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RECENT EQUIVALENT SOURCE METHODS FOR QUANTIFYING AIRBORNE AND STRUCTUREBORNE SOUND TRANSFER

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Summary: Characteristically noise reduction in technical products like road vehicles, ships, aircraft and machines is complicated by the multitude of primary sources and transfer paths. Examples of well-known approaches for transfer path investigations are selective shielding, mechanical uncoupling, "reduced" impedance methods and simple substitution methods (e.g. loudspeakers).
This paper presents a brief survey of some recently developed substitution methods which are somewhat more advanced. A characteristic feature of these methods is that the excitation of an individual transfer path is modeled on the basis of in-situ measured source or path responses. Transfer functions to receiver points are measured by invoking the reciprocity principle. For airborne sound transfer two methods are described that model a radiating surface with the aid of a distribution of point monopole sources. A "deterministic" variant uses the volume acceleration distribution and a "statistical" variant the radiated power distribution. For structure-borne sound transfer two methods are discussed which reproduce approximately the in-situ measured vibrational response on a transfer path. One method uses linear responses (i.e. accelerations) to characterise the vibration field. The other method uses structural intensities.
Some experimental results are shown as examples.

1 INTRODUCTION

Low noise design of products is usually limited by several practical constraints. Obviously, a most important demand is that a costly overkill in noise reduction measures must be prevented. This forms a major incentive to improve sound path quantification procedures, because in many situations the noise transmission occurs simultaneously along various transfer paths. In this paper we present some recently developed methods for quantifying airborne and structure-borne sound transfer.

2 AIRBORNE SOUND

2.1 Equivalent volume acceleration point sources

Consider a machine which radiates airborne sound and also transmits structure-borne sound to the surroundings via mechanical connections. A relevant question might be whether it would be necessary to improve the sound insulation of the whole machine or of partial areas to reduce the sound level at a distant receiver point. To answer this question an accurate method is needed to quantify the contribution of the airborne sound transfer. Fahy et al. have proposed a reciprocity method for this purpose which may be called the "directional Green's function method" [1-3].

Figure 1 shows a practical example. A shipboard machine causes underwater sound. To determine the contribution of airborne sound transfer the radiating surfaces of the machine are subdivided in discrete areas ΔS_i. When these are small compared to the structural wavelengths in the machine and the acoustic wavelength in air, they may be considered as acoustic point monopole sources positioned on the machine surface. The volume acceleration may be written as $\dot{Q}_{1,i} = a_{1,i}\Delta S_i$, where $a_{1,i}$ denotes the acceleration of ΔS_i normal to the surface. With modern equipment the phase relationships between a_i can be easily measured. To calculate the fraction of the underwater sound that is determined by airborne sound transfer one needs to multiply each of the individual "monopole" source strengths with a transfer function $p_w/\dot{Q}_{1,i}$. The determination of these transfer functions becomes practicable by invoking the reciprocity principle, which implies that $p_w/\dot{Q}_{1,i} = p_{1,i}/\dot{Q}_w$. For the example of figure 1 an omnidirectional underwater sound source is used and with the machine switched off, sound pressures $p_{1,i}$ are measured directly against the machine surfaces. Ref. [1] describes the construction of a monopole source for airborne sound.

The simplest expression for the calculated contribution of airborne sound transfer to the underwater sound is the superposition formula:

$$p_w = \sum_i a_{1,i}\Delta S_i \cdot \left[p_{1,i}/\dot{Q}_w\right] \tag{1}$$

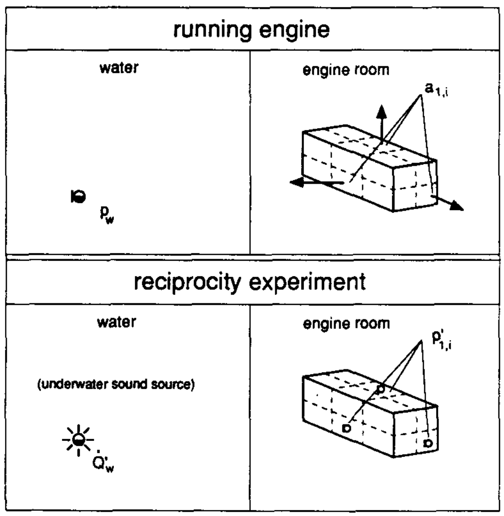

Figure 1 Reciprocity method for quantifying airborne sound transfer from a machine to the receiver at low frequencies. The sound radiation is modelled by discrete point sources on the machine surface. Transfer functions for these point sources are measured reciprocally.

