system ("Rama H") the insulation index of inter-dwelling partitions E_L was lower by about 4 dB from the required values. The following values were measured: transmission loss of floors and ceilings and interdwelling partitions R, and insulation D_N between kitchens, bathrooms, ante-chambers (with installation pipes)

$$R'_w = L_1 - L_2 + 10 \log S/A \quad [\text{dB}], \tag{1a}$$

while

$$D_N = L_1 - L_2 + 10 \log A/A_0 \quad [\text{dB}], \tag{1b}$$

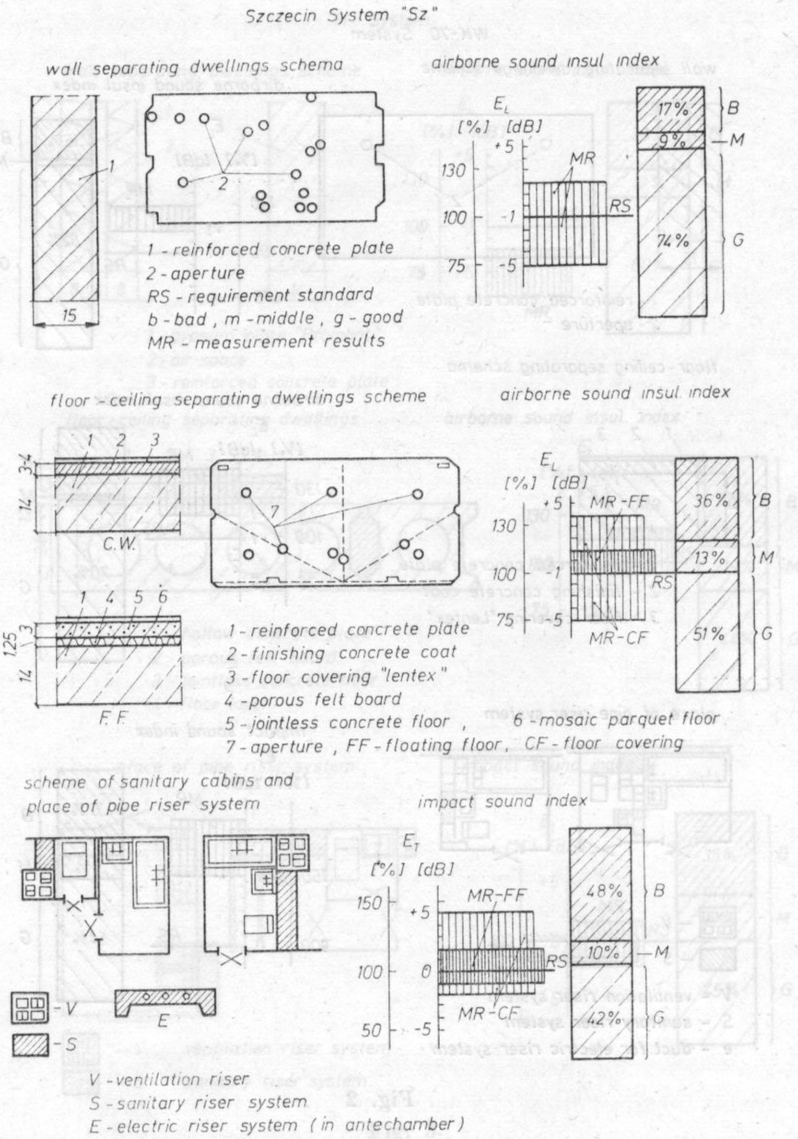

Szczecin System "Sz"
wall separating dwellings schema
airborne sound insul index
1 - reinforced concrete plate
2 - aperture
RS - requirement standard
b - bad , m - middle , g - good
MR - measurement results
15
E_L
[%] [dB]
130
100
75
+5
-1
-5
MR
RS
17%
9%
74%
B
M
G
floor - ceiling separating dwellings scheme
airborne sound insul index
C.W.
F.F.
1 - reinforced concrete plate
2 - finishing concrete coat
3 - floor covering "lentex"
4 - porous felt board
5 - jointless concrete floor , 6 - mosaic parquet floor
7 - aperture , FF - floating floor, CF - floor covering
MR - FF
MR - CF
36%
13%
51%
scheme of sanitary cabins and
place of pipe riser system
impact sound index
E_I
[%] [dB]
150
100
50
+5
0
-5
MR-FF
MR-CF
48%
10%
42%
V
S
E
V - ventilation riser
S - sanitary riser system
E - electric riser system (in antechamber)

