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THE CONFERENCE NOISE CONTROL 98

The 11-th International Conference on Protection Against Noise named Noise Control 98 was held in Krynica on June 2–4, 1998. The conference was arranged by the Acoustic Committee of the Polish Academy of Sciences, the Polish Acoustic Society, the Department of Mechanics and Vibroacoustics of the University of Mining and Metallurgy, the Central Institute for Labour Protection and the Building Research Institute.

The Conference was patronized by:

- the Rektor of the University of Mining and Metallurgy,
- the Minister of National Education,
- the Environment Protection, Natural Resources and Forestry,
- the Minister for Labour and Social Policy.

178 participants from Poland and 13 other countries took part in the conference. The following plenary lectures were given:

1. Z. Engel, *Noise Control in Poland in the last half-century.*
2. I. Malecki, *Scientific problems of Polish acoustics fifty years ago.*
3. A. Illenyi, *Polish and Hungarian cooperation in acoustics in the last forty years.*
4. H. Tachibana, *Current situation and future topics of road trafic noise problem in Japan.*
5. S. Marcus, *The role of damping in noise and vibration control technology.*
6. J. Sadowski, *Acoustical plans as an instrument by means of town-planning and traffic organization solution.*
7. Y. Gabillet, J. Roland, *Traffic noise prediction model for built-up or open area.*
8. J. Blauert, *Fundamental of binaural technology for noise assessment.*
9. G. Makarewicz, W. Zawieska, *Computer system for hazards registration and risk assessment.*

Beside the plenary session, 10 other sessions took place in that 62 papers were presented. Poster sessions (17 papers) are a tradition of the Polish conferences on protection against noise. In the plenary session, a few minutes were given to each author of a poster for the introduction in the problem of his paper.

The large interest in the poster session as well as the long discussions in front of the posters evidence the need of those session.

The topics of the sessions were divided into 10 groups:

1. Fundamental and general problems

2. Noise at work
3. Environmental noise
4. Transportation noise
5. Tire/road noise
6. Protection against noise
7. Active methods
8. Vibration
9. Measaurement and analysis
10. Economic problems

"50 years of Noise Control in Poland" indicate the long period of scientific activity and was the motto of the Eleventh International Conference on Noise Control 98.

The year 1948 is considered to be beginning of different kinds of activities aiming at the protection of the work environment and the people's dwellings against noise. The works of Prof. I. Malecki, especially his book *Building acoustics* edited in 1948, initiated 50 years of scientific and technological studies as well as legislative efforts connected with noise and vibration control.

Admittedly, the first measurements of municipal noise were performed already 65 years ago in Warszawa, Kraków and Vilnius. Some limited measures of diminishing noise were undertaken during the 2-nd World War but only on a very small scale.

In 1948, the education of students in electroacoustics was established (Prof. Z. Żyszkowski). Several chairs and departments have been organized, among others: the Laboratory of Electroacoustics at the Technical University in Wrocław (Prof. Z. Żyszkowski, the Chair of Electroacoustics at the Technical University in Warszawa (Prof. I. Malecki) and the Laboratory of Acoustics developed later into the Department of Acoustics and Vibration Theory at the A. Mickiewicz University in Poznań (Professors: M. Kwiek, E. Karaśkiewicz, H. Ryffert).

The Department of Vibration Research organized in the Institute of Fundamental Technological Resaearch (I. Malecki, S. Ziemba, S. Czarnecki, J. Ranachowski), the Central Institute for Labour Protection (C. Puzyna) and the Institute of Building Technology (J. Sadowski) have also investigated the problems of diminishing noise and vibration.

Several other institutes and chairs have contributed to the curtail of noise and vbibration hazards: the Chair of Mechanical Technology at the University of Mining and Metallurgy (Professors: S. Ziemba, W. Bogusz, Z. Engel), the Main Mining Institute (Dr. Malinowski, Prof. A. Lipowczan).

The Polish Acoustic Association and the Committee of Acoustics of the Polish Academy of Sciences were established in the years 1963 and 1964, respectively.

Conferences on Noise and Vibration Control have been organized since 1964, at first on the national level and later on (since 1976) as international NOISE CONTROL conferences.

The first programs of noise control (the Governement Resolution No 169, 1971) as well as first legal acts dealing with that problem were formulated. In 1987, the first reports concerning the noise hazard in Poland were prepared.

Modern and well equipped acoustic laboratories were organized gradually in many scientific institutions and national research programs enabled several basic and applied investigations concerning the lowering of the noise level in Poland.

In my lecture, I am presenting institutions dealing with noise and vibrations: universities, trade institutes, associations and organizations. I am discussing scientific and technological research and achievements, educational programs and publications together with organizational and legal acts necessary for an effective policy of noise and vibration control.

On the basis of several available reports, I have tried to characterize the up to date situation regarding the noise and vibration hazard in Poland.

The main sources of noise pollution are: road traffic, air services and industry. It has been estimated that a substantial part of the country (approx. 21%) is poluted by traffic noise and 33% of the total population is subjected to an equivalent noise level (i.e. to a level higher than 60 dB). Approx. 330 000 industry workers are edangered by excessive noise doses.

The direction and the spectrum of activities which should be undertaken in the environment protection against noise in order to improve the situation have been shown.

By the year 2000, most of the reasons of the environment deterioration should be cleared away and by the year 2010, an improvement in the acoustic climate is expected.

Several monographs and a large number of scientific papers on that subject have been published in Poland.

In this issue of the Archives of Acoustics, there are published a few papers presented at the conference NOISE CONTROL 98.

Zbigniew Engel

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SCIENTIFIC AND ENGINEERING PROBLEMS FACING POLISH ACOUSTICIANS FIFTY YEARS AGO

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1. Introduction

After the II World War's of awful destructions of the whole country, material goods, and its population, but in the same time the common will of the nation to rebuilt the ruins, to develope the home industry and to assure the acceptable conditions for everyday life created the unique historical chance to undertake the huge engineering and technological tasks. The polish acousticians played they modest nevertheless significant role in the solution of the problems related with the realization of these tasks.

Polish acoustics has the tradition of the research activities since the beginning of thirties, mainly in the field of the architectural and physiological acoustics.

During the occupation time the polish acoustics suffered the hard losts, a number of the acousticians not survived or emigrated abroad, however some theoretical research were pursued. The important for the future was the fact, that the lectures related with acoustics and electroacoustics were continued at the electrotechnique and architecture faculties of the underground Warsaw Technical University.

In spite of everything, just after the war several young acousticians supported by the students of the new founded or old universities were able to participate in the national efforts of rebuilding the ruins. First off all they continued to curry on the intensive efforts required of acoustics concerning the definite solutions of the key engineerings problems to be taken into account by engineers, planners, economists and the local and central authorities.

To fulfill this role of the advisers the home scientific research in the field of acoustics had to be initiated and developed, as the essential basis for the adequate treatment of the engineering problems, all the more that the west-european and american experiences e.g. in building acoustics were not fully applicable for the specific local conditions.

Four engineering problems closely related with acoustics were of crucial importance for the rebuilding of the country fifty years ago:

1) town planning,

2) dwelling houses construction,
3) public halls design,
4) noise control in industry.

2. Town planning

The cooperation of acousticians with the region and cities planners and architects was mostly related with the following objects:

1. On the global country scale — the location of newly founded large industrial centers, the foundation of new or drastic changes of the existing cities sizes and importance, the conception of the basic infrastructure principally the planning of the high-ways.

2. Inside the cities — the mutual situations of the noisy objects like the industrial plants, of heavy traffic routes and the places where relative silence is required like the hospitals or universities. The protection of quiet area e.g. parks for citizens to relax.

3. The architectural plan of the living quarters including the fixing of distances between houses and the number of its stores, the localization of streets and places of public interests.

From the point of view of acoustics, town planning is mainly related to the research of the out-doors sound waves propagation. The studies taught soon after the war included the following themes:

1) the attenuation and deflection of sound wave propagation over ground caused by strips of vegetation and screens,

2) the calculation of minimal admissible distance from the sound source as function of the source parameters and the acceptable noise level,

3) the dependence of large distance sound wave propagation from wind direction,

4) the first stage of the systematic measurements of the "acoustic atmosphere" of large cities (Warsaw) and the preparation of the "acoustic maps" of noisy streets.

To formulate the advise concerning town planning it was necessary to dispose the reference point namely, admissible noise level this general question will be discussed at the end of this paper.

The chance to influence the planners decisions depended on the scope of the projects.

It was not at all possible at global level, from the acoustical point of view, because localization of large industrial centers near the great cities like Cracow and Warsaw had entirely political character and the directions of the principal high-ways had defined by the strategic military reasons. The position of acousticians by the detail town planning e.g. as concern the situation of the hospitals, the universities and the industrial plants of local interests was more favourable. In several cases the close cooperation of the architect and the acoustician was attached first of all in Warsaw and as its outcome the acoustic requirements were complied e.g. for the design of the houses stores, number and orientation to the streets.

3. Dwelling houses construction

In consequence of the lack and vital need of flats throuhout the whole country cheap and efficient technologies of house building were necessary.

The elaboration and realization of new technologies was the priority task of the civil engineers and the entire building industry. The prefabricated elements production was generally recognized as significantly advance as compared with the traditional brick construction. Later on the thermal low pressure steam technology of concrete production was typical, but after the war the prefabricated elements were first of all used for ceilings as the light hollow blocks. The brick walls were built as thin as possible. Naturally the sound insulation between flats was very poor and complaints from the inhabitants about bad acoustic conditions were common.

In theory the technical solution was very simple, by the application of flotting ceilings or floors and by the construction of thicker walls. However, such proposals were unrealistic from an economic point of view.

The polish acousticians underthought the research to find a possibly cheap and effective way to increase the insulation of the prefabricated elements to air-born sound and impact noise.

Several original theoretical works was done on the vibrations of the plates with different types of perforation and boundary conditions. The fulfillment of block hollows by the sound absorptive material, the vibration damping between the plates and the new construction of ceiling blocks were experimentally investigated.

The efforts of the acousticians to ameliorate noise insulation in dwelling houses gave in general only restricted results, nevertheless in a number of flats, the additional means for noise abatement were applied partly from private founds.

4. Design and construction of halls

During the II World War the majority of theaters, concert halls and cinemas were destroyed. In Warsaw after the uprising in 1944 all the building of public interest were completely in ruin.

Just a few months after the end of the war Warsaw was again confirmed as the capital and the restoration of the places important for national culture was generally approved as a priority.

The design of large halls like theaters, concert halls and broadcasting studios the advise and cooperation of the acousticians was indispensable. It was not accidental, that the acoustics of halls or more general architectural acoustics had in this time rather favorable conditions for development.

The part of the research in this field was the continuation and the experimental verification of previous theoretical works.

The scientific research were devoted to two main issues

1) the analysis of sound field distribution in an enclosure,

2) the condition for the optimal subjective perception of music and speech.

As usual the three methods of sound field analysis: geometrical, statistical and waves propagation were applied.

In the range of the geometrical method the polish research concerned:

— The interaction of several waves fronts generated by the given spatial distribution of the "virtual" point sound sources.

— The graphical three dimensions sound rays presentation.

— The application of the three dimensional light beams models. This last method appeared very efficient for the design of the reflective surfaces in the ceilings of the large halls.

The research in the field of the statistical method were interesting

— The calculation of the sound field inhomogeneities due to the distribution of the sound absorptive area.

— The correction of the usual reverberation time formula by taking into account the difference of absorption capability due to the local sound field intensity.

The wave method improvement was at this time one of the leading subjects of interests in several research units in Europe and USA and of discussion at the acoustical international meetings.

Polish acousticians participated in these discussions contributing some remarks upon the more precised definitions of notions of clearness, the spatial diffusity, and limit distance.

Another field of research were the properties of sound absorptive materials. The following items were subjects of studies:

— The influence of material porosity degree and structure.

— The calculation of the materials input impedance in the function of the plane wave incidence angle.

— The designs of the bulk absorptive devices having the absorption coefficient larger then one.

For the design of the theaters, concert halls or auditoria the subjective quality of music and speech perception is a decisive factor of appraisal. The polish acousticians were fully conscious of this requirement and the psychoacoustics was one of the field of research.

Already before the war some works related to the subjective feeling of the reverberation time were done. After the war in the time of fast reconstruction of a large number of halls the well known criteria of the optimal reverberation time (e.g. Knudsen) were applied. Nevertheless some own research were undertaken, for instance it was necessary to elaborate the specific method for estimation of the intelligibility of polish speech. In research initiated in the frame of the International Broadcasting Organization (OIR) had as the aim the method for the comparative evaluation of the subjective quality of the broadcasting studia.

General value had also the measurements of the reverberation time and the estimation by the large teams of listeners the subjective acoustic quality of the new built or reconstructed theaters and concert halls. The comparition of these data with the parameters of the halls internationally recognized as the best was interesting from the scientifical point of view. From my own experience and the relations of my friends it is worth to underline the close and friendly cooperation of the architects and acousticians. The reconstruction of the large halls of the great importance for the national culture should be recognized as the joint creative achievement of the architect and acoustician for this reason I would like to mention the names of Prof. M. Kwiek (Warsaw Great Theater) and dr W. Straszewicz (Warsaw National Philharmonie and Great Theater in Łódź) unhappily both deceased.

5. Noise control in industry

Noise control in industry is an important part of the general problem of labor protection. In principle the assurance of saver conditions of work was one of the key watchword of the government. However, the necessity of the continuous industrial noise control were rather disregarded. Only the spread of professional diseases (deafness) caused by the long-period of work in noisy conditions has been taken into account by decision-makers.

Nevertheless acousticians tried to act in the two directions: reduction of noise and vibrations at the source and the decrease of average noise level in industrial plants. The first task was related mainly with some proposals of changes of details of machineries, the technological processes or transportation means. For instance the changes of the types of the transmission gears or other design of the transportation pipes for high pressure gas or chemical liquids were elaborated. Some research works were initiated concerning the vibration of moving parts of machines and the generation and reduction of flow-induced noise.

However, the acousticians had few occasions for direct cooperation with design institutions. The proposals for the means of noise and vibrations were often rejected, because its overcomed the obligatory rigid rules and methods of the construction of the industrial equipment and systems.

The position concerning the average noise level control in large production halls was more favourable. Some proposals concerning the placement of sound absorptive materials inside the halls, the elimination of reflecting area on the hall ceilings and the screening of very noisy machines were accepted by the managements of the several (however not numerous) industrial plants. The research related with the properties of the sound absorptive materials and sound propagation in the enclosures were useful also for these projects.

6. General problems

To attain the ambitions aims of the polish acoustics fifty years ago some preconditions were indispensable.

To execute or even to formulate the requirements related with noise control, the value of admissible noise level should be defined for several most important cases, namely for trafic noise, out-door of buildings noise, noise inside houses and at the working places. The research into this matter was carried on only in the very restrain range, chiefly directed to the support of some complains e.g. related with the loss of hearing or unpleasant leaving conditions. For general purposes the ISO standards and experiences of well known laboratories were admitted as references. However for efficient activity legal confirmation of this requirements was necessary. For this reasons already few years after the war the Polish State Committee for Standards accepted several polish standards for admissible noise levels. Unfortunately the low or by-low on noise abatement at the parlamentary or governmental level was not considered in these years.

Another important condition for the success of acoustics was the capability of acoustic measurements. Here the significant contribution gave the industry connected with the electroacoustic devices production and the universities and research institutes which elaborated the prototypes of the polish sound level-meter.

It is also necessary to stress the role of the international cooperation for the young polish acoustical community. The political situation at this time was of course unfavorable for the official East-West scientific agreements. However, some scientific contacts with the West Europe based partly on the previous personal links were retained. The most important acoustical journals were available, also the participation of polish acousticians in international scientific meetings gradually increased for instance since the second ICA (International Commission on Acoustics) congress in 1953 (Stuttgart) polish acoustics was represented at these congresses and even in the ICA board. To be quite objective it is necessary to underline the significant role of the close cooperation with the acoustical organizations mostly the committees on acoustics of the Academies of Sciences of the countries of the parts communistic blocks.

Finally it is worth stateing that at this time after all the prime duties of the polish acousticians was the teaching of students and the cooperation with engineers, architects, physicians and last but not least with the decision-makers at different levels. We were obliged to accept this priority in spite of pure cognitive scientific research.

In consequence the major part of publications were addressed to the home specialists and practitioners, edited in polish as the books or the papers in the professional journals. Of course, the contents of these publications are now obviously out-of-date. But in spite of all this was the significant attainments of the polish acoustical community and it seems just to indicate as references the books publish in polish a few years after the war, a few later editions which include some valuable information about the past are also indicated.

This paper was a different character then the scientific papers usually published in this journal. However, it seems that the position of the polish acoustics interests as being part of world history of the development of acoustics as scientific discipline and as a modest participant in changes in social and cultural conditions after the war disasters.