If all responses are generated by a single source mechanism it will suffice to use one of the responses as a reference signal defining the phases of all other responses. If several (partially) incoherent sources are responsible for a_i, more cross-spectra are needed [1].

Ref. [1] presents the experimental results for a mechanically excited aluminium plate (0.48 m × 0.48 m × 6.25 mm). For 1/3-octave bands with centre frequencies 400-2000 Hz the differences between the measured sound pressure level at a distant point and that predicted on the basis of 81 "point sources" remained within 3 dB. However, neglecting the phase relationships between these "point sources" leads to "errors" up to 10 dB.

Practical problems are formed by the selection of an appropriate "mesh" for ΔS_i and by the measurements of a_i. The method is especially promising for sources with rather simple vibration patterns, e.g. car engines at low frequencies. Fahy et al. have recently proposed a method for a contactless measurement of structural volume velocity [4]. For radiators with more irregular surfaces and complex vibration fields an alternative method is proposed next.

2.2 Equivalent power point sources

Another method for the previous problem has been described by Verheij [5]. Figure 2 shows an example to demonstrate the principle. The engine is enclosed by m flat measurement surfaces parallel to the surfaces of the machine. These partial surfaces are selected in such a way that sound intensity measurements may be performed from a stationary position, using "hand-scanning".

For partial surface S_j the radiated power follows from

$$P_{rad}(j) = \int_{S_I} I_n \;\; dS \tag{2}$$

where I_n is the sound intensity vector normal to S_j.

For each surface S_j a transfer function to underwater is defined by $[p_w^2(j)/P_{rad}(j)]$. The contribution to the underwater sound of the airborne path is obtained from

$$p_w^2 = \sum_{j=1}^{m} p_w^2(j) = \sum_{j=1}^{m} [p_w^2(j)/P_{rad}(j)].P_{rad}(j) \tag{3}$$

Therefore, in addition to sound powers according to eq. (2), transfer functions to underwater are to be measured.

The assumption is made that the sound radiation of S_j may be replaced by that of n(j) uncorrelated fictitious monopole point sources with volume acceleration $\dot{Q}_o$. These are located on S_j and the radiated power is given by

$$P_{rad}^{fic}(j) = \frac{\rho}{4\pi c}.\dot{Q}_o^2 \sum_{i=1}^{n(j)} C_{R,i} \tag{4}$$

$C_{R,i}$ is a weighting factor between 1 and 2, which depends on the distance of the fictitious monopole sources to an edge or corner of the machine.

In ref. [5] it has been shown that $C_{R,i} = 2$ would form a good approximation in many cases. Using the reciprocity principle in the same way as in 2.1, the radiated underwater sound due to S_j follows from

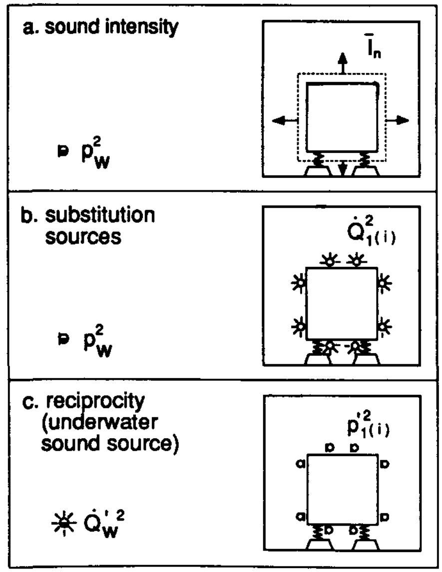

Figure 2 Equivalent power point sources method. A set of fictituous point sources is defined on the machine surface to generate the same sound power as the machine. The transfer function of these point sources to the receiver is measured reciprocally.