Fig. 1

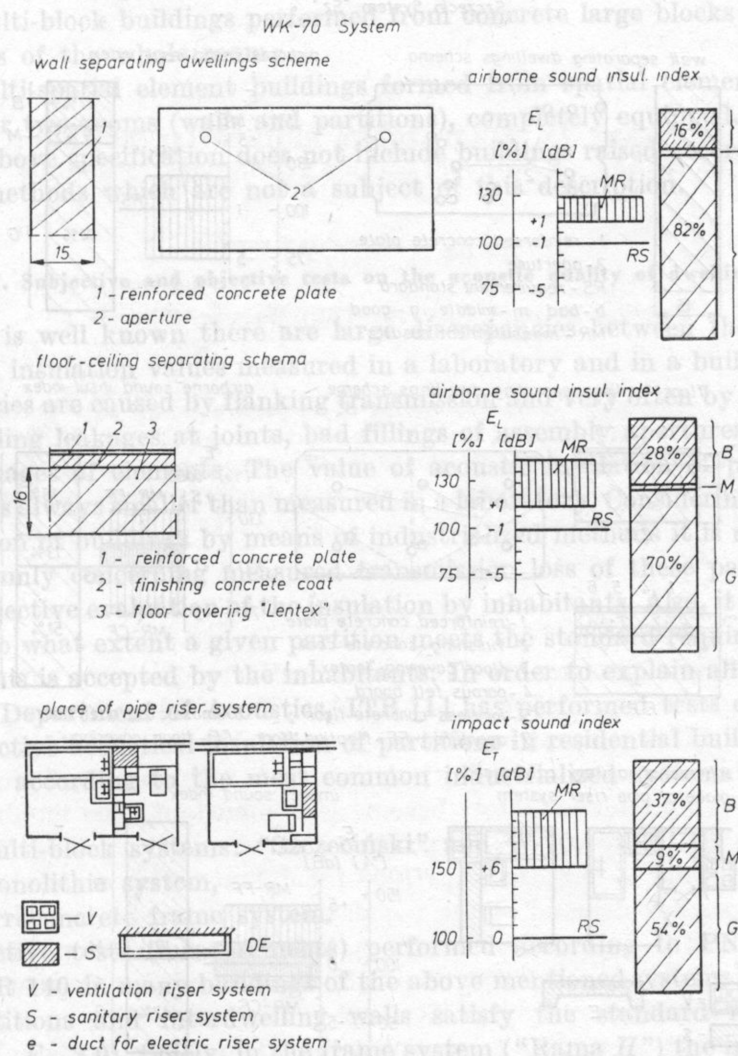

WK-70 System
wall separating dwellings scheme
airborne sound insul. index
15
1 - reinforced concrete plate
2 - aperture
E_L
[%] [dB]
130
100
75
+1
-1
-5
MR
RS
16%
82%
B
M
G
floor-ceiling separating schema
1 2 3
16
1 - reinforced concrete plate
2 - finishing concrete coat
3 - floor covering "Lentex"
28%
70%
place of pipe riser system
impact sound index
E_T
150
100
+6
0
37%
9%
54%
V
S
DE
V ventilation riser system
S sanitary riser system
e - duct for electric riser system

Fig. 2

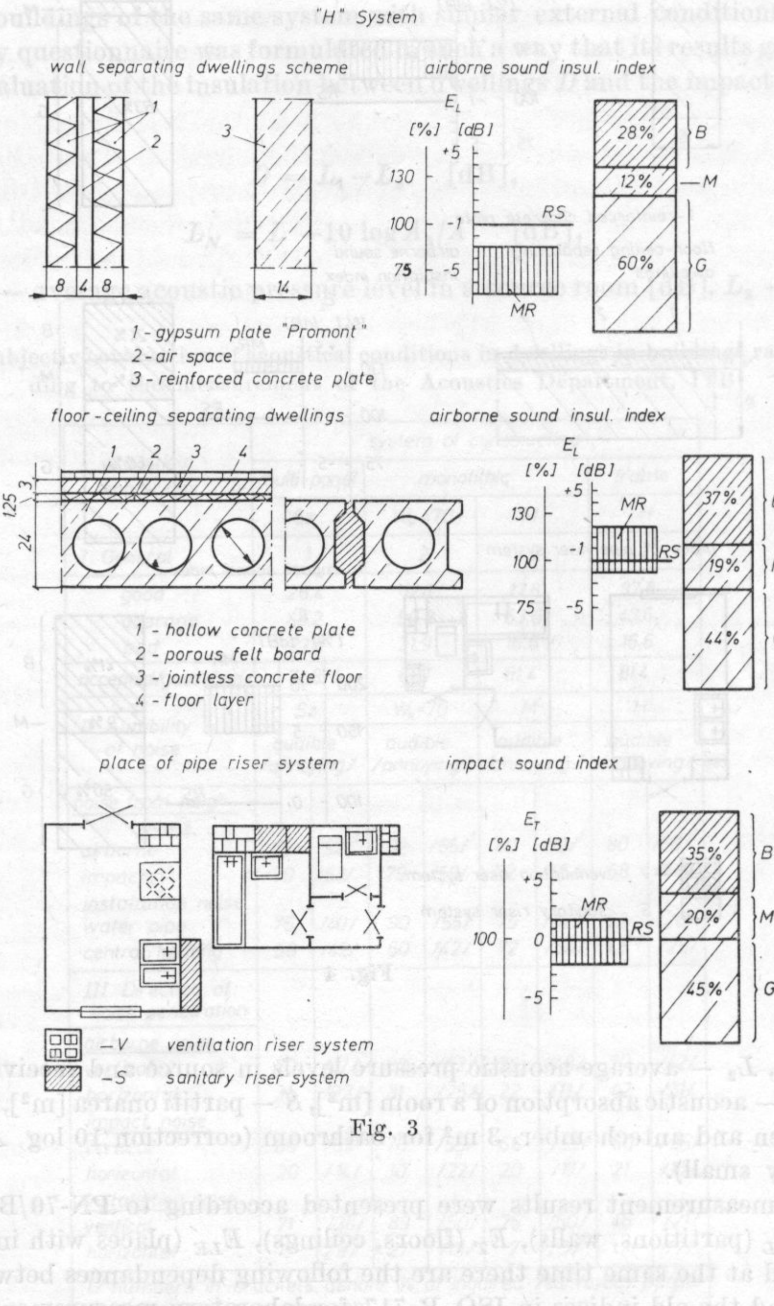

"H" System
wall separating dwellings scheme
airborne sound insul. index
1
2
3
8 4 8
14
E_L
[%] [dB]
+5
130
100
-1
75
-5
RS
MR
28%
12%
60%
B
M
G
1 - gypsum plate "Promont"
2 - air space
3 - reinforced concrete plate
floor - ceiling separating dwellings
airborne sound insul. index
1 2 3 4
125
3
24
E_L
[%] [dB]
+5
130
-1
100
75
-5
MR
RS
37%
19%
44%
B
M
G
1 - hollow concrete plate
2 - porous felt board
3 - jointless concrete floor
4 - floor layer
place of pipe riser system
impact sound index
E_T
[%] [dB]
+5
100
0
-5
MR
RS
35%
20%
45%
B
M
G
-V ventilation riser system
-S sanitary riser system