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ACOUSTIC MODELLING OF MACHINES

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The vibro-acoustics modelling of machines and equipment sound fields occurring in industry has been presented in the paper. Several methods allowing the sound fiels modelling on the bases of a sound pressure and sound intensity measurements have been proposed. These methods allow to model the sound fields of actual sound sources located in a free field as well as a partially diffuse field. They are also useful in modelling of large industrial installations with linear sound sources. Mathematical dependencies, algorithms of modelling and examples of calculation results are included in the paper.

1. Introduction

Investigations of vibro-acoustic processes occurring in an environment are connected with the acoustic modelling of machines and equipment. It allows the introduction of relevant mathematical models and computer calculations. The vibro-acoustic modelling can concern the following problems:

• modelling of sound sources in machines and equipment — it allows to substitute actual sound sources by specific sources of determined directional characteristics of radiation,

• modelling for vibro-acoustic synthesis,

• modelling for vibro-acoustic research and analyses of vibro-acoustic processes.

All machines, equipment, installations present in an environment form a physical system which, by applying proper simplifications, permits the transition to a substitute acoustic model. Methods of the sound field analysis in a limited zone used nowadays are based on far reaching simplifications. The actual sound sources are substituted by point sources of spherical (or close to them) directional characteristics. That, in many cases, limits the usefulness of the results. The experience of the authors indicate that there is a possibility of modelling the complex sound sources by a set of elementary sources which is able to approximate the emission of the sound power by actual industrial sources in a much better way. One of the methods is the simple utilisation of an elementary source system (consisting of monopole sound sources) for modelling the industrial sound sources. The results of sound pressure or sound intensity measurements around the tested machines can be used for assigning parameters for such models.

2. Substitute acoustic models of sound sources determined by the sound pressure method

2.1. Optimal parameters of the sound source models

When the observation of a sound source radiation is made from a distance sufficiently long, it means from the Fraunhofer's zone, the influence of the geometrical dimensions can be neglected and the sound source can be treated as quasi pointed.

In the case of harmonic waves the spatial distribution of the sound pressure around the actual sound source can be presented by the following expression [7]:

$$p(r,\theta,\varphi) = p_0(r,0,0)\cdot R_0(\theta,\varphi), \quad [\mathrm{Pa}], \tag{1}$$

where $p_0(r,\theta,\varphi) = (A/r)e^{-ikr}$ [Pa]; r, θ, φ – spherical coordinates of the observation point, A – amplitude of the actual sound source [Pa], R_0 – radiation directivity coefficient.

The sound field generated by a set of quasi pointed substitute sources can be presented as an elementary wave superposition emitted by the individual sources.

$$p_z(r,\theta,\varphi) = \sum A_j \frac{e^{ikr_j}}{r_j} R_j(\theta,\varphi), \quad [\mathrm{Pa}], \tag{2}$$

where $A_j(e^{-ikr_j}/r_j)$ [Pa] – complex amplitude of the sound pressure in the direction (0,0), r_j – distance of the j-source from the observation point.

The radiation directivity coefficient is a value dependent on the direction and position of the source against the origin of the coordinates.

The dependency of the radiation directivity coefficient on the source position can be presented as [3]:

$$R^*(\theta,\varphi) = R(\theta,\varphi)\exp\Big[ik\,(x_0\cos\varphi\sin\theta + y_0\sin\varphi\sin\theta + z_0\cos\theta)\Big], \tag{3}$$

where $R^*(\theta,\varphi)$ — radiation directivity coefficient displaced against the origin of coordinates by a vector (x_0, y_0, z_0); $R(\theta\varphi)$ — radiation directivity coefficient located at the origin of coordinates.

Comparing the distribution of sound fields generated by an actual source and a set of substitute sources, the parameters of the set (A_n) can be selected in such a way as to increase their similarity. For that purpose we are introducing a functional:

$$K = \frac{1}{4\pi A^2}\iint\limits_S |p - p_z|^2 dS, \tag{4}$$

where $p = p_0(r,\theta,\varphi)$ — sound pressure generated by the actual source [Pa], $p_z = p_z(r,\theta,\varphi)$ — total sound pressure of the set of sources [Pa], A — actual source amplitude [Pa], S — surface of a sphere with radius r [m^2].

By analogy, the introduced functional K can be compared to the mean square functional. The spherical surface integration can be approximated by summation. As the result, the quality functional would be proportional to the square sum of the sound pressure differences.

The mean square functional, due to its well developed mathematical tools, is often applied for the determination of similarities. The assumed critera are of global nature. This means that for the sound fields of the actual source and the set of substitute sources considered to be identical their consistency in all directions is required.

The assumed criterion has also a physical interpretation. It is a relative sound power of the sound source system consisting of the actual source and the substitute sources vibrating in the reverse phase. By analogy with the active methods, it is a relative sound power of the system with an active sound compensation.

Moreover, the assumed functional is dimensionless. In the case when there is no substitute source, the functional value equals 1, while for an ideal substitute source it equals 0.

For a better understanding, a notion of the functional level is introduced here:

$$L_K = -10\log(K). \tag{5}$$

It will vary from 0, in the case when there is no substitute source, to $+\infty$ in the case of ideal consistency of the actual and substitute sources. Applying the functional assumed in such a way, we are able to determine optimal parameters for the set of the substitute sources. Assuming the notations:

$$A_{jx} = \mathrm{Re}\,(Aj) \qquad \text{and} \qquad A_{jy} = \mathrm{Im}\,(Aj) \tag{6}$$

we can formulate the equations:

$$\frac{\partial K}{\partial A_{jx}} = 0, \qquad \frac{\partial K}{\partial A_{jy}} = 0. \tag{7}$$

Those formulae lead to a linear system of algebraic equations:

$$\begin{aligned} \sum_{i=1}^{n} A_{ix}U_{ji} - \sum_{i=1}^{n} A_{iy}V_{ji} &= U_{j0}\,, \\ \sum_{i=1}^{n} A_{ix}V_{ji} - \sum_{i=1}^{n} A_{iy}U_{ji} &= V_{j0}\,, \end{aligned} \tag{8}$$

where

$$\begin{aligned} U_{ji} &= \frac{1}{4\pi r^2}\iint_S \left(R_i\overline{R}_j + \overline{R}_iR_j\right) dS = U_{ij}\,, \\ V_{ji} &= \frac{1}{4\pi r^2}\iint_S \left(R_i\overline{R}_j - \overline{R}_iR_j\right) dS = -V_{ij}\,. \end{aligned} \tag{9}$$

In some cases the integrals can be solved analytically. When it is assumed that the substitute sources are omnidirectional (monopole), the integrals (9) are equal to [7]:

$$U_{ji} = 2\frac{\sin(kl_{ji})}{kl_{ji}} \qquad \text{and} \qquad V_{ji} = 0, \tag{10}$$

where l — distance between the substitute omnidirectional sources [m], k — wave number.

In this method of modelling, the knowledge of the sound pressure amplitude distribution as well as the phase shift of the sound pressure at separate measuring points is required.

2.2. Optimal distribution and parameters of substitute sources in acoustic models of industrial sound sources

Measurements of the sound pressure distribution around selected industrial sound sources were carried out in the Department of Mechanics and Vibro-acoustics of the University of Mining and Metallurgy in Krak˘w. Apart from the measurements of the sound pressure amplitude, the distribution of the phase angle shift between the sound pressure at the measuring points and a single selected point were also checked. A graphic presentation of the results is given in Fig. 1.

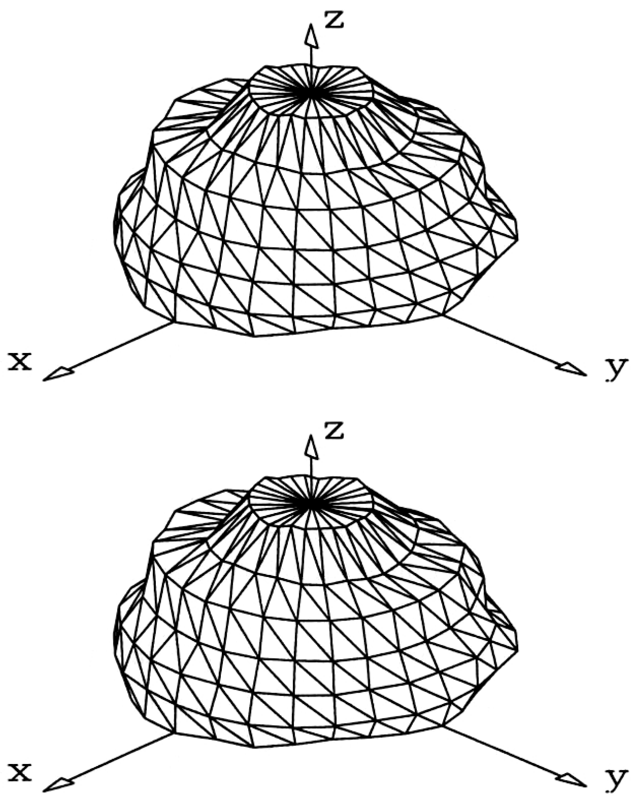

Fig. 1. Amplitude and phase characteristics at frequency = 1 kHz.

The sound field modelling of the actual sound source by an omni-directional sound field over the acoustic screen was performed on the basis of measurements of the sound pressure distribution around industrial sound sources. Formulae (8), (9) and (10) were used in the calculations.

Modelling of the actual sound sources by a greater number of substitute sources can be done in many different ways. One of the procedures and its algorithm is presented in Fig. 2.

At the beginning, the characteristic of the substitute source is assumed to be equal to that one of the actual source. Another substitute source is assumed to be located at the origin of coordinates. The criterion of the similarity gradient and the optimal parameters of the substitute source are estimated. Then the substitute source is displaced in the direction of the maximal gradient. This operation is repeated several times until the local maximum is reached. Later on, we check if the criterion is larger than required. If not, the source characteristic is changed according to the formulae in Fig. 2. A new substitute source located at the origin of coordinates is assumed and all the calculations are performed from the very beginning until the similarity criterion reaches the required value.

As the result of such a procedure, the minimal number of substitute sources in an optimal arrangement and with optimal parameters is found.

An example of the described procedure for a hand drill with the frequency of 100 Hz, is presented in Table 1.

Table 1.

Substitute source number	Location of the substitute source [m]			Sound power level of subst. source [dB]	Phase [deg]	Number of subst. sources	Similarity criterion level [dB]
	x	y	z				
1	0.02	0.11	0.00	67.50	−25.00	1	6.75
2	0.03	0.11	0.26	56.90	134.50	2	8.46
3	0.07	0.26	0.00	51.50	62.10	3	10.21
4	0.10	0.05	0.12	52.70	−134.20	4	10.73
5	−0.06	−0.09	0.27	49.10	−141.90	5	11.20
6	−0.10	0.13	0.12	50.40	−126.10	6	11.57
7	0.00	0.12	0.10	48.00	123.80	7	12.15
8	−0.10	−0.05	0.11	38.20	175.00	8	12.22
9	0.18	−0.22	0.00	38.10	−11.10	9	12.28
10	0.16	−0.03	0.00	37.10	−76.20	10	12.34

As one can see, when the hand drill is substituted by one monopoly, the functional level equals 6.75, by two → 8.46, by three → 10.21 and by ten → 12.34 [dB].

In this method it is not necessary to assume *a priori* the quantity of substitute sources. Their number depends on the distribution of the sound pressure generated by the actual source and the level set for the similarity criterion.

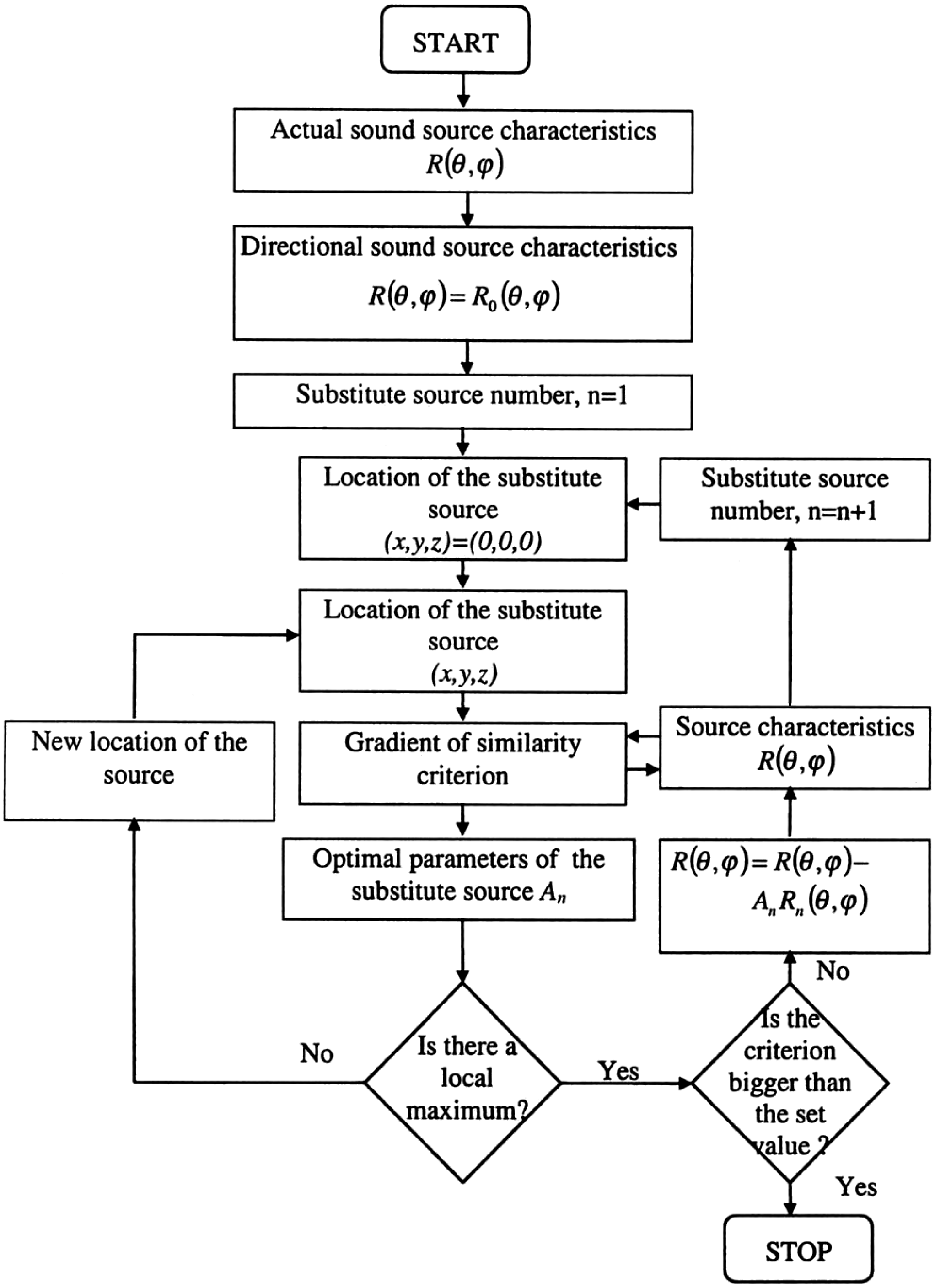

Fig. 2. The algorithm of the sound source modelling

3. Substitute models of sound sources determined by the intensity method

Measurements of the sound intensity around machines performed by a dual-microphone probe are quite often done in the industry. The results of such measurements can be utilised for the development of substitute models.

The velocity potential distribution around omnidirectional sources (monopoles) of harmonic waves can be estimated from the equation:

$$\Phi(x,y,z) = \frac{A}{r}\exp\Big[i(\omega t - kr)\Big]; \qquad [\mathrm{m}^2/\mathrm{s}], \tag{11}$$

where $r = \sqrt{x^2+y^2+z^2}$ — distance from the observation point [m], x, y, z — coordinates of the observation point [m], k — wave number [1/m], ω — angular frequency [1/s], A — source moment [m^3/s].

The sound pressure can be determined from:

$$p = \rho * \frac{\partial \Phi(x,y,z)}{\partial t} \quad [\mathrm{Pa}], \tag{12}$$

where ρ — medium density [$\mathrm{kg/m}^3$].

The sound velocity in the direction of the x-axis equals:

$$v_x = -\frac{\partial \Phi(x,y,z)}{\partial x} \quad [\mathrm{m/s}] \tag{13}$$

while the sound intensity in the direction of the x-axis can be written as:

$$J_x = \frac{1}{2} p \overline{v_x} \quad [\mathrm{W/m}^2]. \tag{14}$$

After simple rearrangements, we are getting the equation for the sound intensity in the x-axis direction as:

$$J_x = \frac{A^2(i+kr)\rho\omega x}{2r^4} \quad [\mathrm{W/m}^2]. \tag{15}$$

Similarly, the equations for the sound intensity in the y-axis and z-axis directions equal:

$$J_y = \frac{A^2(i+kr)\rho\omega y}{2r^4} \qquad \text{and} \qquad J_z = \frac{A^2(i+kr)\rho\omega z}{2r^4}\,, \tag{16}$$

respectively.