$$p_w^2(j) = P_{rad}(j).\left[\frac{p_w^2(j)}{P_{rad}^{fic}(j)}\right] = P_{rad}(j).\left[\dot{Q}_o^2 \sum_{i=1}^{n(j)} \frac{p_{1,i}^2(j)}{\dot{Q}_w^2}\right].\frac{1}{P_{rad}^{fic}(j)} = \frac{P_{rad}(j)}{n(j)}.\frac{2\pi c}{\rho}.\sum_{i=1}^{n(j)} \frac{p_{1,i}^2(j)}{\dot{Q}_w^2} \qquad (5)$$

In practical cases it will be sufficient to measure the transfer functions to just a small number n(j) of microphone positions on each S_j.

In ref. [5] test results have been presented for an internally excited steel box, constructed of 5 mm steel plates and profiles (1.6 m × 1.0 m × 0.5 m). This structure represented a 1:2 scale model of a shipboard diesel engine. The sound transfer was studied from an "engine room" to the water in a large laboratory tank (360 m^3). For two underwater positions the measured sound pressure levels agreed within 3 dB for 1/3-octave bands in the frequency range 200-4000 Hz (corresponding with 100-2000 Hz at full scale for the shipboard diesel engine). Recently additional experiments revealed the same accuracy for lower frequencies starting at 100 Hz (i.e. corresponding to 50 Hz for the full scale engine). The method also provides detailed information on the contributions of the individual partial areas.

Compared to the method discussed in 2.1, the attractive aspect is that the "source strength" measurement, i.e. of $\overline{I_n}(j)$, is relatively simple, even in cases with irregular surfaces. It was rather surprising that the method appeared quite accurate even at low frequencies where the wavelength in air becomes much larger than the dimensions of the machine. However, for strongly directional sources the method is expected to fail. Potential applications could be in aircraft and vehicle interiors, where the proposed method may quantify the contribution of specific parts of the boundary to the sound level at specific receiver points.

3 STRUCTURE-BORNE SOUND

3.1 Equivalent forces method

There are a number of instances in the literature where matrices of frequency response functions (FRF) are inverted, in order to determine force spectra. In several cases, the objective is to determine (indirectly) the forces that are acting during service operation of a machine, via measurements of responses. This is done in combination with a matrix of FRF's which have been determined separately with the machine switched off.

For path quantification purposes the principle is applied in a different manner. The force spectra to be determined will only be used in a mathematical prediction of responses due to a particular structural path [6, 7, 8 and recent unpublished work at TNO]. After a brief discussion of the method some results will be presented for engine noise transmission into a truck cabin via vibration isolators, for sound transfer to underwater from a propeller shaft via a journal bearing and for sound transfer to underwater via a cooling water pipe.

Method

1. Whilst the machine is running

(a) A series of m accelerations $\{a_i\}_{me}$ at various points and in various directions (positions "i") is measured. These positions should be chosen so that between them they represent the "typical" vibration transmitted along the path under study.

(b) Measure the total sound pressure $\{p_k\}_{tot}$ at (one or several) receiver positions "k".

2. With the machine switched off

(c) Choose another set of n positions "j" at which a set of external forces $\{F_j\}$ can be applied (one at a time). In principle these lie on the transfer path of interest between the "source" and the response positions "i". Determine the matrix of transfer accelerances $A_{ij} = a_i/F_j$ (if more convenient this can be done reciprocally).

(d) Determine the transfer functions from each of these force positions to the sound pressure at the receiver positions $H_{kj} = p_k/F_j$ (usually reciprocal measurements are most convenient).