Fig. 3

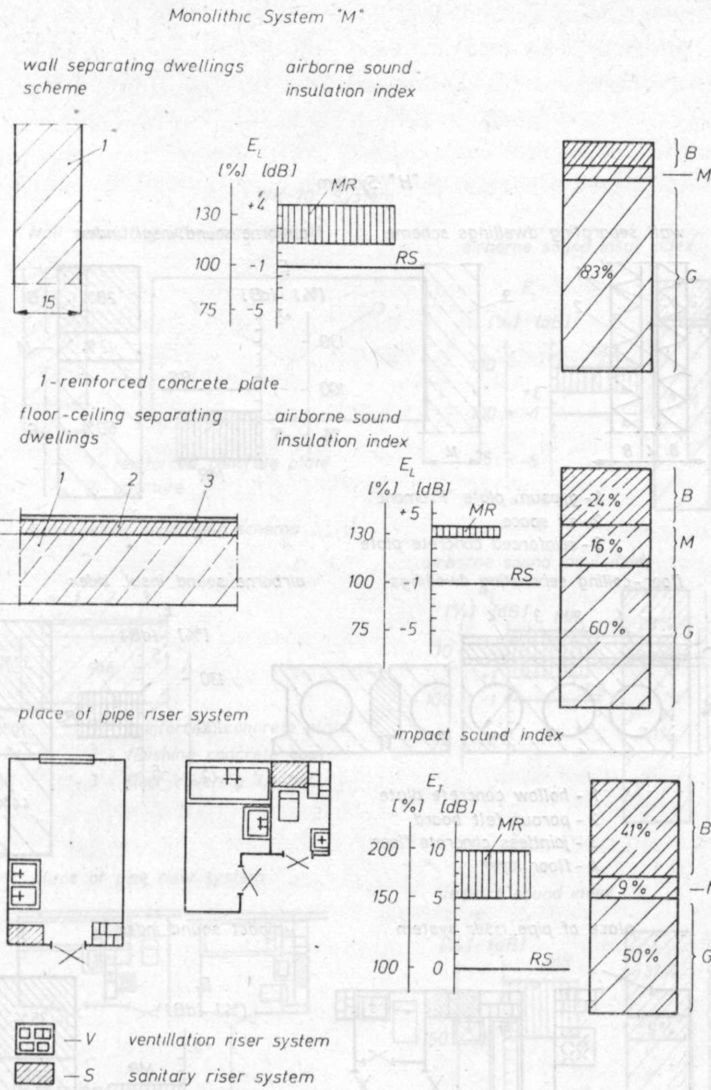


Fig. 4

where L_1, L_2 — average acoustic pressure levels in source and receiving rooms in dB, A — acoustic absorption of a room [m²], S — partiti onarea [m²],A_0 — 5 m² for kitchen and antechamber, 3 m² for bathroom (correction 10 log A/A_0 was neglegibly small).

The measurement results were presented according to PN-70/B-02151 as indices E_L (partitions, walls), E_T (floors, ceilings), E_{LE} (places with installation pipes) and at the same time there are the following dependances between these indices and the old indices in ISO R 717 for laboratory measurements $-I_a =$

$=54+E_L$; for field measurements $-I_a=52+E_L$, $I_a=52+E_{LE}$; despite of a measurement point $-I_i=68-E_T$.

In the same buildings in which the insulation of partitions was measured subjective measurements were performed according to a special social survey, initially checked in some buildings. And so, 160–200 dwellings were tested in different buildings of the same system with similar external conditions. The social survey questionnaire was formulated in such a way that its results gave a statistical evaluation of the insulation between dwellings D and the impact level L_N, and

$$D=L_1-L_2 \quad [\mathrm{dB}], \tag{2a}$$

$$L_N=L-10\log A_0/A \quad [\mathrm{dB}], \tag{2b}$$

where L_1 – average acoustic pressure level in a source room [dB], L_2 – average

Table 1. Subjective evaluation of acoustical conditions in dwellings in buildings raised according to the measurement of the Acoustics Department, ITB

acoustical condicions	system of constructing							
	multi-panel		monolithic				frame	
	Sz		W_k=70		M		H	
I. General	1		2		3		4	
good	26.4		17.6		17.8		37.8	
avarage	48.2		50.3		63.6		43.6	
bad	23.4		31.9		16.6		16.6	
accepted/good+bad/	74.6		68.1		81.4		81.4	
	Sz		W_k=70		M		H	
II. Audibility of noise	audible	/annoying/	audible	/annoying/	audible	/annoying/	audible	/annoying/
noise from neighbours								
airborne	75	/52/[1]	92	/65/[1]	80	/52/[1]	80	/50/[1]
impact	70	/53/	79	/59/	72	/55/	68	/58/
installation noise								
water pipe	76	/40/	90	/55/	75	/52/	52	/26/
central heating	58	/42/	60	/42/	72	/56/	48	/25/
III. Direction of noise penetration								
airborne noise								
vertical	70	/47/	88	/62/	75	/48/	70	/42/
horizontal	31	/22/	31	/25/	22	/11/	42	/31/
impact noise								
vertical	66	/51/	70	/55/	66	/55/	60	/35/
horizontal	20	/14/	30	/22/	20	/17/	21	/14/
installation noise								
vertical	71	/38/	89	/50/	76	/50/	46	/21/
horizontal	13	/9/	21	/13/	7	/5/	17	/7/

1/ numbers in brackets denote % of inquired /surveyed/ people describing noise as annoying pressure level of the noise coming from neighbours [dB], A — absorption of a room [m²], A_0 — reference acoustic absorption, $A_0 = 10\ \mathrm{m}^2$.

Also the L_N values were measured in buildings.

Table 1 illustrates subjective evaluation of the acoustic conditions in buildings representing each system. Acceptable conditions (a sum of good and average evaluations) occur from 68 % for system WK-70, to 81 % for system H. Surveyed people evaluate 17.8 % of the dwellings in the WK-70 system as good and 37.8 % for system H. The inquired (surveyed) people stated that 14 % of the dwellings in the "Rama H System" and 32 % for the WK-70 system had bad conditions. The vertical direction is a dominating direction of noise penetration from dwelling to dwelling. It clearly shows that floors, ceilings and installation ducts are the greatest problem considering acoustic insulation. Analysis of measurement results (partially shown in Table 1) has proved that system WK-70 was evaluated negatively in view of intensive noise exposure of the dwellings adjacent to installation lines. The best subjective evaluation was attained by ferroconcrete walls 15 cm, particulary in the M and WK-70 systems (Tab. 2).