The criterion of similarity (quality functional) should be introduced to enable the comparison of the sound intensity of the actual and substitute sources. A mean square criterion is very convenient due to its universal and global nature. In this case, the distribution of the sound density in every direction should be as consistent as possible. The consistence in one direction only is not sufficient. The versatility of the criterion comes from the simplicity of its mathematical description and from the linearity of the equations optimising the parameters of the substitute sources.

When restricted to the real part of the sound intensity only, the criterion of similarity has the form:

$$K = \oint |J_R - J_{zas}| dS \quad [\mathrm{W}], \tag{17}$$

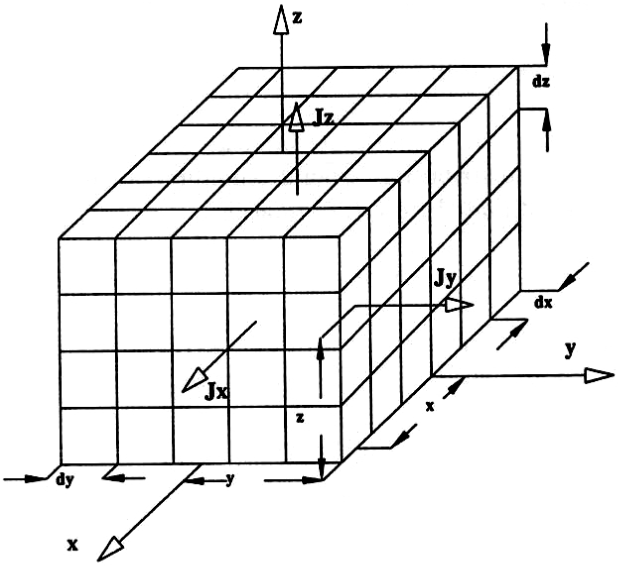

Fig. 3. The measuring grid utilised during the sound intensity measurements around machines.

where J_R — actual sound source intensity [W/m^2], J_{zas} — set of substitute sound sources intensity [W/m^2], S — elementary surface vector [m^2].

In case of discrete measurements result, e.g. those gathered at the grid points on a rectangular prism surface (Fig. 3), the criterion is as follows:

$$K = \frac{1}{N_R} \sum_{j=1}^{m} |J_R - J_{zas}| \Delta S, \tag{18}$$

where N_R — actual source sound power [W] (normative factor), m — mesh grid number, ΔS — mesh grid surface [m^2], J_{R_j}, J_{zas_j} — sound power at the j-point in the direction perpendicular to the grid surface [W/m^2] for the actual and substitute source, respectively.

A following notation of the functional quality level can be introduced:

$$L_K = -10 \log(K). \tag{19}$$

The optimal parameters A_i can be determined on the basis of the system of equations:

$$\frac{\partial K}{\partial A_i} = 0, \tag{20}$$

where A_i — complex moment of the i-source taking into account the phase shift in the sound power generated by different sources.

Applying the above method and utilising the algorithm of the optimal parameter determination, one can determine parameters of the optimal model of the actual sound source radiation with an arbitrary (chosen *a priori*) accuracy.

Table 2 illustrates results of the determination of the compressor substitute source parameters. The intensity method was applied in this example. The distribution of the substitute sources is shown in Fig. 4.

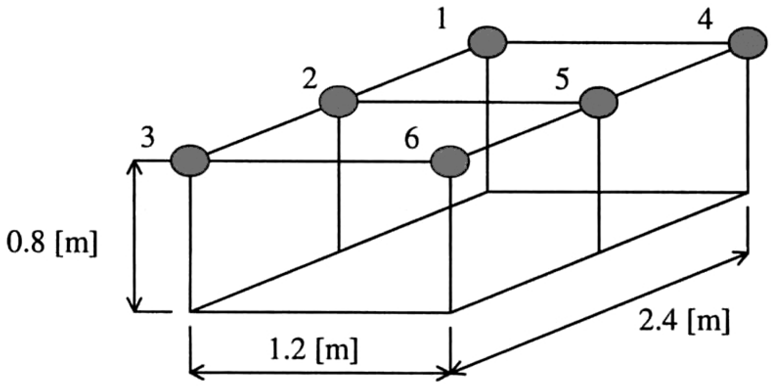

Fig. 4. Example of the substitute sources distribution scheme.

Table 2. Results of modelling of the compressor WSBW-8/220 by 6 substitute sources performed by the intensity method.

Frequency [Hz]	**103**	**130**	**163**	**205**	**259**	**326**	**410**	**516**	**649**	**818**	**1029**	**1296**
Actual source	**84.6**	**80.4**	**92.0**	**90.8**	**94.3**	**80.8**	**79.6**	**75.7**	**72.3**	**71.0**	**67.7**	**71.2**
Source No 1	47.3	48.7	50.5	50.9	52.6	48.3	43.9	43.1	40.1	39.6	38.2	37.5
Source No 2	50.0	51.5	53.1	52.9	54.5	48.8	44.4	43.3	39.6	39.5	37.8	36.7
Source No 3	51.3	54.0	56.0	59.2	61.6	67.0	54.7	67.2	60.6	58.1	55.8	59.2
Source No 4	73.3	71.1	76.5	79.2	82.8	73.0	69.6	60.4	61.3	60.0	56.9	60.6
Source No 5	78.1	77.1	87.7	88.5	89.2	74.9	74.5	70.4	67.0	67.0	62.7	68.9
Source No 6	48.6	50.6	52.7	54.8	57.0	57.3	66.4	65.5	60.3	57.8	54.0	56.2
Total of subst. sources	79.4	78.1	88.0	89.0	90.1	77.5	76.2	73.2	69.4	68.6	64.8	70.0
Similarity criterion, [dB]	**4.00**	**6.41**	**3.45**	**3.95**	**3.22**	**3.95**	**4.57**	**4.74**	**5.22**	**6.90**	**5.70**	**6.07**

4. Substitute models of pipelines

The method of the substitute sources construction presented in the previous section can be useful also in modelling of pipelines, especially in the gas reduction sequences. The reduction sequences (pipeline segments, valves, knees, reducers) were treated as a system of cylindrical sources. For the purpose of the substitute source formation, pipeline segments and linear elements of complicated shapes were substituted by a single cylindrical source of length equal that one of the actual pipe line. Knees and other small fittings (with none distinguished dimensions) were modelled by point sources.

Directional characteristics of a linear source (a system of point sources vibrating in phase and located in infinitesimal distances on the segment of length l) is given by the relationship, [8]:

$$R(\Theta) = \frac{\sin\left(\dfrac{kl}{2}\sin(\Theta)\right)}{\dfrac{kl}{2}\sin(\Theta)}\,, \tag{21}$$

where l — segment length [m], $k = 2\pi/\lambda$ — wave number [1/m], λ — wave length [m].

Then, the sound pressure of an arbitrary point (provided that it is located far from the radiating part, i.e. in the Fraunhofer's zone) can be calculated from:

$$p(x,y,z) = A\frac{R(x,y,z)}{r}\exp(-ikr) \quad [Pa], \tag{22}$$

where r — distance from the observation point to the end of the segment, [m].

If the location of the segment can be described by the coordinates of its beginning and end, (x_1, y_1, z_1) and (x_2, y_2, z_2), the distance from the observation point to the end of the segment will be equal to

$$r = \sqrt{(x-x_s)^2 + (y-y_s)^2 + (z-z_s)^2} \quad [\mathrm{m}]$$

where x_s, y_s, z_s — coordinates of the middle of the segment:

$$x_s = \frac{x_1 + x_2}{2}\,, \qquad y_s = \frac{y_1 + y_2}{2}\,, \qquad z_s = \frac{z_1 + z_2}{2}\,.$$

Then, sine of Θ will then be equal:

$$\sin(\Theta) = \frac{cx\cdot(x-x_s) + cy\cdot(y-y_s) + cz\cdot(z-z_s)}{R}\,, \tag{23}$$

where cx, cy, cz — direction cosines of the straight line on which the segment is located, given by the equations:

$$cx = \frac{x_2 - x_1}{l}\,, \qquad cy = \frac{y_2 - y_1}{l}\,, \qquad cz = \frac{z_2 - z_1}{l}\,.$$

Therefore, the radiation direction characteristics will be equal to

$$R(x,y,z) = \frac{2R\cdot\sin\left(kl\dfrac{cx\cdot(x-x_s) + cy\cdot(y-y_s) + cz\cdot(z-z_s)}{2R}\right)}{kl(cx\cdot(x-x_s) + cy\cdot(y-y_s) + cz\cdot(z-z_s))}\,. \tag{24}$$

The sound pressure generated at an arbitrary point by all pipeline segments and other elements of fittings can be calculated from the equation:

$$p_z(x,y,z) = \sum_{i=1}^{n} A_i\frac{R_i(x,y,z)}{r_i}\exp(-ikr_i). \tag{25}$$

The similarity criterion for the sound fields can be given as:

$$K = \sum_{j=1}^{m}(p - p_z)^2 \quad [\mathrm{Pa}^2]. \tag{26}$$

Minimising the criterion gives the solution by the method of least squares. Therefore, the optimal parameters of sound source A_i can be found from the formula:

$$A = (S^T S)^{-1} S^T P, \tag{27}$$

where $P^T = (p_1, p_2, p_3, ...)$ — complex amplitudes (taking into account a phase shift) of the sound pressure at observation points,

$$S = \begin{bmatrix} s_{11} & s_{12} & s_{13} & \dots \\ s_{21} & s_{22} & s_{23} & \dots \\ s_{31} & s_{32} & s_{33} & \dots \\ \dots & \dots & \dots & \dots \end{bmatrix},$$

where $s_{ij} = \dfrac{R_{ij}(x,y,z)}{r_{ij}} \exp(-ikr_{ij})$, i — substitute source number, j — observation point number. If we don't know the phase shifts of the sound pressure at separate points, we can assume the criterion to be:

$$K = \sum_{j=1}^{m} (p^2 - p_z^2)^2. \tag{28}$$

Using this criterion one does not get optimal parameters in a such easy way. In this case the problem requires the solution of a system of nonlinear equations.

5. Conclusions

The determination of acoustic models of machines by the pressure method requires the determination of the spatial distribution of the sound pressure. The machine under test should be placed in a dead-room and the measurements made in the Fraunhofer's zone (it means sufficiently far from the machine). This limits the applicability of the method to small machines and equipment. On the other hand, the results show smaller deviations and the procedure itself is much easier.

Large industrial installations (e.g. pipelines) are located normally in an open area. In this case it is possible to determine the sound pressure distribution around the installation in the Fraunhofer's zone on the spot. The modelling method connected strictly with the location and sizes of the actual elements seems to be optimal for such cases.

Measurements of the sound intensity around machines performed by the dual-microphone probe are very common in industry. Those results can be utilised in the construction of substitute sound models based on the intensity method. There is no need to displace the machine into a dead-room and the measurements are done close to the source. However, one has to deal with all the deviations occurring when the measurements are carried out by a dual-microphone probe in a partially diffuse field (such as normally in the industry).

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VIBRATIONS OF CIRCULAR PLATE INTERACTING WITH AN IDEAL COMPRESSIBLE FLUID

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In this work, numerical simulations describing the circular plate vibration suppression are presented. It was assumed that the vibrating plate is clamped at the circumference of a planar finite baffle and that it interacts with an ideal homogeneous compressible fluid. The formal solution of the fluid-plate-coupled equation is given for a plate driven by a harmonic surface force with constant density; the state-space realisation of the model is given. Three parameters that characterise fluid loading, internal damping of the plate material and the ratio of the plate radius to the baffle size are included in this model. The modern control theory is then applied to the system state-space equation. An optimal reduction of the plate vibrations was obtained for the point control force located centrally using a linear quadratic regulator (LQR). The simulations of the active attenuation of the plate vibrations were made with a *Simulink/Matlab*® computer program. The results indicate that it is possible to achieve a significant reduction of the vibration amplitude using only one control force.

1. Introduction

The determination of the real vibration source deflection is a very important element of the active control. It is well known that when a structure radiates into air, the radiation field generated by the structure does not contribute significantly to the surface velocity distribution and the interaction effect is negligible. This assumption is often referred to as the "uncoupled" assumption. However, for structures radiating into relatively dense fluids, such as water, the effect of the radiated sound field on the structural response cannot be ignored. In this case the acoustic pressure generated by the structure reacts with the source surface and changes its response. As a result of the fluid coupling, the response of the structure can be significantly different in fluid from those in vacuum or air [1].

Another problem arises from the fact that vibrating plates are usually characterised by low-frequency harmonic vibrations. This means that it can be very difficult to satisfy the condition of an "infinite baffle", which is often applied for the calculation of acoustical quantities because of the constructional reasons. In such situations the lengths of the emitted waves are comparable with the geometric size of the source and the finite baffle dimension has an influence on the system acoustic radiation [6, 11, 15]. To develop successfully an effective solution of the active vibration control solution it is necessary to take into account the phenomena described above in the mathematical model of the considered system.

There are many cases of practical interest to the industry and marine engineering in which the control of the sound and vibrations of plates interacting with a fluid is important. Active vibration control of a flexible structure is a subject that has been vastly researched and described in the recent years. Knyazew and Tartakovskii (1967) were the first who investigated the control of sound radiation using control forces on the vibrating structure [17]. A large number of studies on the active vibration control have been reported. In those studies the classical control, feedforward control, modern control and robust control have been used [17 and references cited inside]. The more recently published research works, dealing with the active control of harmonic sound radiation from planar vibrating structures situated in an infinite baffle, make use of either acoustic or vibration control sources [4, 5, 14]. The amplitudes of the control forces are achieved by applying point forces; the quadratic optimisation is used to calculate the optimal control gains that are necessary to minimise a performance index (cost function) proportional to the radiated acoustic power or to the acoustic pressure. However, reducing structural source vibrations can increase the life of underwater equipment as well as decrease the noise radiated into the surrounding medium. Another approach consists in the suppression of the vibrations of the structure. This work is aimed at this subject. The optimal reduction of plate vibrations is obtained for the point control force located centrally using a linear-quadratic regulator (LQR). The method presented does not assure, however, that the radiated sound will be properly reduced in each case.

To the author's knowledge, the problem of the cancellation of the active vibration of a plate located in a finite baffle and interacting with a fluid has not been treated in the literature, but different aspects of this problem are dealt with in various separate papers. The influence of the interaction fluid with the with the radiation of the circular plate has been considered by several authors [1, 7, 9, 12, 13]. In some works the radiation of sources vibrating in a finite baffle have been investigated. In most of them this problem has been solved by applying the properties of the oblate spheroidal coordinates system [6, 11, 15]. For a circular plate supplied with a finite rigid baffle, the oblate spheroid is also particularly suited for the study of sound radiation. Therefore the basic quantities that characterise the acoustic field were calculated by the author in a similar way [8, 9, 11].

In this paper, the problem of the active vibration control of a circular fluid-loaded plate is analysed. It was assumed that the plate excited harmonically at low frequencies is clamped at the circumference of a limited baffle and that it radiates into a moderately "heavy" fluid. The fluid-plate coupled partial differential equation had been solved previously [8]. The determination of the acoustic pressure is based on the admissible functions for the homogeneous plate in vacuo and on the properties of the oblate spheroidal coordinates. Modern control theory is applied to reduce the considered plate vibrations using a linear-quadratic regulator (LQR) with position and vibration velocity errors feedback signals. In order to design the optimal controller, the equation of motion of a fluid loaded plate driven by a primary external force is expressed in the state-space form. The secondary control force is determined by solving the poles placement problem at the desired locations given by the Ackermann's formula. Graphical representations of the results are presented.

2. The fluid-plate coupled equation

A circular thin plate of radius a and thickness H is surrounded by a lossless liquid medium with static a density ρ_0. It is assumed that the plate clamped in a flat, rigid motionless and finite baffle of radius b is made of a homogeneous isotropic material with density ρ, Poisson's ratio ν, Young's modulus E and has a Kelvin–Voit internal damping.

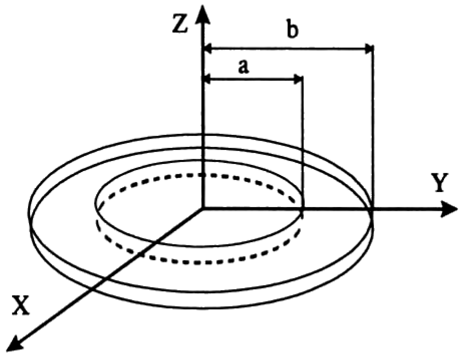

Fig. 1. A circular plate in a rigid baffle of radius b.