3. Analytically

(e) Determine a set of equivalent forces $\{F_j\}_{eq}$ which between them would generate the same or closely similar responses $\{a_i\}_{eq}$ at the positions "i" as the original source does, $\{a\}_{me}$. This involves the inversion of the $m \times n$ matrix A_{ij} to solve:

$$\{a_i\}_{eq} = [A_{ij}]\{F_j\}_{eq} \approx \{a_i\}_{me} \tag{6}$$

(f) Calculate the radiated sound $\{p_k\}_{eq}$ which would be caused by this combination of equivalent forces:

$$\{p_k\}_{eq} = [H_{kj}]\{F_j\}_{eq} \tag{7}$$

This represents the radiated sound purely due to the path under study.

(g) Comparison of $\{p_k\}_{eq}$ with $\{p_k\}_{tot}$ indicates the relative importance of this path.

An important issue to be dealt with in implementing this method, is the solution of eq. (6) by matrix inversion. To improve the confidence, m is usually chosen to be greater than n. A least squares solution is sought using "pseudo-inversion". As a further refinement the "Singular Value Decomposition" method is used to control the condition of the matrix inversion. An error threshold for rejecting singular values can be estimated from the accelerances and the coherence functions for the corresponding accelerance measurements [9, 10].

Truck cabin noise

Figure 3 shows results from an experiment on a truck cabin [8]. The cabin was mounted on the chassis via four vibration isolators. Together with the adjacent structures the isolators at the front were totally different from those at the rear. Accelerations $\{a_i\}_{me}$ were measured at 12 positions during a road test at 50 km/h (3 positions for 3 directions close to each isolator). The sound pressures $\{p_k\}_{tot}$ were measured at two positions in the cabin. Accelerances and transfer functions H_{kj} were measured with the cabin lifted for 12 forces, three perpendicular forces (x, y, z) at each isolator attachment point (notice that the system used in this case was not "overdetermined"). Figure 3a shows the cabin sound pressure measured on the road and that calculated according to eq. (7) for 12 forces. Figure 3b shows the contribution of one of the equivalent forces (i.e. at one isolator position, for one excitation direction). The response due to the total of all forces shows that the isolator path is predominant at most frequencies. The individual force of figure 3b, however, is relatively unimportant. Study of the accelerance matrix revealed that the coupling between the isolators was weak in a large part of the frequency range. Therefore, using this procedure it was easily found which isolators and which vibration direction were primarily responsible for the interior sound levels.

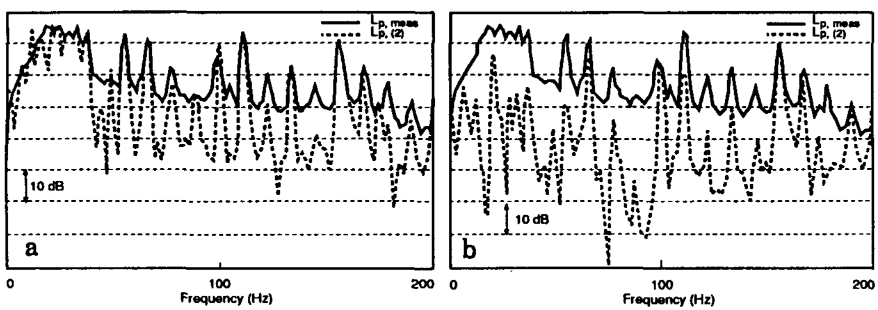

Figure 3 Equivalent forces method. (a) Sound pressure level in truck cabin during road test (meas.) and calculated (2) for 12 forces. (b) During road test (meas.) and calculated (2) for single isolator, single direction.

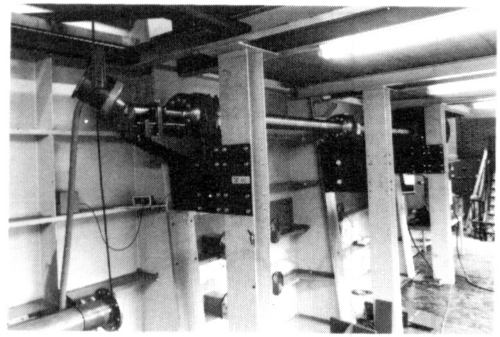

Figure 4 Laboratory shaft arrangement (scale 1:3).