Table 2. Number of inquired people in %, complaining of audiobility of noise annoyance penetrating into residential rooms along partitions or walls, according to measurements of Dep. of Acoustics, ITB

constructing system	construction of floor, ceiling panels	construction of floor	audibility (noise annoyence) %		construction of walls	noise audibility (annoyence) %
			airborne	impact		
Sz	ferroconcrete panel 14 cm	30% of panels-cement setting coat + floor covering and 50% floating floor	49/35/[1]	59/49/[1]	ferroconcrete panel 15 cm	26/19/[1]
W_k-70	ferroconcrete panel 16 cm	cement setting coat + floating floor	30/28/[1]	45/35/[1]	as above	18/16/[1]
M	as above	as above	40/24/[1]	50/40/[1]	as above	15/10/[1]
H	ferroconcrete channel 24 cm /Żerań/	floating floor	56/38/[1]	56/38/[1]	ferroconcrete panel 14 cm or 2 gypsum boards in 6 cm distance	40/27/[1]

1/ numbers in brackets denote % inquired people describing the noise as annoying

Worse evaluation was attained by the same walls in the Sz system, what is clearly evidenced by large discrepancies in the indices E_L measured for these walls (from −5 to +2). This is caused by insufficient quality of performance (bad placing of concrete at joints and assembly apertures). Measurements show that properly built ferroconcrete walls of 15 cm have the index E_L values from +1 to +4 dB and are accepted by 80–90 % of the inhabitants. Faulty performance worsens the insulation of these walls even to $E_L = -5$ dB and the % of people satisfied with the conditions falls below 70. Double gypsum walls 7 cm with distance of 7 cm and a ferroconcrete wall of 14 cm have the index E_L value from -2 to -7 dB. Only 52 % of the inquired inhabitants evaluated these walls positively. The negative evaluations increased up to 28 % after carrying measurements for the walls and it results that at index $E_L \simeq -1 (I_a \simeq 51$ dB) $-$ 20 to 25 % of the inquired persons evaluated these walls as decidedly bad. The analysis has shown that in order to decrease the number of unsatisfied inhabitants to 15 % the index E_L for an inter-dwelling wall would have to be equal $+2$ dB ($I_a = 54$ dB). Inter-dwelling floor and ceiling panels were evaluated subjectively considerably tower than the walls (Tab. 2), from 9 to 13 % negative evaluations more than for the walls in relation to the airborne noise insulation; 29–48 % negative evaluations due to the penetration of impact noise. Best subjective evaluation in view of airborne noise insulation was attained by ferroconcrete

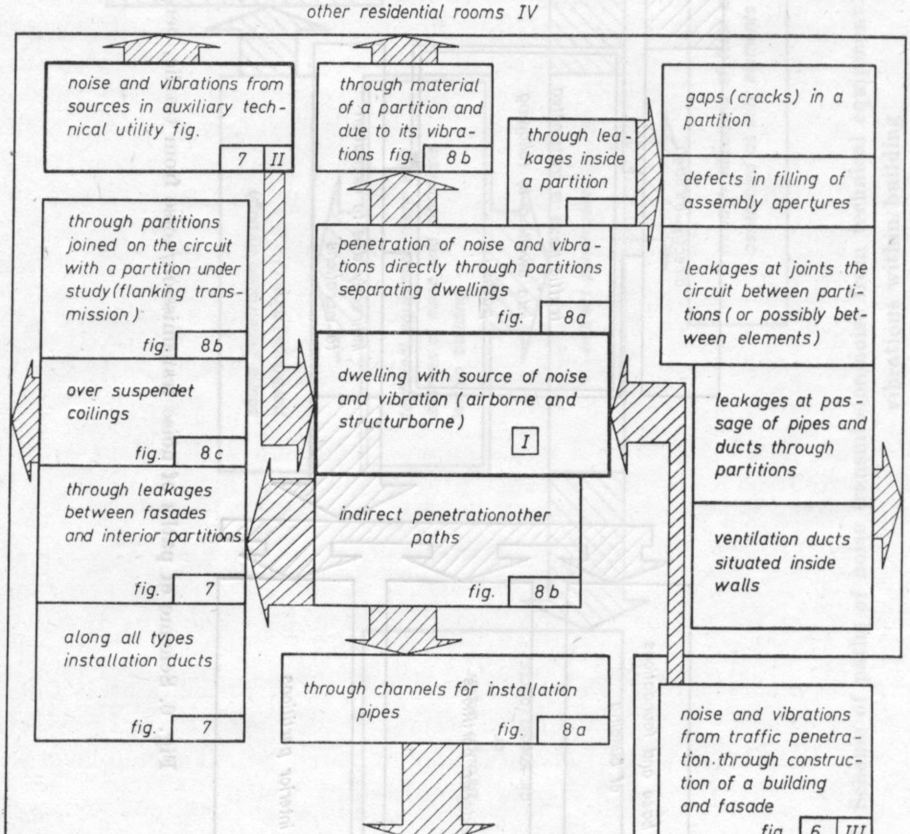


Fig. 5. Collective scheme of sound energy penetration. Sound energy emission from airborne, structure-borne noises sources into residential rooms. I, II, III – position of noise and vibration sources, I – residential dwelling, II – technical utility space in building, III – street with heavy traffic, I, IV – residential rooms protected against noise