Under the time-harmonic external excitation with constant amplitude, which acts on the whole plate surface, the structural dynamics of the plate is reduced to a one-dimensional problem (axially symmetric vibrations) and the structural waves generate a two-dimensional fluid motion in the $x - z$ plane. Taking into account the influence of radiated waves on the plate vibrations as well as internal damping inside the plate material, the plate differential equation of motion can be written as follows [13, 16]:

$$B\nabla^4 w(r,t) + 2\mu \frac{\partial}{\partial t}\left[\nabla^4 w(r,t)\right] + \rho H \frac{\partial^2}{\partial t^2} w(r,t) = f(r,t), \tag{2.1}$$

where $\nabla^4 = \nabla^2\nabla^2$, $\nabla^2 = \left[\frac{1}{r}\frac{\partial}{\partial r}\left(r\frac{\partial}{\partial r}\right)\right]^2$ is the Laplace operator, $B = Eh^3/12(1-\nu^2)$ is the bending stiffness, μ is the coefficient of internal damping, and $w(r,t)$ is the displacement at of points on the plate surface time t which satisfies the following boundary conditions

$$\begin{aligned} w(r,t)|_{r=a} &= 0, \\ \frac{\partial}{\partial r} w(r,t)|_{r=a} &= 0. \end{aligned} \tag{2..2}$$

The eigenvalue problem associated with above differential equation is [18]:

$$\nabla^4 w_m(r) = \lambda_m m w_m(r), \tag{2.3}$$

where λ_m is the m-th eigenvalue, m is the mass per unit area, and $w_m(r)$ is the associated eigenfunction (modal shape function). For the clamped circular plate the eigenfunctions $w_m(r)$ have the form [16]:

$$w_m(r) = u_{0m}\left[J_0\left(\gamma_m \frac{r}{a}\right) - \frac{J_0(\gamma_m)}{I_0(\gamma_m)} I_0\left(\gamma_m \frac{r}{a}\right)\right], \tag{2.4}$$

where $J_0(x)$, $I_0(x)$ designate the cylinder functions, $\gamma_m = k_m a$ is the m-th root of the frequency equation

$$J_0(\gamma_m)I_1(\gamma_m) + J_1(\gamma_m)I_0(\gamma_m) = 0, \qquad m = 1, 2, ..., \tag{2.5}$$

that describes the natural frequency of for m-th mode of the plate

$$\omega_m = \frac{1}{a^2}\gamma_m^2 \left(\frac{B}{\rho H}\right)^{1/2}. \tag{2.6}$$

The eigenfunctions have an orthonormal property if [7, 13]:

$$u_{0m} = 1/(aJ_0(k_m a)), \tag{2.7}$$

and the eigenvalues are related to natural plate frequences by

$$\lambda_m = \omega_m^2\,. \tag{2.8}$$

For the analytical development being undertaken in this paper, the right hand side of Eq. (2.1) can be expressed as follows [12]:

$$f(r,t) = f_w(r,t) + f_s(r,t) + f_p(r,t). \tag{2.9}$$

It is assumed that the plate is excited to vibration by an external time-harmonic force:

$$f_w(r,t) = F_0 e^{-i\omega t} \tag{2.10}$$

and that the control force $f_s(r,t)$, which will minimise the radiated acoustic pressure, is a point force located at the origin:

$$f_s(r,t) = u(t)\delta(r - r_s)|_{r_s=0}\,, \tag{2.11}$$

where r_s is the location of the point force input on the plate surface.

The third component of the right hand side of equation (2.1), $f_p(r,t)$, represents the acoustic fluid-loading acting on the plate as an additional force. The value of this force exerted by the fluid on the plate surface can be calculated as follows [12]:

$$f_p(r,t) = -p(r,z,t)|_{z=0}\,, \tag{2.12}$$

where $p(r, z = 0, t)$ is the acoustic pressure at the point on the surface of the plate.

The acoustic waves propagating through the fluid must satisfy the wave equation

$$\nabla^2 p(r,z,t) = \frac{1}{c^2}\,\frac{\partial^2 p(r,z,t)}{\partial t^2}\,, \tag{2.13}$$

where ∇^2 is the two-dimensional Laplace operator, and c is sound velocity in the fluid. At the fluid-structure interface, the pressure must satisfy the boundary condition [12]

$$\left.\frac{\partial p(r,z,t)}{\partial n}\right|_{r=a} = -\rho_0 \frac{\partial^2}{\partial t^2} w(r,t) = -\rho_0 \ddot{w}(r,t), \tag{2.14}$$

with n denoting the normal to the structure.

If the acoustic pressure radiates from the plate vibrating harmonically, the wave equation reduces to the Helmholtz equation

$$\left(\nabla^2 + k_0\right) p(r,z) = 0, \tag{2.15}$$

where $p(r,z)$ is the pressure amplitude and $k_0 = \omega/c$ is the acoustic wave number at frequency ω.

3. Solution of the Helmholtz equation

The acoustic radiation from surfaces, that are separable for the Helmholtz wave equation, may be calculated by the eigenfunction expansions. For the circular plate located in a finite baffle, the problem of determining a distribution of the acoustic pressure has been obtain by using the method of separation of variables in the oblate spheroidal coordinate system (OSCS) [8]. The OSC system is particularly suitable for the study of radiation of circular sources because one limit of these shapes is approached.

Due to the symmetry of the radiated waves with respect to the z axis for $p(r,z) = p(\eta,\xi)$, where η, ξ are the spheroidal coordinates (Fig. 2), the following equation for the outgoing waves has been obtained [3, 8]:

$$p(\eta,\xi) = -i\omega\rho_0 \sum_{l}^{\infty} A_l S_{0l}^{(1)}(-ih,\eta) R_{0l}^{(3)}(-ih,i\xi), \tag{3.1}$$

where $S_{0l}^{(1)}(-ih,\eta)$ denotes the angular spheroidal function of the first kind, $R_{0l}^{(3)}(-ih,i\xi)$ is the radial spheroidal function of the third kind, A_l are the expansion coefficients and $h = k_0 b$.

The coefficients can be derived from the boundary condition (2.14), which in oblate spheroidal system has the form:

$$\frac{\partial p}{\partial n} = \frac{1}{h_\xi} \left.\frac{\partial p}{\partial \xi}\right|_{\xi=\xi_0} = \begin{cases} -\rho_0 \ddot{w}(\eta,0,t) & \eta_0 \le \eta \le 1, \\ 0 & \eta_0 \ge \eta \ge -\eta_0\,, \\ \rho_0 \ddot{w}(\eta,0,t), & -1 \le \eta \le -\eta_0\,. \end{cases} \tag{3.2}$$

Applying the orthogonal property of the angular spheroidal functions [3]

$$\int_{-1}^{1} S_{mn}(-ih,\eta) S_{mn'}(-ih,\eta)\, d\eta = \delta_{mn'} N_{nm}\,, \tag{3.3}$$

it is possible to determine an expression for the eigenfunction expansion coefficients

$$A_l = \frac{ibW_l(-ih,t)}{\omega \dfrac{\partial R_{0l}^{(3)}(-ih,0)}{\partial \xi} N_{0l}}\,, \tag{3.4}$$

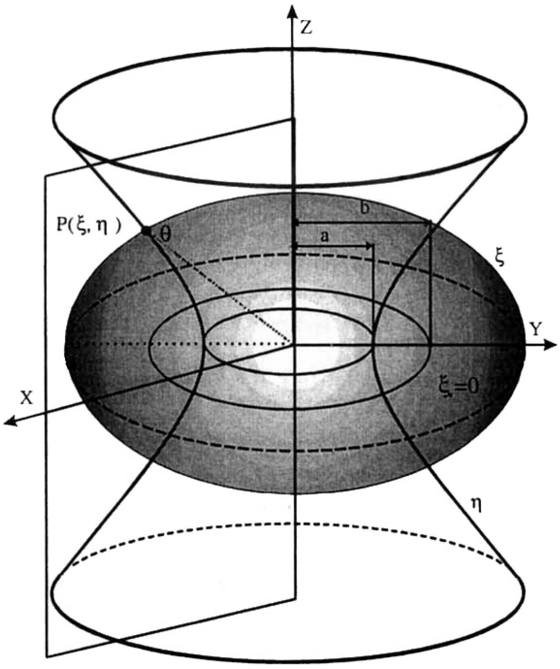

Fig. 2. Geometry of the system.

where N_{0l} denotes the norm factor and

$$W_l(-ih,t) = \int_{-1}^{1} \ddot{w}(\eta,t) S_{0l}(-ih,\eta)\eta \, d\eta \tag{3.5}$$

is the characteristic function in the oblate spheroidal coordinate system. Finally, we obtain the expression for the acoustic pressure written as:

$$p(\eta,\xi) = -b\rho_0 \sum_{l}^{\infty} W_l S_{0l}^{(1)}(-ih,\eta) \frac{R_{0l}^{(3)}(-ih,i\xi)}{\dfrac{\partial R_{0l}^{(3)}(-ih,i0)}{\partial \xi} N_{0l}(-ih)} . \tag{3.6}$$

4. Relation between the sound pressure and the plate vibration

Let us assume that the plate displacement can be expressed in the form of series

$$w(r,t) = \sum_{m}^{\infty} s_m(t) w_m(r), \tag{4.1}$$

where $w_m(r)$ are eigenfunctions described by (2.4), and $s(t)$ is the modal amplitude in time t. Making a double differentiation with respect to the variable t, we get:

$$\ddot{w}(r,t) = \sum_{m}^{\infty} \ddot{s}_m(t) w_m(r). \tag{4.2}$$

The series (4.2) is now expressed in the oblate spheroidal coordinate system (OSCS) using the following transformation [3]

$$r = b\left[(1-\eta^2)(\xi_0^2+1)\right]^{1/2}. \tag{4.3}$$

Using the properties of the OSCS and assuming $\xi_0 = 0$, $r = b(1-\eta^2)$, the expressions obtained become appropriate for the plate in the finite baffle:

$$\ddot{w}(\eta,t) = \sum_m^{\infty} \ddot{s}_m(t) w_m(\eta). \tag{4.4}$$

The eigenfunctions take the form [8]:

$$w_n(\eta) = u_{0n}\left[J_0\left(\gamma_n \frac{b}{a}\sqrt{1-\eta^2}\right) - \frac{J_0(\gamma_n)}{I_0(\gamma_n)} I_0\left(\gamma_n \frac{b}{a}\sqrt{1-\eta^2}\right)\right] \tag{4.5}$$

and they remain orthonormal if

$$u_{0n} = b/(aJ_0(k_n a)). \tag{4.6}$$

Considering relation (4.4), the characteristic function (3.5) can be formulated as follows:

$$W_l(-ih,t) = \sum_m \ddot{s}_m(t) \int_{-1}^{1} w_m(\eta) S_{0l}(-ih,\eta)\eta\, d\eta. \tag{4.7}$$

In this way, the acoustic pressure acting on the surface of the considered plate can be described in terms of a spatial and time-dependent part as follows:

$$p(\eta,\xi=0,t) = -i\frac{\rho_0}{k_0}\sum_m^{\infty} \ddot{s}_m(t) \sum_{l=0}^{\infty} W_{ml} S_{0l}(-ih,\eta)\chi_{0l}(-ih,\xi=0), \tag{4.8}$$

where the transfer impedance χ_{0l} was introduced

$$\chi_{0l}(-ih,\xi) = (-ih)\frac{R_{0l}^{(3)}(-ih,i\xi)}{N_{0l}(-ih)\dfrac{\partial}{\partial\xi}R_{0l}^{(3)}(-ih,i0)} \tag{4.9}$$

and

$$W_{ml} = \int_{-1}^{1} w_m(\eta) S_{0l}(-ih,\eta)\eta\, d\eta. \tag{4.10}$$

Using asymptotic properties of the spheroidal functions [3]:

$$R_{0l}^{(3)}(-ih,i\xi) \xrightarrow{\xi\to\infty} (-i)^{l+1}\frac{e^{ih\xi_\infty}}{h\xi_\infty}, \tag{4.11}$$

the far-field acoustic pressure can be calculated from the following formula:

$$p(\eta,\xi_\infty,t) = -\frac{b\rho_0 e^{ih\xi_\infty}}{h\xi_\infty}\sum_m^{\infty} \ddot{s}_m(t) \sum_{l=0}^{\infty}(-i)^{l+1}\frac{W_{ml}S_{0l}(-ih,\eta)}{R_{0l}^{(3)\prime}(ih,0)N_{0l}(-ih)}. \tag{4.12}$$

5. System discretization

Having defined several properties of the system of interest to us, it is straightforward to re-express the equation (2.1) as a set of modal equations. As mentioned earlier, the typical approximation of such a partial differential equation can be obtained from the relationship

$$w(r,t) = \sum_{m}^{N} s_m(t) w_m(r), \tag{5.1}$$

where $w_m(r)$ represent the known eigenfunctions described by (2.4). In theory, $N = \infty$. However, in practice N is considered to be a finite number suitably large for the accurately modelling of the system dynamics. In a similar way let us expand the right hand side of the plate equation of motion (2.1) into series:

$$f_w(r,t) = \sum_{m}^{N} r_m(t) w_m(r), \qquad (5..2)$$

$$f_s(r,t) = \sum_{m}^{N} u_m(t) w_m(r), \qquad (5..3)$$

$$f_p(r,t) = \sum_{m}^{N} z_m(t) w_m(r). \qquad (5..4)$$

Inserting the above expansions into equation (2.1), multiplying both sides by the orthonormal eigenfunction $w_n(r)$, and integrating over the surface of the structure S, the governing equation of motion can be re-expressed as:

$$\sum_{m=1}^{N} \left[\ddot{s}_m(t) + 2\mu\omega_m^2 \dot{s}_m(t) + \omega_m^2 s_m(t) = r_m(t) + u_m(t) + z_m(t) \right], \tag{5.5}$$

where

$$\left.\begin{array}{l} r_m(t) \\ u_m(t) \\ z_m(t) \end{array}\right\} = \int_S f_j(r,t) w_m(r)\, ds, \qquad j = w,s,p, \quad m = 1,2,...,N \tag{5.6}$$

mean the modal generalised forces.

6. Derivation of modal generalised forces

To derive the modal generalised forces $r_m(t)$, $u_m(t)$, $z_m(t)$, it is necessary to integrate the analytical expressions according to the formula (5.6). In the case of a driving force (2.10), from the integration results:

$$r_m(t) = k_{wm} r(t), \tag{6.1}$$

where

$$k_{wm} = \frac{aF_0 J_1(\gamma_m)}{\gamma_m J_0(\gamma_m)\rho H}\,. \tag{6.2}$$

For the point control force $f_s(r,t)$ described by (2.11), the modal generalised force is simply equal to the value of the eigenfunction at the control force application point:

$$u_m(t) = k_{sm}u(t), \tag{6..3}$$

$$k_{sm}(t) = \frac{1}{aJ_0(\gamma_m)\rho H}\left(1 - \frac{J_0(\gamma_m)}{I_0(\gamma_m)}\right). \tag{6..4}$$

The third component of the right hand side of Eq. (2.1) can be calculated according to (2.12) and (4.8). As the result we obtain

$$z_m(t) = \frac{b}{a}\varepsilon_1 \frac{1}{J_0(\gamma_m)} \sum_{n=1}^{N} \ddot{s}_n(t)c_{mn}\,, \tag{6.5}$$

where $\varepsilon_1 = \rho_0/\rho H k_0$ represents the fluid-loading parameter [1, 7] and

$$c_{mn} = \sum_{l=0}^{\infty} W_{ml}\chi_{0l}(-ih)W_{nl}^T\,. \tag{6.6}$$

7. Transformation into the state-space form

The state space modelling is based on the fact that a continuous, linear system can be characterised by a set of first order differential equations. State variables are those, which comprise the smallest set of variables, which are needed to describe completely the behaviour of the dynamics of the system of interest. Grouped together, the state variables form a state vector.

As the result of the previous calculations, the following equation describing the behaviour of the m-th mode of the considered system is obtained:

$$\ddot{s}_m(t) + 2\mu\omega_m^2\dot{s}_m(t) + \omega_m^2 s_m(t) = k_{wm}r(t) + k_{sm}u(t) + \sum_{n}^{N}\frac{d^2}{dt^2}s_n(t)c_{mn}\,. \tag{7.1}$$

The matrix notation simplifies significantly the mathematical representation of the system, and provides a form of the problem expression, which is readily amenable to a computer solution. So, we write Eq. (7.1) in the matrix form

$$(\mathbf{I}+\mathbf{C})\ddot{\mathbf{s}}(t) + 2\mu\boldsymbol{\Omega}^2\dot{\mathbf{s}}(t) + \boldsymbol{\Omega}^2\mathbf{s}(t) = \mathbf{K}_{\mathbf{u}}(t) + \mathbf{K}_w\mathbf{r}(t). \tag{7.2}$$

In the above expression $\mathbf{I}$ denotes the identity matrix, $\mathbf{K}_s$ and $\mathbf{K}_w$ are coefficient vectors calculated for each mode with the expression (6.2) and (6.4), respectively, $\mathbf{C}$ represents the fluid-plate interaction matrix obtained with (6.6), $\boldsymbol{\Omega} = \mathrm{diag}[\omega_1, \omega_2, ..., \omega_N]$.