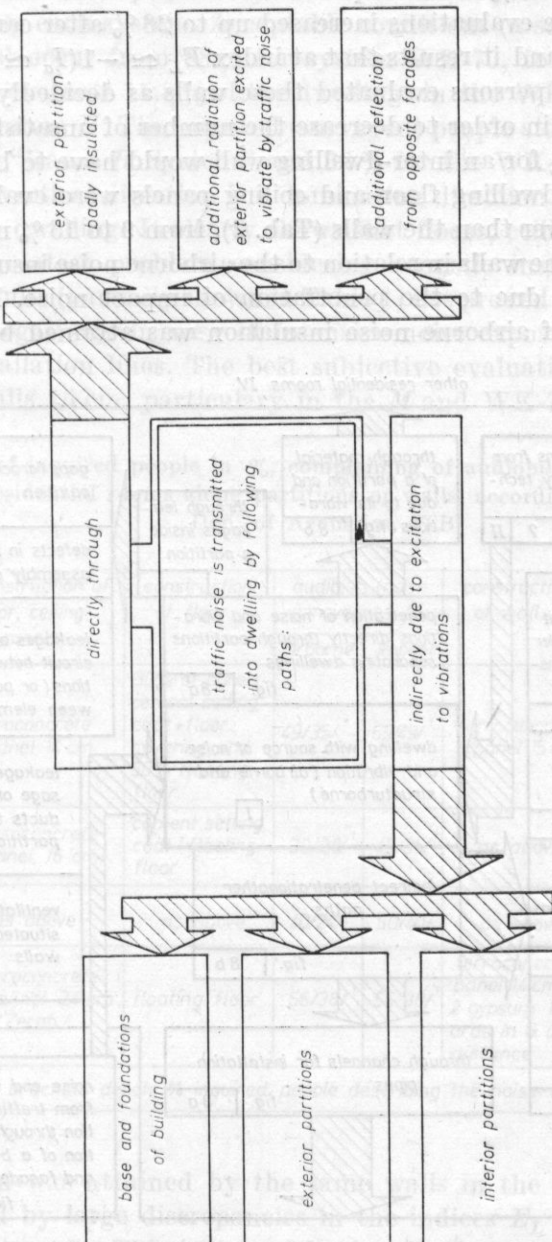


Fig. 6. Scheme of paths of noise transmission/noise from traffic, penetrating into dwellings

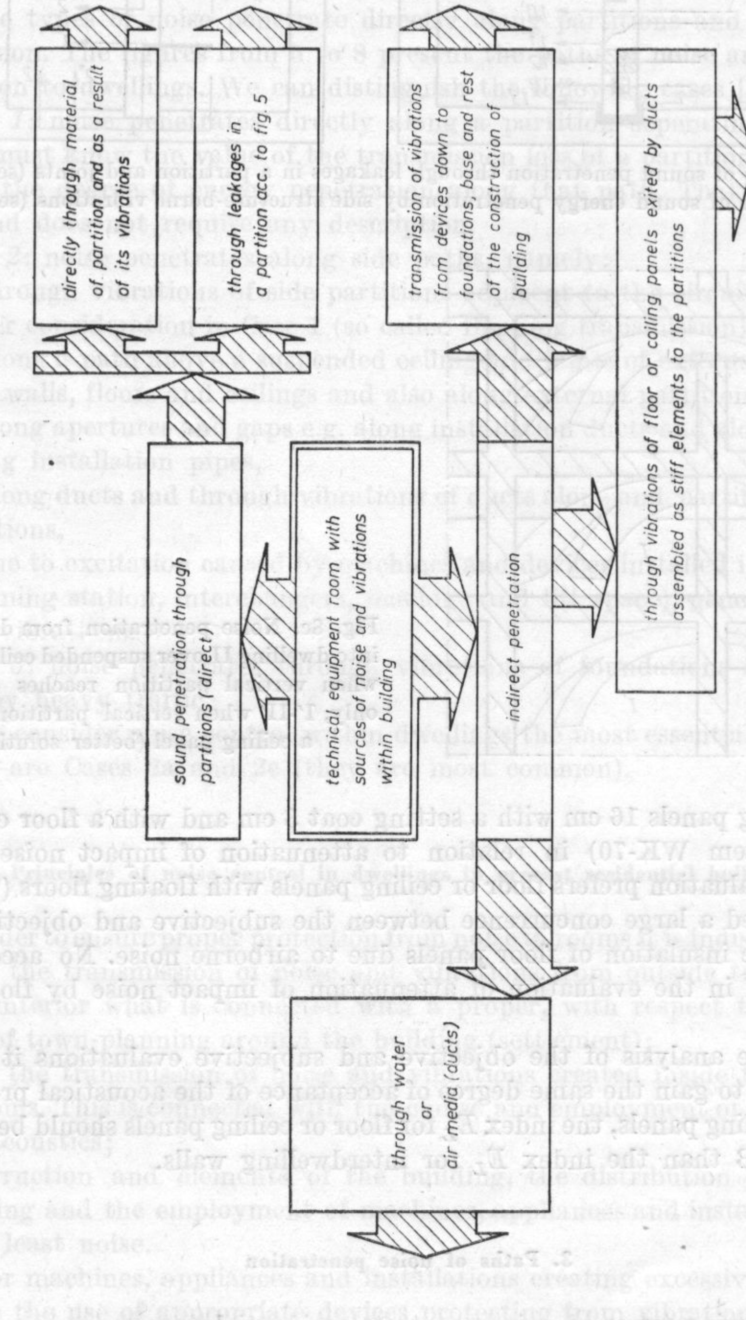
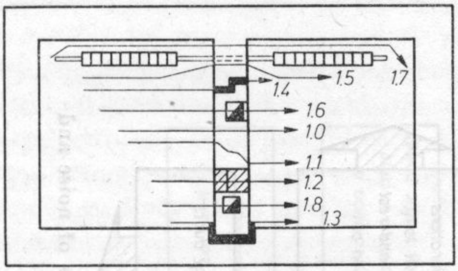


Fig. 7. Scheme of paths of noise transmission/noise from technical equipment rooms with sources of noise and vibrations within building

a



b

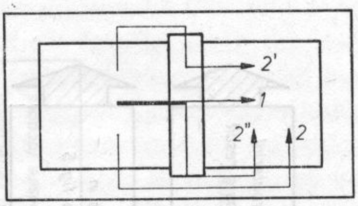


Fig. 8a. Scheme of sound penetration through leakages in a partition and joints (see Fig. 5)
Fig. 8b. Scheme of sound energy penetration by side structure-borne vibrations (see Fig. 5)