The modal model presented above can be expressed now in the state space format. To do so, let us define the state vector

$$\mathbf{x}(t) = \begin{bmatrix} \mathbf{s}(t) \\ \dot{\mathbf{s}}(t) \end{bmatrix}. \tag{7.3}$$

Equation (7.2) can be expressed as:

$$\dot{\mathbf{x}}(t) = \mathbf{A}\,\mathbf{x}(t) + \mathbf{B}\,\mathbf{u}(t) + \mathbf{V}\,\mathbf{r}(t), \tag{7.4}$$

where the dot denotes differentiation with respect to time, $\mathbf{x}$ is the $(n \times 1)$ state vector, $\mathbf{u}$ is the $(m \times 1)$ control vector, and $\mathbf{A}$ is the $(n \times n)$ state matrix, $\mathbf{B}$ is the $(n \times m)$ control input matrix and $\mathbf{V}$ is the $(n \times 1)$ disturbance matrix.

$$\begin{aligned} \mathbf{A} &= \begin{bmatrix} \mathbf{0} & \mathbf{1} \\ -(\mathbf{I}+\mathbf{C})^{-1}\mathbf{\Omega}^2 & -2\mu(\mathbf{I}+\mathbf{C})^{-1}\mathbf{\Omega}^2 \end{bmatrix}, \\ \mathbf{B} &= \begin{bmatrix} \mathbf{0} \\ (\mathbf{I}+\mathbf{C})^{-1}\mathbf{K}_s \end{bmatrix}, \qquad \mathbf{V} = \begin{bmatrix} \mathbf{0} \\ (\mathbf{I}+\mathbf{C})^{-1}\mathbf{K}_w \end{bmatrix}, \end{aligned} \tag{7..5}$$

The above state-space model of the considered system will be used in the process of designing the optimal feedback control so as to suppress the plate vibrations.

8. Computer simulation of the feedback control

The optimal linear system theory can be used now to derive the response of the considered system including the feedback control. We want to modify the dynamic response of the system by introducing a control input $\mathbf{u}(t)$ derived from the state feedback as follows [18]:

$$\mathbf{u}(t) = -\mathbf{K}\,\mathbf{x}(t), \tag{8.1}$$

where the problem is to determination of the gain matrix $\mathbf{K}$ which facilitates our requirements. The performance index has been chosen as:

$$J = \int_0^\infty \left(w^2 + \alpha \dot{w}^2 + \beta \frac{u^2}{u_{\max}^2} \right) dt, \tag{8.2}$$

where α and β are the weight coefficients.

There are different methods, which can be used to obtain a control gain matrix minimising the optimal control performance index. In this paper, the optimal LQR controller is determined by solving the problem of poles placement at desired locations given by the Ackermann's formula [18]. This algorithm is effective for a single input system when the rank of state matrix $\mathbf{A}$ is less or equals 10. The result of the Ackermann's formula is the automatic calculation of the matrix $\mathbf{K}$ required to place the closed-loop poles Λ_C in the desired locations:

$$\Lambda_C(s) = |s\mathbf{I} - \mathbf{A}_C| = |s\mathbf{I} - (\mathbf{A} - \mathbf{B}\,\mathbf{K})|, \tag{8.3}$$

where $s = i\omega$, $\mathbf{I}$ is the identity matrix, $\mathbf{A}_C$ is the close-loop system matrix.

Table 1 contains the pole values for the open and closed-loop systems calculated with the following values of the physical parameters of the plate:

$$\rho = 2700\,\mathrm{kg/m^3}, \qquad \nu = 0.33, \qquad \mathrm{E} = 7.1 \cdot 10^{10}\,\mathrm{N/m^2}, \qquad \mu = 0.00011.$$

In the simulations, the model including the four first modes of the aluminium plate of a radius 0.2 m and a thickness of 1 mm was applied. In order to determine the dynamics of

Table 1.

Open-loop poles of the system [rd/sec]	Desired poles of the closd-loop system [rd/sec]
$1.0e+003*$	
$-4.2778+4.4528i$	-300
$-4.2778-4.4528i$	-300
$-1.3516+3.1968i$	-45
$-1.3516-3.1968i$	-45
$-0.2722+1.5337i$	-5
$-0.2722-1.5337i$	-5
$-0.0180+0.4000i$	-15
$-0.0180-0.4000i$	-15

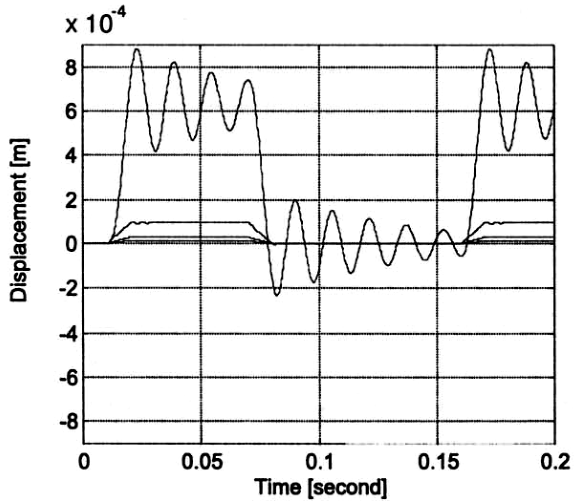

Fig. 3. The open-loop system response (four first modes) to a rectangular periodic signal with constant amplitude.

the fluid-plate system, the obtained model was first subjected to a rectangular periodic signal with constant amplitude and frequency. Figure 3 presents the behaviour of the system without control feedback.

Figure 5 presents the plate response for the gain matrix **K** obtained. It can be seen that the control input (Fig. 4) raises during 0.01 sec until the value of 0.35, which equals 35% of maximum control signal. In this time the plate vibrations are damped sufficiently — the first mode amplitude does not exceed $1.5 \cdot 10^{-4}$ m. The higher modes are damped almost completely. Because of the very low control signal within the time interval of $0.08-0.18$ second, very small plate oscillations remain until next cycle. Comparing the plate vibration amplitude for the open-loop system, one can see that for the LQR controller the output signal was damped more than seven times.

The system response to a uniform periodic excitation over the plate surface, which will be realised in practice on the constructed experimental plant, has been examined as the

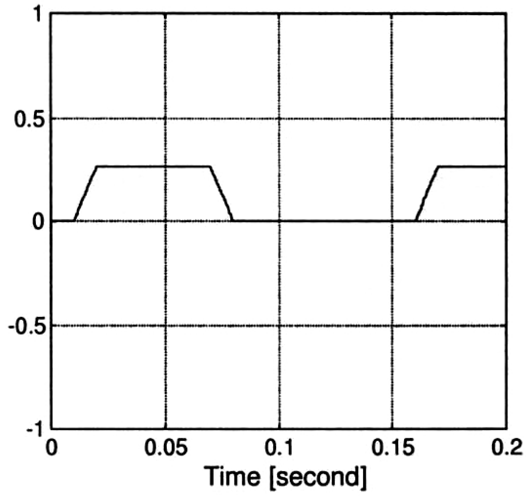

Fig. 4. Control input.

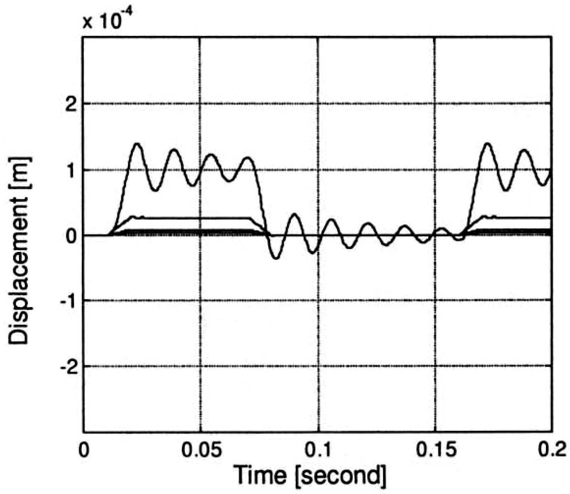

Fig. 5. Plate response.

second example. The chosen excitation of 80 Hz is close to the natural plate frequencies of the first plate mode of 63 Hz.

It can be seen in Fig. 6 that the system response is transient, however, it becomes sinusoidal when the system is allowed to run further out in time. This remaining response should be cancelled; the LQR controller designed could do it effectively as observed in Fig. 7.

The plate displacement response in Fig. 7 illustrates that for the sum of the four observed plate modes, the transient responses decay over time but due to the coupling mechanism, this modes persist as sinusoidal with a dominant frequency. The apparent

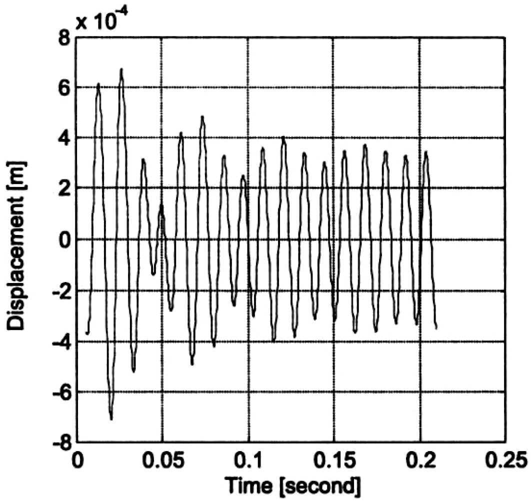

Fig. 6. The open-loop system response (the sum of four modes) to a uniform periodic excitation.

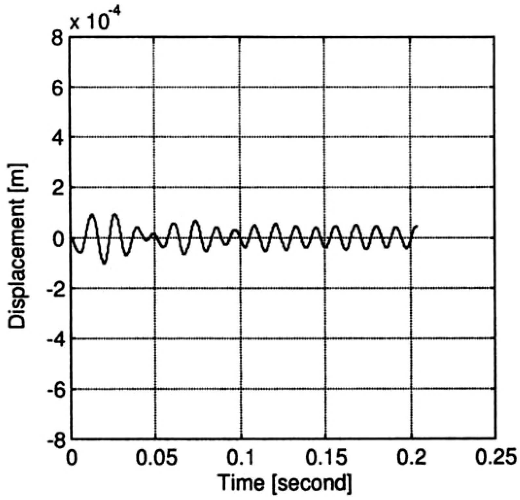

Fig. 7. The close-loop system response to a uniform periodic excitation.

beating effect is related to the difference in frequencies between the system resonant response and the forced response. The output signal is damped approximately seven times.

9. Conclusions

In this work the analysis of the fluid-plate system dynamics excited harmonically at low frequencies has been presented. The mathematical model of the considered system include two major difficulties: the plate vibrates in a finite baffle and the acoustic wave radiated by the plate interacts on its surface due to the coupling mechanism. In addition the Kelvin-Voigt damping in the plate material has been taken into account.

For the system under consideration, the state-space model has been constructed. This model was used during numerical simulations of the active attenuation of plate vibrations realised in Simulink/Matlab. The optimal control theory was applied to the state equation and optimal reductions of plate vibrations were obtained for point control force located centrally using the LQR controller.

Two examples demonstrating the response of the system to two different external forces were presented. In the first case the LQR controller performance was verified by testing the behaviour of a plate driven by the rectangular periodic signal with constant amplitude and frequency. Comparing the plate vibration amplitude for the open and close-loop coupled systems, it was found that the output signal was damped more than seven times.

In a second example, a periodic forcing function with a frequency close to the plate resonance was chosen. The plate displacement response illustrates that for the sum of four observed plate modes the transient responses decay over time, but due to the coupling mechanism these modes persist as sinusoidal with a dominant frequency. These remaining mode responses should be cancelled; the designed LQR controller can do it effectively as shown in Fig. 7.

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COMPUTER SIMULATION OF THE INDICES OF THE ACOUSTIC ASSESSMENT OF MACHINES

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Two indices of of the acoustic assessment of machines are proposed: a power index and an emission index which enable the potential user to carry out an acoustic assessment of the machine to be installed in the operation room. The worked out indices are functions of several parameters such as e.g. variations of the operational conditions of the machine, or the acoustic properties of the room. The results of the simulation tests, illustrating the effects of the variation of different parameters on the values of the indices of the acoustic assessment, are given.

1. Introduction

The efficiency of an acoustic assessment determines whether a machine, which can be dangerous for man because of excessive noise emission is approved for use or not. The methods that heve been used for the acoustic assessment until now do not take into consideration, among other things,

• the real operation variants of the machine and their duration,

• the parameters of the operation room and their influence on the noise level at a work station.

Furthermore, the sound power level limit values have not been established for most of the machines. In this connection, the principles of the acoustic assessment of machines have not been fully useful in most cases for the machine users. This results, among other things, from the fact that, although the service manual contains data on the sound pressure level at the work station and the sound power level values, it does not provide full information on the machine safety under the operational conditions. Therefore, it is proposed to introduce two alternative indices of the acoustic assessment of machines [1, 3, 6]: a power index and an emission index which enable the potential user of a machine to carry out an acoustic assessment of the machine to be installed in the operation room. The power and the emission indices are functions of the parameters characterizing both the machine and the operation room and influencing the value of the sound pressure level at the work station. In connection with these indices, two new quantities to characterize the noise emitted by a machine are also proposed: the real global A-weighted sound power level and the real global A-weighted sound pressure level caused by the machine at the work station [3, 6].

2. Power index

For a given machine to be installed under specific operational conditions, a power index of the acoustic assessment W_E, in dB, is given as a folowing function:

$$W_E = f_1(C_A,\, L_W,\, x,\, y,\, z,\, d,\, D,\, L_L), \tag{1}$$

where C_A is the primary acoustic climate in the operation room, L_W is the sound power level of the machine, in dB, x, y, z are the coordinates of the machine location, d is the distance between the machine and the work station, in m, D are the acoustic properties of the operation room, L_L is the admissible value of the equivalent A-weighted sound pressure level at the work station, in dB.

The primary acoustic climate C_A at the point in that the work station of the machine being assessed will be located is composed of sound waves coming from all the original noise sources in the room. This climate is a function of the partial climates produced by the individual noise sources existing up to now and may be expressed by the equivalent A-weighted sound pressure level L'_{Aeq} [4].

The parameter which characterizes the noise of the machine is the sound power level L_W, which may be written as follows:

$$L_W = f_2(n_1,\, n_2,\, n_3,\, \Delta t), \tag{2}$$

where n_1 are non-acoustic parameters which characterize the machine and influence the noise emitted (e.g. rotational speed, motive power), n_2 are parameters which characterize the way of mounting the machine, n_3 are parameters which characterize the operating material, Δt is the time interval in which the sound power level is determined.

By making a synthesis of the parameters n_1, n_2 and n_3 in the specified time intervals Δt_i, it is possible to determine k possible technological-kinematic and structural variants of the operation of the machine during its operation. Each of these variants is characterized by the partial A-weighted sound power level L_{WAi}. Thus, in order to characterize the machine radiation by an energy quantity which is the sound power level, its definition should be extended by the notion of the real global A-weighted sound power level L_{WRGA}; the latter is described by the formula:

$$L_{WRGA} = 10 \lg \frac{1}{\sum\limits_{i=1}^{k} \Delta t_i} \left(\sum_{i=1}^{k} \Delta t_i 10^{0.1 L_{WAi}} \right), \tag{3}$$

where Δt_i is the time duration of the i-th variant of the operation of the source, in s.

In accordance with the relation (1), the machine location in the operation room is determined by the coordinates x, y, z. There are five basic ways of the machine location: suspended in the middle of the room, suspended on the middle of the wall, placed in the middle of the floor, placed on the edge of the surfaces (e.g. of the wall and the floor) or placed in the corner of the room. For each location the characteristic feature is the shape of the radiation surface expressed by the radiation index Q. The values of the radiation index Q for the basic machine locations are given in Table 1 [5].

Table 1. Values of the radiation index Q for basic machine locations.

Machine location	Radiation surface	Radiation index Q
Suspended in the middle of the room	Spherical	1
Suspended in the middle of the wall	Hemispherical	2
Suspended in the middle of the floor	Hemispherical	2
Placed on the edge of two surfaces	Quarterspherical	4
Placed in the corner	One-eighthspherical	8

The influence of the D factor on sound waves between the machine and the work station may be characterized by the equivalent sound absorption area of the room A in m^2:

$$A = \alpha S, \tag{4}$$

where α is the mean acoustic absorption coefficient determined according to ISO 3744 [7], S is the total area of the surface of the operation room (walls, ceiling and floor) in m^2.

On the basis of the above analysis, the following definition of the power index W_E may be given:

$$W_E = L_{WRGA} - L_{W\mathrm{ref}}\,, \tag{5}$$

where $L_{W\mathrm{ref}}$ is the reference sound power level in dB.

Assuming that the value of the reference sound power level $L_{W\mathrm{ref}}$ should be equal to the maximum value of the A-weighted sound power level of the omnidirectional source installed in the operation room at the place of machine location when the following condition is fulfilled:

• The value of the equivalent A-weighted sound pressure level in the place at that the machine's work station will be located does not exceed the admissible value.