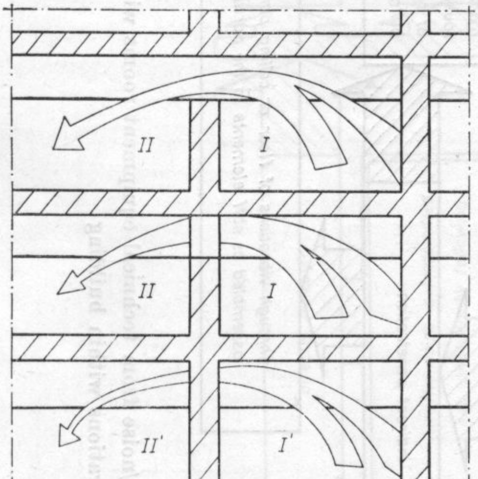


Fig. 8c. Noise penetration from dwelling I into dwelling II over suspended ceilings. I-II when vertical partition reaches a ceiling only, I′-II′ when vertical partition reaches a ceiling panel (better solution)

foor or ceiling panels 16 cm with a setting coat 3 cm and with a floor covering l*Lentex* (System WK-70) in relation to attenuation of impact noise. A the subjective evaluation prefers floor or ceiling panels with floating floors (Tab. 2). There occurred a large concurrence between the subjective and objective evaluation of the insulation of floor panels due to airborne noise. No accordance was observed in the evaluation of attenuation of impact noise by floors and ceilings.

From the analysis of the objective and subjective evaluations it follows that in order to gain the same degree of acceptance of the acoustical properties of floor or ceiling panels, the index E_L for floor or ceiling panels should be greater by 2 to 3 dB than the index E_L for interdwelling walls.

3. Paths of noise penetration

The following types of noise penetrate residential rooms:
outdoor noise (traffic, industry),

from technical utility space within buildings (noise permanently connected with buildings),

from adjacent dwellings (noise from household appliances, radio and TV),

the noise produced by people themselves.

These types of noise penetrate directly along partitions and as flanking transmission. The figures from 5 to 8 present the paths of noise and vibration penetration to dwellings. We can distinguish the following cases [2]:

Case 1: noise penetrates directly along a partition seperating dwellings.

We must know the value of the transmission loss of a partition in order to evaluate the degree of energy penetration along that path. That case is well tested and does not require any description.

Case 2: noise penetrates along side paths, namely:

a) through vibrations of side partitions adjacent to the circuit of a partition under consideration in Case 1 (so called flanking transmission),

b) along a path above a suspended ceiling and joints of external walls with partition walls, floors and ceilings and also along external partitions,

c) along apertures and gaps e.g. along installation ducts and along channels containing installation pipes,

d) along ducts and through vibrations of ducts alone and partitions excited by vibrations,

e) due to excitation caused by machines and devices installed in a building (transforming station, interchangers, machine and lift space), generating construction vibrations.

Case 3: noise penetrates through vibrations of foundations and facades excited by heavy traffic.

If we consider noise control within dwellings the most essential for precast buildings are Cases 2a and 2e (they are most common).

4. Principles of noise control in dwellings in precast residential buildings

In order to ensure proper protection from noise in rooms it is indispensable to:

limit the transmission of noise and vibrations from outside the buildings to their interior what is connected with a proper, with respect to acoustics, solution of town-planning around the building (settlement);

limit the transmission of noise and vibrations created inside the building to the rooms. This is connected with the choise and employment of proper — as regards acoustics;

construction and elements of the building, the distribution of rooms in the building and the employment of machines, appliances and installations making the least noise.

As for machines, appliances and installations creating excessive vibrations and noise the use of appropriate devices protecting from vibrations and noise is essential [3].

4.1. Protection from noise in rooms with the aid of town-planning solutions

Protection from noise in rooms consists in the employment of one or more of the following means:

1. Correct, with respect to acoustics, location within urban areas of noise creating projects and those requiring quietness and, where possible, grouping noise creating projects in especially delimited and acoustically isolated zones in a given city;
2. Neutralization of noise sources, for instance by situating transportation routes in tunnels or excavations, or the use of partitions with required acoustic insulation around the source of noise (an industrial building for instance);
3. Realization of protection from the vibration of roads and tramlines and realization of proper roads with respect to acoustics;
4. Keeping a safe distance between a building and the source of noise;
5. The use of objects screening buildings requiring quietness. These night be trade pavillions, storehouses or special *wall-screens* adequate land shaping (earth embankments), strips of dense and high trees;
6. Use of exterior walls and windows with acoustic insulation suited to the noisiness of the surroundings of a given building.

4.2. The influence of architectural-construction solutions of a building and its elements on the acoustic climate in dwellings

As it results from the tests described in Section 2 the most important role in the shaping of the acoustic climate is played by the construction partitions. The acoustic insulation of a partition against air sounds and impact sounds depends on the following factors:

a) the properties of the material used for the partition and its structure;

b) the system of construction of the building and the employed method of industrialization;

c) the quality of used materials and workmanship.

4.2.1. General dependences concerning the insulation efficiency of partitions against air soands

As it is known the standard acoustic insulation efficiency of individual uniform and pseudo-uniform partitions depends on sound frequency, mass of the partition and the physical properties of the material used for its construction. Thus, it is different for partitions with the same mass but made from materials with different physical properties such as porosity, rigidity and others (Fig. 9). The characteristic curve of standard insulation efficiency as a function of frequency depends on the phenomenon of coincidence which is influenced by the physical properties of the material. For construction practice the fact that the value of the standard acoustic insulation efficiency of vertical and horizontal partitions with the same mass is different (generally greater for horizontal partitions) is very important. This is influenced by the direction of action and distribution of the mass of the partition as well as by different values of lateral transmission in walls and floors. This problem demands detailed justification and future theoretical investigation.