The following final formula determining the power index W_E is obtained:

$$W_E = 10\lg \frac{1}{\sum\limits_{i=1}^{k} \Delta t_i} \left(\sum_{i=1}^{k} \Delta t_i 10^{0.1 L_{WAi}} \right) - 10\lg \frac{10^{0.1L_L} - 10^{0.1L'_{\mathrm{Aeq}}}}{\dfrac{Q}{4\pi d^2} + \dfrac{4}{A}}\,. \tag{6}$$

The general principle of the acoustic assessment of a machine on the basis of the power index is as follows:

• the machine is acoustically safe (i.e. the equivalent A-weighted sound pressure level at the work station in the operation room during its operation does not exceed the admissible value) if the condition $W_E \le 0$ dB is fulfilled,

• the machine is acoustically dangerous (i.e. the equivalent A-weighted sound pressure level at the work station in the operation room during the operation does not exceed the admissible value) if the condition $W_E > 0$ dB is fulfilled.

3. Emission index

For a given machine to be installed under specific operating conditions, an emission index of the acoustic assessment W_I (in dB), is given as the folowing function:

$$W_I = f_3(C_A,\, L_P,\, d,\, L_L), \tag{7}$$

where L_P is the emission sound pressure level of the machine at the work station (in dB).

The parameter characterizing the noise of the machine is the emission sound pressure level L_P which may be written as a function analogous to the function (2). Therefore the real global A-weighted emission sound pressure level L_{PRGA} is described by the formula:

$$L_{PRGA} = 10 \lg \frac{1}{\sum\limits_{i=1}^{k} \Delta t_i} \left(\sum_{i=1}^{k} \Delta t_i 10^{0.1 L_{PAi}} \right), \tag{8}$$

where L_{PAi} is the equivalent A-weighted emission sound pressure level in the i-th variant of the operation, in dB.

Similary to the power index, the following definition of the emission index W_I may be given:

$$W_I = L_{PRGA} - L_{P\mathrm{ref}}\,, \tag{9}$$

where $L_{P\mathrm{ref}}$ is the reference emission sound pressure level, in dB.

Assuming that the value of the reference emission sound pressure level $L_{P\mathrm{ref}}$ should be equal to the maximum value of the A-weighted emission sound pressure level of the omnidirectional source installed in the operating room of the place of the location of the machine when the following condition is fulfilled:

• The value of the equivalent A-weighted sound pressure level at the place of the location of the machine's work station does not exceed the admissible value.

The following final formula determining the power index W_I is obtained:

$$L_{PRGA} = 10 \lg \frac{1}{\sum\limits_{i=1}^{k} \Delta t_i} \left(\sum_{i=1}^{k} \Delta t_i 10^{0.1 L_{PAi}} \right) - 10 \lg \frac{10^{0.1 L_L} - 10^{0.1 L'_{\mathrm{Aeq}}}}{1 + 4 \dfrac{2\pi d^2}{A}}\,. \tag{10}$$

The general principles of the acoustic assessment of the machine on the basis of the emission index are as follows:

• the machine is acoustically safe (i.e. the equivalent A-weighted sound pressure level at the work station in the operation room during the operation does not exceed the admissible value) if the condition $W_I \le 0\,\mathrm{dB}$ is fulfilled,

• the machine will acoustically dangerous (i.e. the equivalent A-weighted sound pressure level at the work station in the operation room during its operation exceeds the admissible value) if the condition $W_I > 0\,\mathrm{dB}$ is fulfilled.

4. Distance correction

On the basis of formula (10) it is possible to calculate the value of the emission index W_I only for such a distance d between the machine and the work station for which the real global A-weighted emission sound pressure level L_{PRGA} from the machine at the work station was determined at laboratory conditions. Because the real global A-weighted emission sound pressure level by the machine at the work station depends on the distance between the machine and the latter, it was necessary to modify formula (10) for the simulation tests. Therefore, a distance correction DC, in dB, was introduced, which takes into account the drop in the emission level with the variation of the distance.

The fact that the emission level of the source is determined by the sound pressure level is the starting-point for working out the distance correction DC. For the sound pressure level of the source, it is possible to assume that its value in the free field (in the far field) is inversely proportional to the square of the distance from the source. At the same time, it is possible to accept that this relation is valid for the indoor environment if the measurement points are located in the area restricted by the limiting distance, i.e. at the distance from the source at which the sound intensity determined by the reverberant field and the sound intensity determined by the free field are in equilibrium. Consequently, assuming that if the range of the variability of the distance between the assessed machine and the workstation is within the area restricted by the limiting distance, the distance correction DC in dB is described by the formula:

$$\mathrm{DC} = -20 \lg \frac{d_1 + \Delta d}{d_1}\,, \tag{11}$$

where d_1 is the distance between the machine and the work station at which the real global A-weighted emission sound pressure level was determined experimentally, in m, Δd is the change of the distance between the machine and the work station in relation to the distance d_1.

Thus, the formula modified for the simulation tests, which makes the determination of the value of the emission index W_I possible, is:

$$W_I = L_{PRGA} + DC - 10 \lg \frac{10^{0.1 L_L} - 10^{0.1 L'_{\mathrm{Aeq}}}}{1 + 4 \dfrac{2\pi (d_1 + \Delta d)^2}{A}}\,. \tag{12}$$

5. Simulation test results of the power index

The influence of the distance d between the machine and the work station and the radiation index Q is presented in Fig. 1. This figure shows that an increase in the distance between the machine and the work station as well as in the value of the radiation index cause a decrease of the value of the power index.

Simulation test results showing the influence of the primary acoustic climate L'_{Aeq} in the operation room and the distance d between the machine and the work station are presented in Fig. 2. In this case, on the basis of the results obtained it is possible to state that an increase of the value of the power index is followed by an increase of the

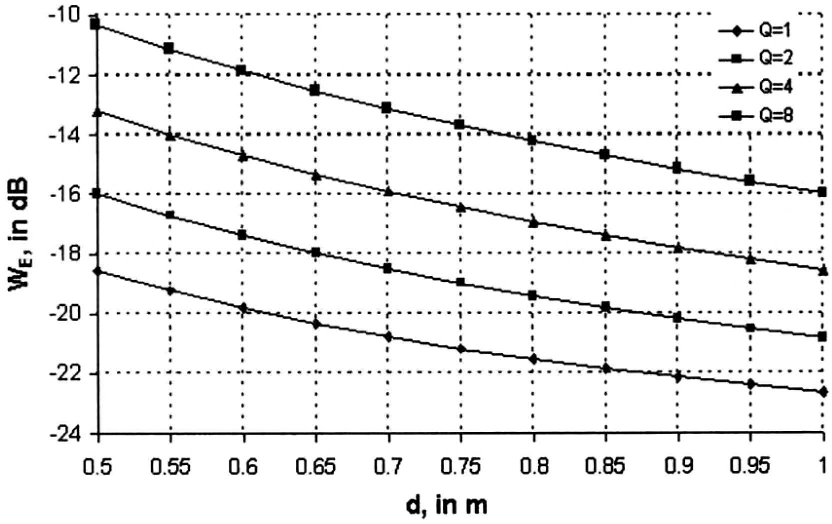

Fig. 1. Influence of the distance d and the radiation index Q on the power index W_E ($A = 54\,\mathrm{m}^2$, $L'_A\mathrm{eq} = 35.4\,\mathrm{dB}$, $L_L = 85\,\mathrm{dB}$, $L_{WRGA} = 84.7\,\mathrm{dB}$).

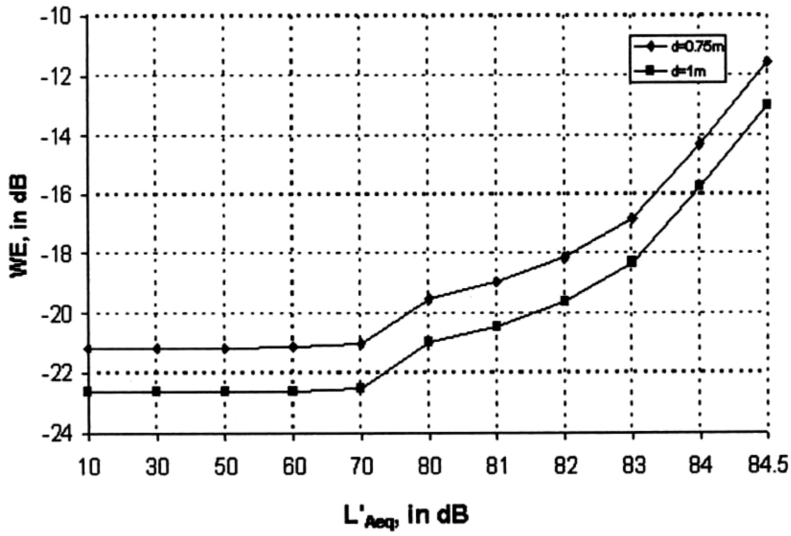

Fig. 2. Influence of the primary acoustic climate L'_{Aeq} and the distance d on the power index W_E ($A = 54\,\mathrm{m}^2$, $Q = 1$, $L_L = 85\,\mathrm{dB}$, $L_{WRGA} = 70.5\,\mathrm{dB}$).

equivalent A-weighted sound pressure level characterizing the primary acoustic climate in the operation room and the reduction of the distance between the machine and the work station. At the same time, the results show that the influence of the equivalent A-weighted sound pressure level characterizing the primary acoustic climate, are especially significant if the difference between the admissible value of the equivalent A-weighted sound pressure level at the work station and the value of the equivalent A-weighted sound pressure level characterizing the primary acoustic climate this level, is less than 15 dB (e.g. in the case when the admissible value of the equivalent A-weighted sound pressure level at the work station is 85 dB, the influence of the equivalent A-weighted sound pressure level characterizing the primary acoustic climate is significant if its value exceeds 70 dB).

Figure 3 presents the influence of the equivalent sound absorption area of the operation room A and the distance d between the machine and the work station on the

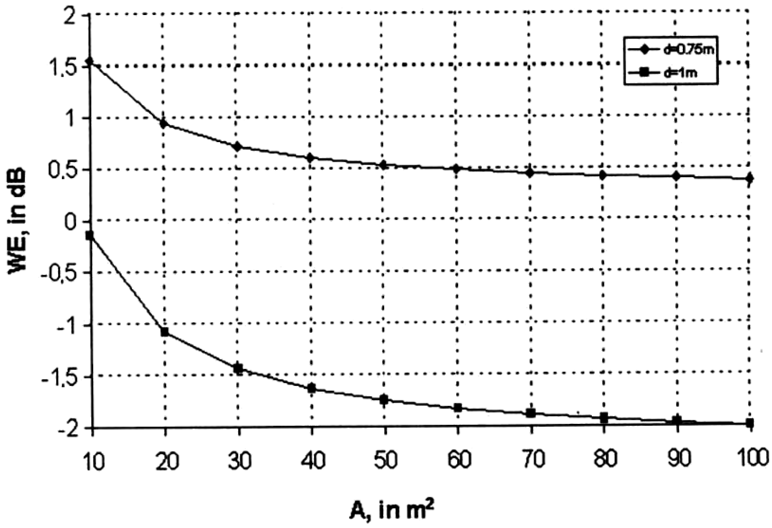

Fig. 3. Influence of the equivalent sound absorption area of the room A and the distance d on the power index W_E ($Q = 8$, $L'_{A\mathrm{eq}} = 35.4$ dB, $L_L = 85$ dB, $L_{WRGA} = 85$ dB).

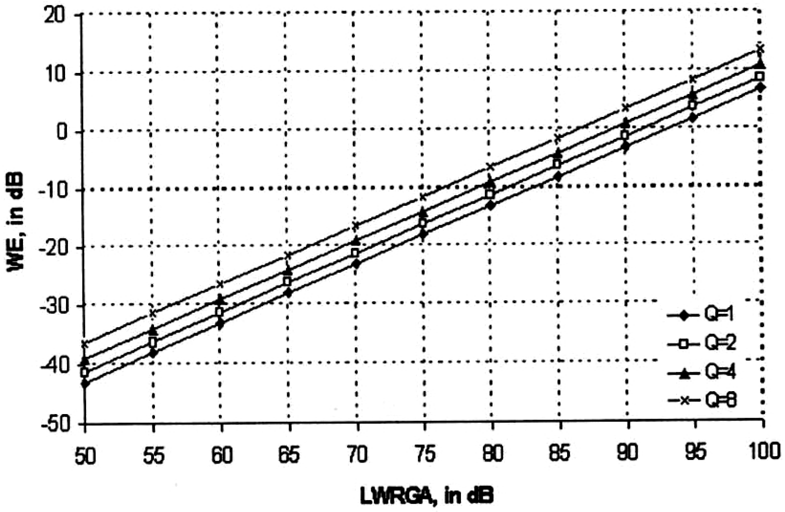

Fig. 4. Influence of the real global A-weighted sound power level L_{WRGA} on the value of the power index W_E ($A = 54\,\mathrm{m}^2$, $d = 1$ m, $L'_{A\mathrm{eq}} = 35.4$ dB, $L_L = 85$ dB).

value of the power index W_E. On the basis of the results obtained it is possible to state that an increase of the value of the equivalent sound absorption area of the operation room causes a decrease in the value of the power index. The most significant influence of the changes of the values of the equivalent sound absorption area of the room on the changes of the values of the power index is observed in the region below $70\,\mathrm{m}^2$. At the same time, the acoustic assessment result will be more favourable if the distance between the machine and the workstation is greater.

The influence of the changes of the values of the real global A-weighted sound power level L_{WRGA} on the value of the power index W_E is presented in Fig. 4. This figure shows that an increase in the value of the real global A-weighted sound power level causes a linear increase of the value of the power index.

6. Simulation test results of the emission index

Figures 5 and 6 present the simulation test results of the emission index W_I as a function of the real global A-weighted emission sound pressure level from the machine at the work station L_{PRGA} and the change of the distance between the machine and the work station Δd. The test results obtained in this case show that an increase in the value of the emission index is followed by an increase in the value of the real global A-weighted emission sound pressure level of the machine at the work station and the reduction of the distance between the machine and the work station. At the same time, there is a linear relation between the emission index and the value of the real global A-weighted emission sound pressure level of the machine at the work station.

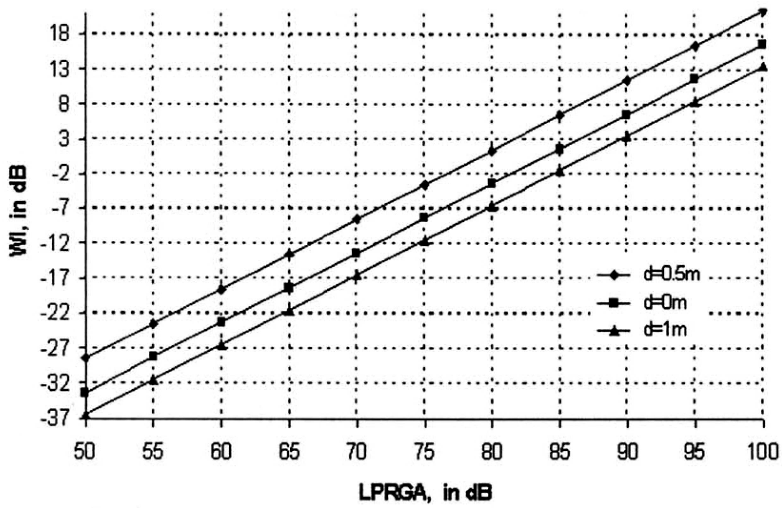

Fig. 5. Influence of the real global A-weighted emission sound pressure level L_{PRGA} on the value of the emission index W_I ($A = 54\,\mathrm{m}^2$, $L'_{\mathrm{Aeq}} = 35.4\,\mathrm{dB}$, $L_L = 85\,\mathrm{dB}$).