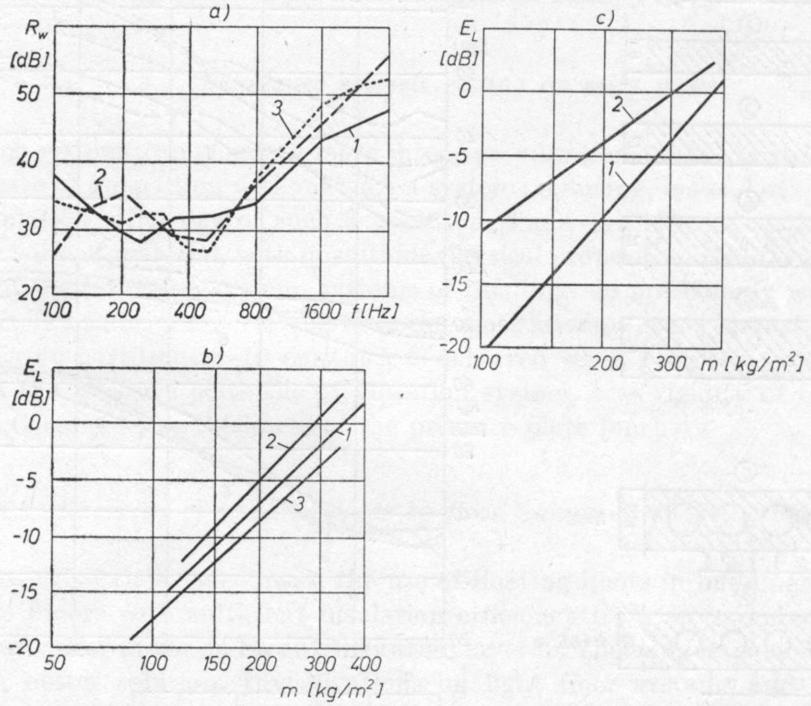


Fig. 9a. Comparison of air sound insulation of different walls (unit masses 125 kg/m but different porosities); *1* — gas concrete wall (24 cm) $R_{w\,\mathrm{mean}} = 36$ dB, *2* — gypsum concrete wall (11 cm), $R_{w\,\mathrm{mean}} = 38$ dB, *3* — reinforced concrete (5 cm) $R_{w\,\mathrm{mean}} = 38$ dB

Fig. 9b. Air sound insulation (empirical mass law) for walls and floors of; *1* — reinforced concrete ($\gamma = 2400$ kG/m^3), *2* — light weight concrete ($\gamma = 1600$ kG/m^3), *3* — gypsum concrete ($\gamma = 1100$ kG/m^3), E — air sound insulation index

Fig. 9c. Air sound insulation (empirical mass law) for *1* — floors, *2* — walls, E_L — air sound insulation index

4.2.2. The acoustic insulation efficiency of channaled slab partitions

A separate case is a partition with cylindrical holes also called a channel partition (the length of the cavity is much greater than its diameter). The standard acoustic insulation efficiency R of a partition with cylindrical holes is, in the band of 100–3200 Hz, much higher than the standard acoustic insulation efficiency of uniform slab partitions with the same mass made from the same kind of material (Fig. 10).

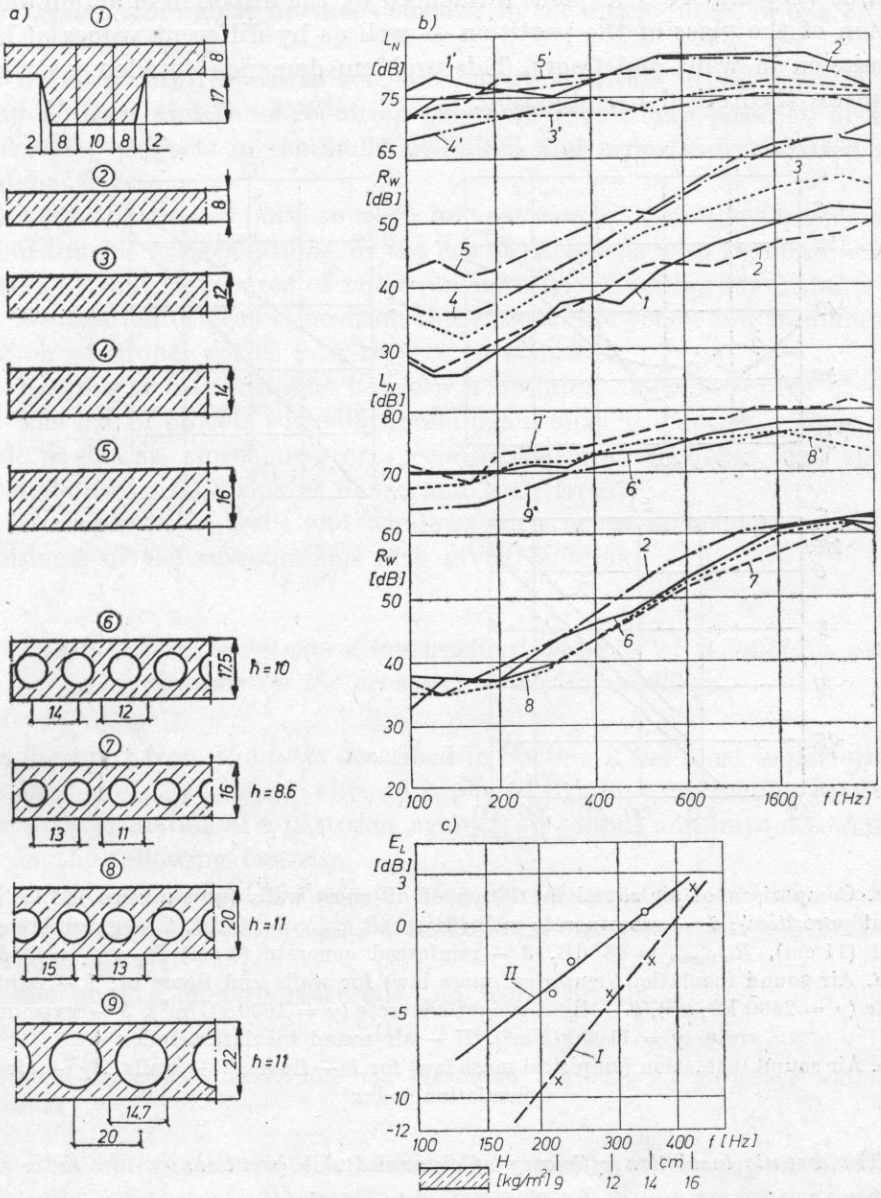


Fig. 10. Airborne and impact sound insulation of concrete floors: a) reinforced concrete slab (1–5) and holow concrete slab (6–9), b) characteristics R and Lur for slab (1–9), air sound insulation (empirical mass law) for I — reinforced concrete plates, II — hollow concrete slabs

Unlike partitions with rectangular air chambers, channel partitions display lesser deterioration of the insulation efficiency due to the phenomenon of coincidence and do not display negative action of the resonant frequencies of cylindrical chambers. This has been confirmed by the results of detailed investigations carried out on two floors from both groups. As can be easily proved the increase in the insulation efficiency of a channel floor in comparison with a slab floor with the same mass results from the greater rigidity of a channel floor.

4.3. Resonance systems resting on solid walls

Such systems can generate more intensive vibrations in the partitions than in the case of a partition without such a system; reducing, instead of increasing, the insulation efficiency of such a partition. Thus, in many cases, because of wrong choise of material, with unsuitable physical properties, used for individual layers of the vibration system, systems of this kind do not comply with requirements. It has been ascertained that the most advantageous increase of ΔR_{wu} for concrete partitions 8–10 cm thick is achieved when $((M_3/s)h_3 \geqslant 28)$, M_3 — mass of the pressure plate the of vibration system, s — rigidity of the elastic layer [kG/cm³], h_3 — thickness of the pressure plate [cm].

4.4. Damping of impact sounds by floor linings on silencing layers

Because of their large mass, the use of floating floors in buildings is inconvenient. Floors with sufficient insulation efficiency from air sounds and light floor systems or lining on an anti-vibration layer for silencing impact sounds are a much better solution. Investigations on light floor systems and lining on silencing layers have shown that their wrong utilization can worsen the insulation efficiency of a floor instead of improving it. It has been ascertained that the better a lining silences impact sounds, the greater drop it causes in the insulation efficiency of a floor from air sounds. This phenomenon has also been confirmed by research centers in other countries. In order to explain the theory of this phenomenon in detail the conduction of further acoustic investigations is necessary.

4.5. Influence of the construction system of a building and method of industrialization on the acoustic insulation efficiency of a partition

The choice of construction of a building and technology of its erection exerts particular influence on the acoustic conditions in rooms. The height of a building creates additional difficulties as regards the use of vertical and horizontal partitions with required acoustic insulation, because of limitations concerning the mass of the partitions and difficulties with ensuring sound-tight joints between partitions.

Also in buildings with skeleton constructions difficulties occur in the ensurance of adequate damping of vibrations transmitted by the skeleton.

Additional installations, such as hydrophores ensuring the flow of water to top storeys, elevators for quick transportation, air conditioning or mechanical ventilation, because of the height of a tall building, create additional difficulties as regards proper insulation of the sources of vibrations from the construction of a high-rise building as well as the ensurance of sufficient silencing of air sounds. Particularly difficult problems arise in the design of:

acoustic insulation of elevator devices – resulting from difficulties in the dilatation of engine and elevator hoistways, because of the height of the building,

insulation from sound and vibrations caused by hydrophores and ventilators mounted on middle storeys,

acoustic insulation of joints between exterior walls and floors of interior walls,

acoustic insulation of chutes, as well as water-sewage pipes, central heating pipes; the acoustic insulation of whole ducts for the passage of these pipes.

The construction of a tall building influences to a great extent the acoustic parameters of partitions. It has been ascertained that the coefficients of sound insulation of concrete partitions with the same thickness and surface display considerable differences in buildings with different construction systems what also testifies to various degrees to lateral transmission. It has also been determined that worse acoustic insulation efficiency of floors is accompanied by worse acoustic insulation efficiency of walls and vice versa. In order to determine the degree of sound transmission along the construction of a building, the acoustic insulation between rooms above each other and diagonally on one or more storeys was investigated in several buildings. In turn different sources of sound were used:

creating air sounds only, such as a loudspeaker,

creating both air and material sounds (e.g. a standard sounder, jack mounted firmly on the construction, piano).

Also, measurements of acceleration of vibrations near partition joints on different storeys were made, making use of the listed sources of vibrations in turn in a building with the same type of floors on all storeys and walls made on all storeys of the same material, but with various thickness (mass). Measurement conditions in all cases were identical. The results of the measurements show that sound energy to a considerable extent is transmitted "side" – ways and the worse the acoustical properties of "side" partitions the greater the transmission. When the source causes material vibrations in the partition, then more intensive transmission of energy to the receiving room takes place, what confirms the thesis concerning the participation of acoustic energy transmission from the sending room to the receiving room by material vibrations.

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THE CONFERENCE ON PROSPECTS IN MODERN ACOUSTICS EDUCATION AND DEVELOPMENT (HOW TO TEACH ACOUSTICS?)

ON THE SPONSORSHIP OF INTERNATIONAL UNION OF PURE AND APPLIED PHYSICS C.7 COMMISSION ON ACOUSTICS

19–21 MAY 1987, GDAŃSK

General information

The aim of the Conference is presentation, discussion and a wide exchange of experience in teaching acoustics in many different world centres which have elaborated programmes and methods in various specializations and in different relations to the educational disciplines in various schools (universities, technical universities etc.) as well as to show what the proper und up to date acoustics ought to be taken into account in the basic, intermediate and advanced courses in physics.

Invited lectures by the world known specialists on the field from different countries will be read and a deal of round table discussions are expected. A limited number of original papers if they are offered by the authors may be presented after accepting them by the Organizing Committee from the point of view of the school profile of the meeting.

The Conference Committee reserves the right to choose contributions and include them into the program. Announcements about acceptation of papers will be sent untill 31-st December 1986. Full text of papers maximum 20 typewritten pages should be sent not later than 28-th February 1987.

There is forseen a discussion during the conference on educational programs of acoustics being realized in different institutions and schools. The organizers would like to distribute the programs during the conference in the form of xerocopies. If you have such programs and you would like to participate in the discussion, please send the materials until 28-th February 1987.

The official languages will be English, Russian and Polish. Translation will take place.

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Secretary

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