Simulation test results showing the influence of the primary acoustic climate L'_{Aeq} in the operation room and the change of the distance between the machine and the work

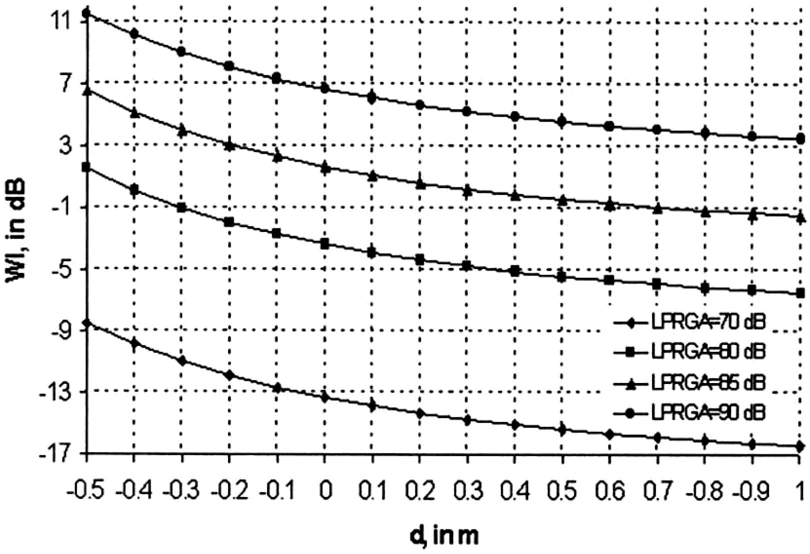

Fig. 6. Influence of the change of the distance Δd on the value of the emission index W_I ($A = 54\,\mathrm{m}^2$, $L'_{\mathrm{Aeq}} = 35.4\,\mathrm{dB}$, $L_L = 85\,\mathrm{dB}$).

station Δd are presented in Fig. 7. On the basis of the results obtained it is possible to state that an increase of the value of the emission index is followed by increase in the equivalent A-weighted sound pressure level characterizing the primary acoustic climate in the operation room. In the same way as in the case of the power index, this influence of the equivalent A-weighted sound pressure level characterizing the primary acoustic

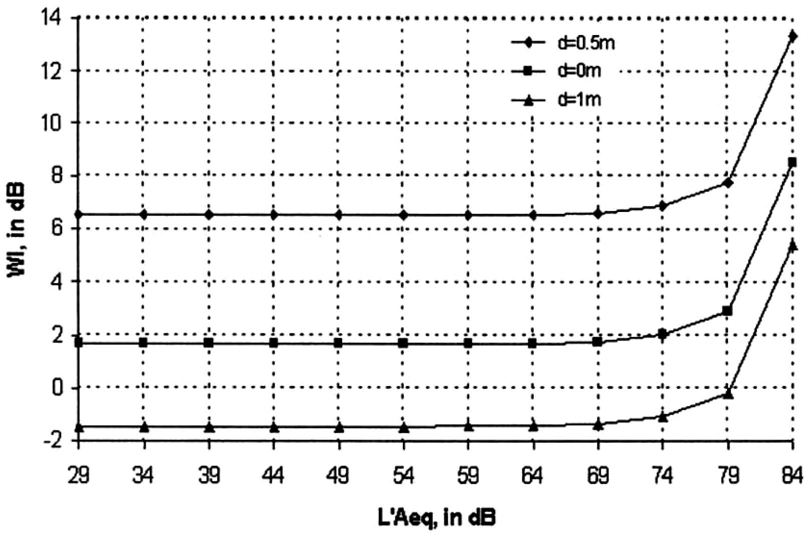

Fig. 7. Influence of the primary acoustic climate $'_{\mathrm{Aeq}}$ and the change of the distance Δd on the value of the emission index W_I ($A = 54\,\mathrm{m}^2$, $L_L = 85\,\mathrm{dB}$, $L_{PRGA} = 85\,\mathrm{dB}$).

climate is especially significant if the difference between the admissible value of the equivalent A-weighted sound pressure level at the work station and the value of the equivalent A-weighted sound pressure level characterizing the primary acoustic climate is less than 15 dB.

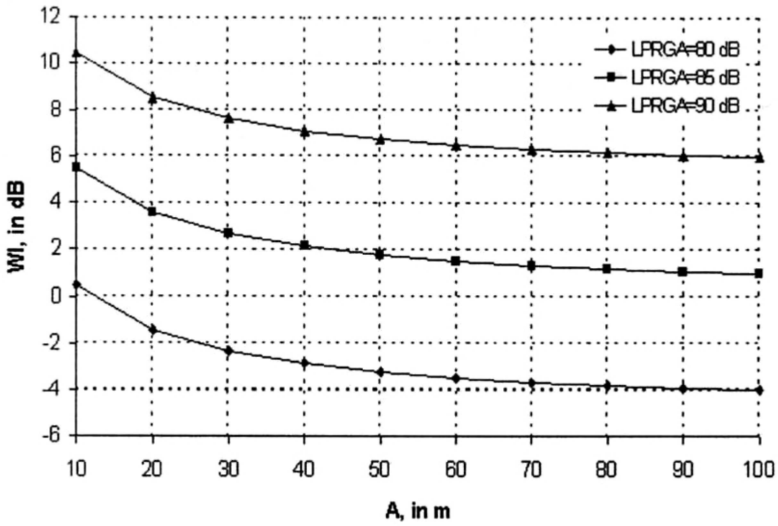

Fig. 8. Influence of the equivalent sound absorption area of the operation room A on the value of the emission index W_I ($L'_{\mathrm{Aeq}} = 35.4$ dB, $L_L = 85$ dB, $d = 1$ m).

Figure 8 presents the influence of the equivalent sound absorption area of the operation room A. The test results obtained show that an increase in the value of the equivalent sound absorption area of the operation room causes a decrease of the value of emission index.

7. Summary

The simulation test results are consistent with the experimental tests results [4], with the general principles of sound wave propagation and with the noise control methods. Sample results of the assessment of different noise sources using the power and emission indices (assuming that $L_L = 85$ dB), cofirmed by the measured values of the equivalent A-weighted sound pressure levels ($L_{\mathrm{Aeq}M}$), are given in Tables 2 and 3.

Table 2. Results of assessments using the power index.

Noise source	drill		centrifuge	
L_{WRGA}, in dB	84.7		73.2	
d, in m	0.75	1	0.5	1
W_E, in dB	0.52	−1.78	−7.6	−13
$L_{\mathrm{Aeq}M}$, in dB	85.5	84.2	75.6	74.8

Table 3. Results of assessments using the emission index.

Noise source	grinder		mixer	
L_{PRGA}, in dB	78		68.8	
A, in m^2	5.7	13.3	5.7	13.3
W_I, in dB	0.3	−2.4	−13	−14.5
L_{AeqM}, in dB	89	82.2	74.1	70.1

It is possible to achieve a favourable acoustic assessment result (i.e. the value of the assessment index is not greater than 0 dB) in the following ways:

• by reducing the noise emission of the machine and, by decreasing thereby the real global A-weighted sound power level (the real global A-weighted emission sound pressure level), and

• by shaping suitably the operational conditions, i.e. by increasing the equivalent sound absorption area of the operation room, changing the machine location in the room, by increasing the distance between the machine and the work station and decreasing the equivalent A-weighted sound pressure level characterizing the primary acoustic climate.

Acknowledgements

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CURRENT SITUATION AND FUTURE TOPICS OF ROAD TRAFFIC NOISE PROBLEMS IN JAPAN

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Regarding the road traffic noise problem in Japan, the current situation and the administrative countermeasures are reviewed. Next, from the technological viewpoint, the progress of regulation of the vehicle noise emission and various kinds of noise reduction technologies are introduced. Regarding the assessment and prediction methods for road traffic noise, the historical course in Japan and future problems are briefly described. Lastly, as future subjects, not only technical developments but also administrative and socio-psychological measures are discussed.

1. Introduction

Japan has made a great stride industrially and economically during its rapid industrial development in the decade from about 1965. From this period, the volume of goods' transport has much increased. The amount of goods' transport in 1994 has become three times larger compared to that in 1965 and about 50% of the total amount of goods' transport is dependent on road vehicles. Meanwhile, motorization has greatly progressed and the number of motor vehicles owned has increased at almost the same rate (from 21 000 000 vehicles in 1971 to 68 000 000 in 1994).

Although road vehicles are indispensable for industrial activities and our everyday life, the progress of motorization has caused such environmental problems as air pollution and noise. In this paper, the current situation and future topics of road traffic noise problems in Japan are reviewed.

2. Current situation and administrative countermeasures

In Japan, we have "Basic Environmental Law" which aims to maintain standards desirable for the protection of human health and preservation of living environments. Based on this law, "Environmental Quality Standards" are specified. Regarding environmental noise, the desirable standards were decided in 1971 for general areas which are classified into two categories facing roads and other areas. According to these standards, various environmental conservation plans have so far been established. The annual "Report on the Environment" published in 1995 reported that the attainment rate of the environmental quality was more than 60% for areas other than areas facing roads, whereas the rate was only 14% for areas facing to roads. These statistics indicate that the road traffic noise problem is still very serious in Japan. Especially in urban roadside areas on trunk roads, the influence of road traffic noise is very serious.

3. The progress in noise abatement technologies

In alignment with Fig. 1, the present state of noise control technologies is reviewed below.

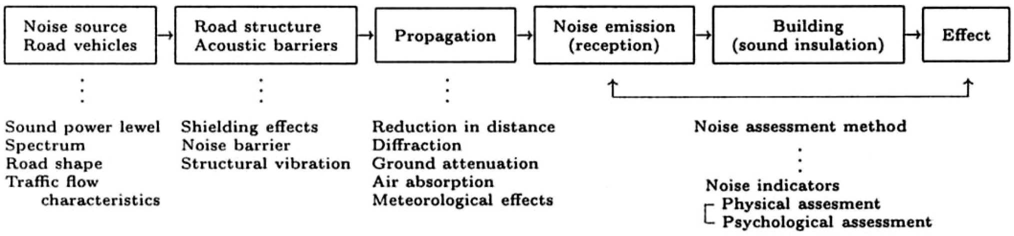


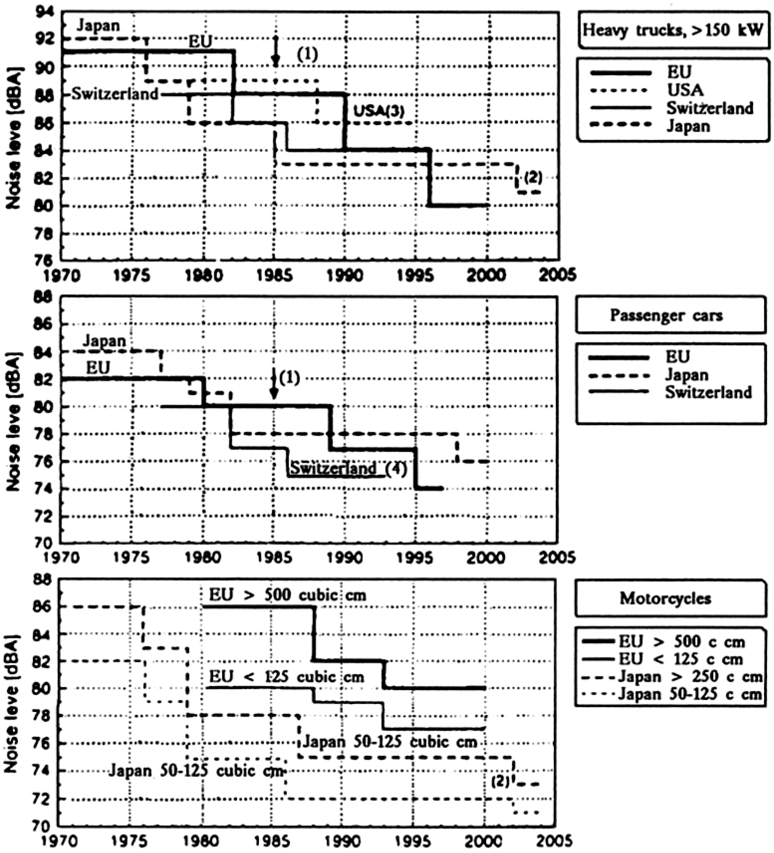
Fig. 1. Flow of the road traffic noise problem [Tachibana].

3.1. Regulation of vehicle noise emission

As a primary measure for the reduction of road traffic noise, noise abatement of road vehicles must be considered. Many industrialized countries have introduced regulations regarding maximum noise emission of road vehicles. Figure 2 shows the development of the vehicle noise emission limits regarding acceleration pass-by noise over the years in the EU, USA, Switzerland and Japan [1]. In Japan, the regulation for constant speed pass-by noise and exhaust noise was established in 1951. Regarding the acceleration pass-by noise, which is most important in urban areas, the regulations for heavy trucks, passenger cars and motorcycles began in 1971 and have been strengthened in three steps in 1976–1977, 1979 and 1982–1987.

Meanwhile, automobile manufacturers have been making a big effort to reduce the vehicle noise emission. The technical points are mechanical improvements of the engine (combustion process, intake system, etc.), acoustic treatments of the engine compartment, prevention of structure-borne sound, improvement of the transmission, differential gears, gear box and muffler, control of the fan, damping treatment of the propeller shaft, reduction of aerodynamic noise and so on.

To see the effect of such vehicle noise regulations on the actual traffic noise, a Working Party was set up in the I-INCE in 1992 and conducted international research until 1994. In the report of this research, it was concluded that, in spite of phased vehicle noise emission regulations in many countries, a tendency of decreasing of road traffic noise could not be clearly seen. The report described the reasons for this discouraging result



Notes:
1. The arrow indicates that in EU there was a change in the measuring procedure in 1985. For trucks, this corresponded to approximately 4 dB (A) of more stringent requirements on top of the other changes.
2. In Japan limit changes as indicated are targeted by 2002.
3. Truck values for USA increased by 6 dB to compensate for twice as large a measuring distance.
4. Cars with 4 gears (manual) may emit up to 77 dB (A).
5. In the USA, there are no noise requirements for cars.

Fig. 2. The development of vehicle noise emission limits over the years, including projected limits. (Top parts of the figure for trucks of 150 kW, middle part for cars and bottom part for motorcycles.) [1].

indicating that the contribution of the tire/road noise, which is dominant in actual road traffic noise, can not be assessed by the vehicle type approval test according to ISO 362 and the replacement of vehicles in service takes a long time and that the effect of vehicles noisier than average tends to dominate the results of the field measurements. However, some promising results were also introduced in the report. For example, Fig. 3 shows the data of acceleration noise measured at intersections with traffic signals in Japan. In these results, a slight decrease of the acceleration pass-by noise can be seen for both heavy trucks and passenger cars between the vehicles in conformity with the regulation in 1979 and in conformity with those in 1985. Figure 4 shows the sound power levels of vehicles under the constant speed running condition measured by a special technique

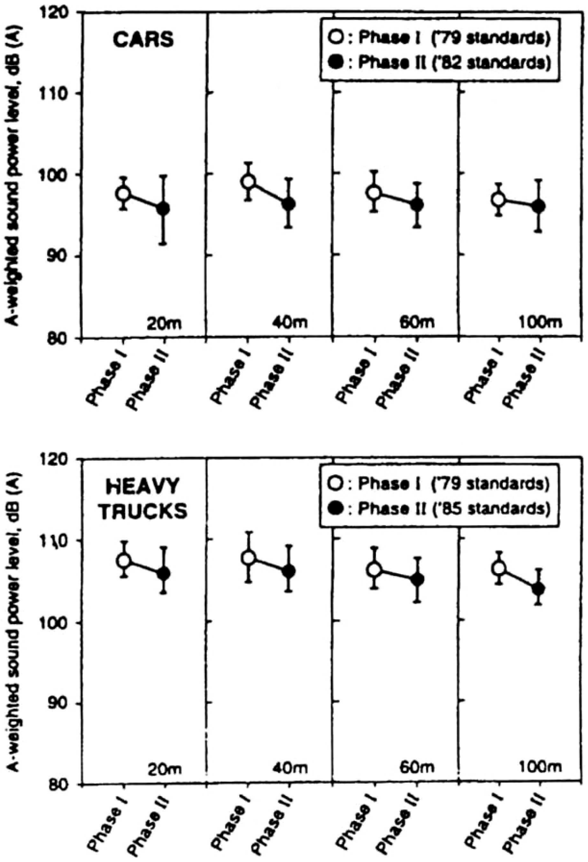


Fig. 3. Sound power level (maximum at pass-by 7.5 m from a microphone) from accelerating vehicles measured at different distances from a stoplight. Cars in the left part and heavy trucks in the right part of the figure [1].

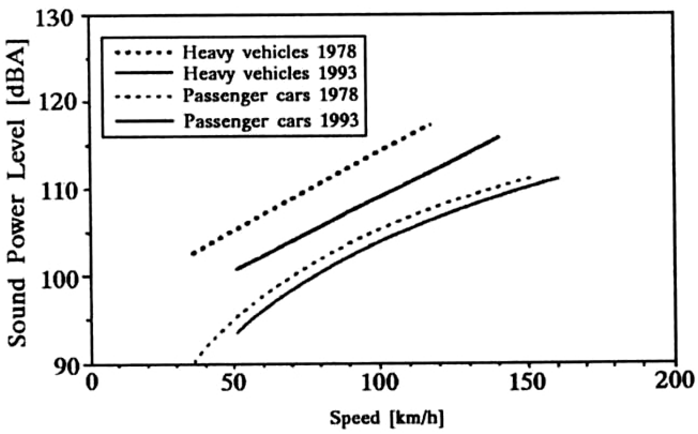


Fig. 4. Change in sound power levels of Japanese vehicles 1978–93, as measered in a tunnel [1, 2].

using a reverberand tunnel [2]. In this result, a decrease of the sound power level by 1 to 2 dB for passenger cars and by 4 to 5 dB for heavy trucks was seen between the data obtained in 1978 and those in 1993.

3.2. Reduction of tire/road noise

The dominant components of vehicle noise are the engine noise and the tire/road noise. The former has been reduced to some extent by the technical improvements mentioned above and, consequently, the latter has become relatively prominent. To reduce the tire noise, various improvements are being tried on the tread pattern, the structure and materials of tires.

Regarding the road surface, several kinds of low-noise pavements have been developed.

The most hopefull type for practical use is the drainage asphalt pavement which was originally developed for safety under rainy conditions. The porosity of the surface of this pavement is effective for sound absorption and, consequently, noise radiation can be reduced by 3 to 5 dB (A) [3]. The application of this type of pavement has already been started as an innovative noise reduction measure, whereas it is necessary to examine its endurance property and to develop a technique of recovering its sound absotption efficiency.

3.3. Measures by road structures

As a matter of course, the underground tunnel structure is the best one for roadside noise problems, but it is necessary to treat exhaust gases by artificial ventilation tech-

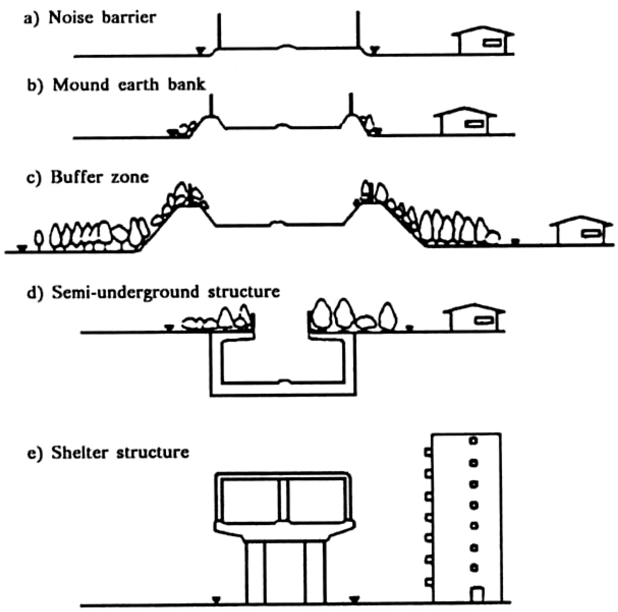


Fig. 5. Measures for highway structures.

niques. As the second best measure, semi-underground structures with openings in the ceiling have been developed and are being adopted in those areas, where a serious noise problem is estimated (Fig. 5).

In this type of structure, the sound absorption treatment on the side walls is very effective to suppress the multi-path reflection inside the structure and consequently to reduce the noise propagation outside. Recently, shelter type structure is introduced to densely built-up urban areas (Tokyo Outer Ring Road).

3.4. Use of noise barriers

Noise barriers are the most effective measure for the reduction of road traffic noise. In Japan, it has been widely used for highways for more than twenty years and the total length of noise barriers amount to about 3 460 km at the end of the 1994 fiscal year. The height of the barriers was about 3 meters previously, whereas it has recently become necessary to construct barriers of 5 to 8 meters high because of the rapid increase of the traffic volume. Since such a high barrier causes secondary problems of deterioration of the landscape and the obstruction of sunshine, various types of noise barriers with relatively low height and high noise reduction performance are being developed (Fig. 6).

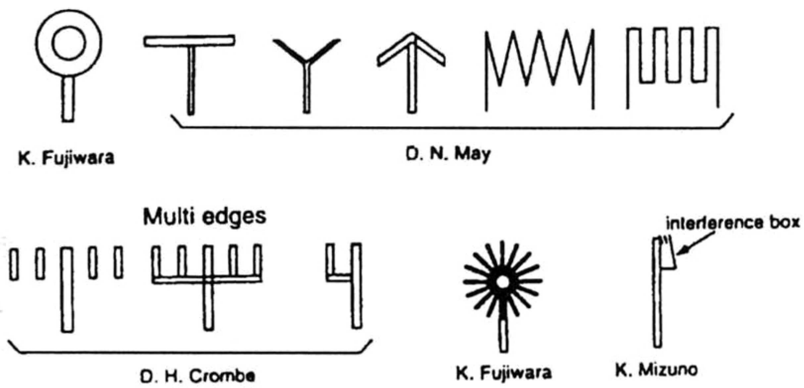


Fig. 6. Noise barriers of special shapes.

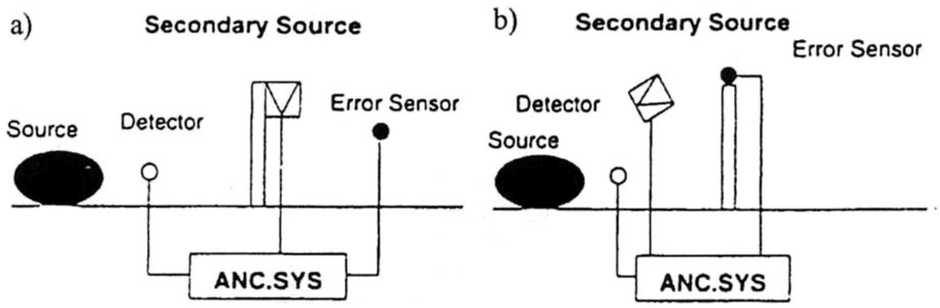


Fig. 7. Experiments of active noise barrier; a) S. Ise *et al.*, b) A. Omoto *et al.*

As a new idea, the application of an active control technique to noise barriers is being investigated on the experimental stage (Fig. 7).

3.5. Environmental buffer zone

For noise problems, it is the most basic principle to keep a long distance between the noise source and the area influenced. For the traffic noise problem, it is the most desirable measure to provide an environmental buffer zone (Fig. 8). In Japan, the total length of buffer zones made for this purpose was only about 590 km in 1995 but it is gradually increasing. Here, it should be noted that the planting and greenfication of the ground cover are effective for landscape beautification and subjective impression, but they are not so effective for noise reduction as it is generally expected.

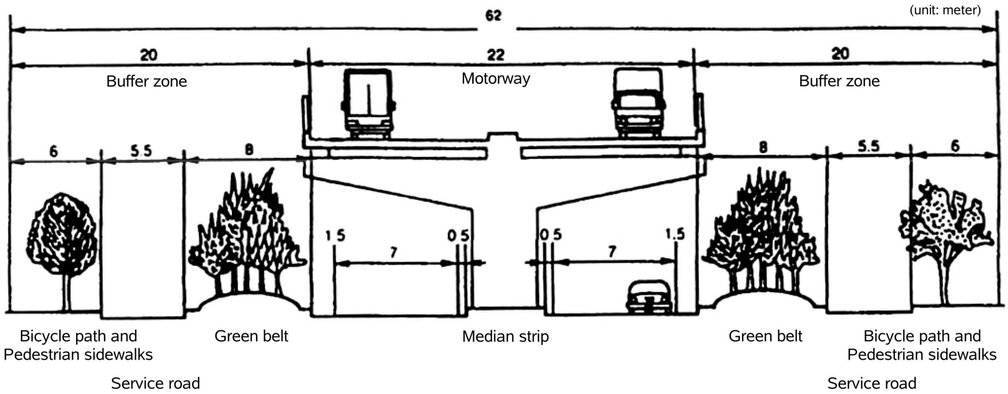


Fig. 8. Highway with buffer zone (Tokyo Outer Ring Road).

3.6. Sound insulation of roadside buildings

According to the draft recommendation by the WHO, the desirable noise level inside residential houses should be lower than 30 dB (A) for healthy sleeping and therefore lower than 45 dB (A) outside when considering the sound insulation performance of common houses. In the case of arterial roads in urban areas, however, it is almost impossible to sustain such a low noise level outside the buildings and therefore it is necessary to improve the sound insulation of buildings to realize desirable noise levels inside the buildings. As an administrative measure regarding sound insulation of roadside buildings, the "Law for improvement of the areas along trunk roads" was established in 1980 and partially revised in 1996. The aim of this law is to prevent serious road traffic noise hazards and to promote land use of roadside areas. In this law, the subsidy system for sound proofing works is specified, which can be applied if the noise level is over the specified value (for example, 65 dB (A)) both in the measurement and in the calculation for specified areas. If the conditions are fulfilled, such sound proofing works as reinforcement of building facades, provision of double glazed windows and air-conditioning systems are performed at the expense of the road administrators. This law also specifies the procedures of improving areas along trunk roads, such as urban planning in which the enhancement of the construction of buffer buildings and sound insulation of houses and schools are included. Up to this time, however, it can not be said that this law has fulfilled its function because of the difficulties of reaching a consensus among residents for the roadside measures and because the budget is not sufficient. In the future, this law should be improved and expanded so as to promote the construction of roadside buffer buidings which have sufficient sound insulation and are effective to protect the back areas from the road traffic noise.

4. Assessment method for road traffic noise assessment

As an indicator for environmental noises, L_{50} has been videly used in Japan so far and the Environmental Quality Standards are specified by this indicator. On the other hand, $L_{A\mathrm{eq}}$ is being used almost all over the world, and recent physiological and psychological researches have indicated that the doze-response relationship of noise is well assessed by $L_{A\mathrm{eq}}$. For these reasons, the Environmental Agency has started an investigation of the introduction of $L_{A\mathrm{eq}}$ into the Environmental Quality Standards. Thus, $L_{A\mathrm{eq}}$ will be adopted as the main indicator for the assessment of environmental noises in the near future in Japan.

5. Progress of the prediction method of road traffic noise

When dealing with noise problems, noise prediction methods are indispensable. Regarding road traffic noise, the Technical Committee of Road Traffic Noise Prediction was established at the Acoustical Society of Japan in 1974 and it proposed the first prediction method in 1975. This prediction model (ASJ-Model 1975), which provides L_{50} values through relatively simple calculations, has been widely used for the impact assessment of road traffic noise in Japan. On the other hand, it has become necessary to expand the applicability of the prediction method and to predict $L_{A\mathrm{eq}}$ at roadside areas. Hence, in 1988, the Committee started research work to develop an energy-based road traffic noise prediction model. As a result, a new model (ASJ-Model 1993) has been proposed in 1993 [4, 5]. In this model, a time pattern of sound pressure (unit pattern) at an observation point is calculated first and by integrating the pattern and considering such traffic conditions as the traffic volume, mean speed and vehicle constitution, $L_{A\mathrm{eq}}$ can be obtained as an energy-based time averaged value. In principle, the model can be applied to almost all types of roads including such special parts as interchanges and junction. The concrete calculation procedures are now being investigated by the Committee.

6. Conclusions

To conclude this paper, future topics for the improvement of the road traffic noise problem are enumerated below.

Firstly, from the engineering viewpoint, the following subjects should be further investigated.

• On-the vehicle noise source control: reduction of engine noise and tire/road noise, development and diffusion of low pollution cars;

• Improvement of road structures: underground, semi-underground and shelter type structures, development of effective noise barriers (phase control and active control), development of low noise pavements, absorption treatment of road structures, design of environmental buffer zone including planting and greenfication;

• Traffic control: speed control, heavy truck control, optimum control of traffic signals;

• Development of roadside buildings: new concepts of integrating road structures and buildings, new designs of building facades, development of building elements and facilities with high sound insulation;

• Improvement of the noise prediction methods.

Since the road traffic noise problem is a very complex social phenomenon, it can not be settlet only be engineering measures and the following counterplans must be considered at the same time.

• Urban planning: land use plans, transfer of urban structures, roadside countermeasures;

• Economic and transport system: total transportation schemes, modal shift, introduction of new transportation systems, road network planning, proper configuration of facilities for physical distribution, development of new physical distribution systems, traffic control measures;

• Legal and administrative improvement and enforcement of related laws (Law for improvement of areas along trunk roads, Building Codes, etc.);

• Socio-psychological approach: establishment of environmental ethics, consciousness of the environmental load, consensus of roadside measures.

In previous days, construction and preservation of the environment were apt to be considered contrary to each other but they have to be reconciled together to the future.

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C H R O N I C L E

General Assembly of Delegates of Polish Acoustical Society

On September 13th in Zakopane took place the General Assembly of Delegates of Polish Acoustical Society (PAS). In Assembly the new Authorites of PAS were elected.

In 1999–2000 term the Authorities are:

President of PAS – prof. Jerzy Ranachowski

Main Board:

V-president – prof. Aleksander Opilski
General Secretary of PAS – dr hab. Tadeusz Pustelny
Treasure of PAS – dr Roman Bukowski.

Members of Main Board:

prof. Antoni Śliwiński,
dr Maria Rabiega,
dr Henryka Czyż,
dr Bogumił Linde.

Board of Control:

prof. Roman Salamon,
dr Marianna Mirowska,
dr Lucyna Leniowska,
dr Marek Iwaniec.

Arbitration Court:

prof. Eugeniusz Kozaczka,
prof. Mikołaj Łabowski,
dr Jacek Cieślik.

The General Assembly of Delegates of PAS in the ballet has conferred the Title of Honour Member of Polish Acoustical Society:

doc. Mariannna Sankiewicz (Gdańsk Division of PAS),
prof. Adam Lipowczan (Upper Silesia Division of PAS),
prof. Leif Bjorno (from Denmark).

The ceremony of delivery of Diplomas of Honour Member of PAS will take place at the next Open Seminary on Acoustics in 2000.

The XLVI Open Seminar on Acoustics OSA'99 took place in Zakopane on September 14–17, 1999.

The main Organiser of OSA'99 was Kraków Division of Polish Acoustical Society.

The Seminar was attended by about one and a half acousticians from Poland and even one hundred scientific lectures and short lecture were presented from all acoustic domains. In Seminar were also the guests from Geremany, United Kingdom, Denmark and Russia. According the opinion of participates the scientific level of Seminary was high.

During the Opening Ceremony of OSA'99 there were handed Diplomas of Honour Membership of Polish Acoustical Society to:

prof. Antoni Śliwiński – the many years standing President of PAS,

prof. Jens Blauert – the many years standing Chairmen of the Board of European Acoustic Association.

DISSERTATIONS

Determination of the spatial distribution of spectral components of ship's noise [in Polish]
by Paulina Bittner
23 December 1998
Adam Mickiewicz University, Institute of Acoustics,
Faculty of Navigation, Naval Academy,
ul. Śmidowicza 71, 81-919 Gdynia, Poland
Supervisor: Prof. D.Sc. Eugeniusz Kozaczka

The investigations of vibroacoustic activity of a ship is a topic of interest for technical diagnostics, ecological and military reasons. From the acoustics point of view a ship may be treated as a coaxial sound source. Descriptions of the field distribution for such sources are common in literature. The methods used for solving the radiation problem include analytical solutions, numerical methods and experimental investigations. As a ship is a complex sound source it is difficult to find an analytical description of its radiation in the near field region. Therefore the thesis describes a method which has the elements of a numerical and experimental one. The hull is surrounded with a surface which replaces a ship as a source. Experimentally obtained surface distributions of acoustic quantities are used to calculate the pressure value in any point in space outside the virtual surface. The numerical method applied in the thesis has been worked out basing on the Helmholtz solution of the exterior radiation problem and the superposition method. It was verified by comparing its results with the approximate analytical dependence found out for a simple source of the shape similar to the virtual vibrating surface. The method of measuring the acoustic quantities is described next where the special attention was paid to the particle velocity measurements. The theoretical background is followed with the results of laboratory measurements. The next step was to check the method during "in situ" measurements. They have been carried out in two measuring set-ups over different types of the sea bed. The worked out method can be used to estimate the acoustic noise of the ship during its exploitation.

Quality of synthesized organ pipe sounds versus parameters of the digital waveguide models [in Polish]

by Sławomir K. Zieliński
15 December 1997
Sound Engineering Department, Technical University of Gdańsk,
Contact: Technical University of Gdańsk, Main Library,
ul. Narutowicza 11/12, 80-952 Gdańsk, Poland
Supervisor: Andrzej Czyżewski

The objective of the work was to prove that the digital waveguide models of the organ pipe could be used for synthesis of organ sound with high quality of transient states.

The digital waveguide model of the organ flue pipe was elaborated. This model takes into account both geometrical dimensions of the pipe and air pressure. The elaborated model is computationally efficient, so its implementation on a digital signal processor (DSP) was possible.

The first group of experiments was related to simulations of presssure changes in the elaborated model. Obtained results showed that the simulated pressure affects pitch of the synthesized sound. Moreover, various kinds of overblowing can also be obsereved. This result remains in accordance with real pipe behaviour.

The second group of experiments concerned analysis of the model response to the simulated changes of the pipe mouth and to the simulated changes of the displacement between the upper lip and the air jet. Simulation of changes of the pipe mouth height results in pitch variation, whereas simulated changes of the displacement between the upper lip and the air jet lead to spectral variations (balance between odd and even harmonics). Mentioned results also agree with results of similar experiments concerning real pipes.

The last part of dissertation contains results of systematic subjective tests. These tests showed that synthetic sounds obtained using the elaborated model are satisfactory.

Exemplary synthetic sounds obtained using the elaborated model are attached to the dissertation on the CD-ROM.