Shaft bearing

Figure 4 shows a laboratory test arrangement of a 1:3 scale model of a propeller shaft which was connected to a water basin (360 m^3) via a journal bearing. As a preparation for shipboard use, this test set-up was used to validate the F_{eq}-method for quantifying the sound transfer via a bearing. The machinery excitation was simulated with a vibration exciter, both bending and longitudinal waves being excited in the shaft. The radiated underwater sound was measured at 4 hydrophone positions. The acceleration responses $\{a_i\}_{me}$ were measured at 11 positions on the seating and basin structure closely underneath the (centre) bearing. Six forces F_j were applied, using hammer impacts, to measure 11 × 6 accelerances. The excitation positions were on the bearing housing. They were chosen to cover more or less independent excitation by 6 orthogonal forces and torques. Rejecting singular values below a well defined error threshold, the number of singular values used in the pseudo-inversion of the 11 × 6 [A] matrix varied as function of frequency between 3 and 6. The frequency response functions H_{jk} were measured reciprocally using an underwater sound source with known volume acceleration $\dot{Q}_k$. The reciprocity relation is

$$[p_k / F_j] = [a_j / \dot{Q}_k] \tag{8}$$

For the frequency range of the measurements (200-2000 Hz) the agreement between measured sound levels and those calculated using eq. (7) were excellent at all individual hydrophone positions. Figure 5 shows a result averaged over the 4 hydrophone positions for two types of source spectra. Also shown is the result without rejection of singular values. Around 1 kHz this leads to larger errors. Detailed analysis of the results showed that with hindsight two equivalent orthogonal forces acting on the bearing in directions perpendicular to the shaft would have been sufficient to give an "accurate" reproduction of the underwater sound field. The appropriate phase relationship between these two forces is indispensable. The F_{eq} method provides this.

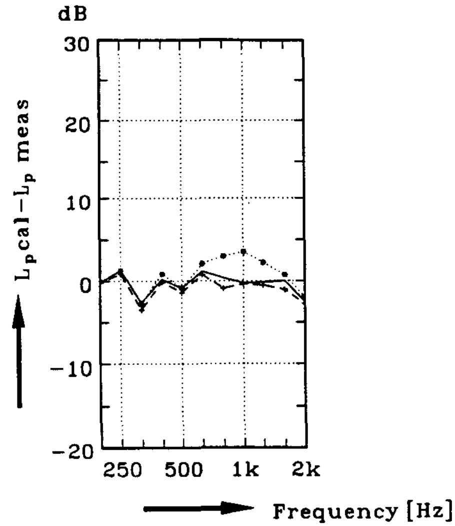

Figure 5 Equivalent forces method. Difference between calculated and measured underwater sound levels for two types of excitation signals. The dotted line represents the calculation with an ill-conditioned matrix inversion (none of six singular values set to zero).

Fluid filled pipe

The same type of experiment as described for the bearing was performed for a fluid-filled pipe system. Figure 6 shows a sketch of the non-planar system with a total length of 8 m. D is a flexible bellows; E, F and G are bracket connections to the water basin and close to H is a hull penetration. The machinery excitation was simulated by a vibration exciter at position B. Acceleration responses were measured along the pipe (14 in total: 5 at D; 3 at E; 2 at F; 1 at G and 3 at H. Six forces F_j were applied at flange D on the basin (i.e. lower) side of the bellows. Using different sets of response positions, different results were obtained for the equivalent forces and the underwater sound. For example using a set of 8 responses close to D and E yielded a set of equivalent forces which accurately reproduced the responses $\{L_{a,i}\}_{eq} \approx \{L_{a,i}\}_{me}$ at D, but at E, F, G and H the relative error in $\{L_{a,i}\}_{eq}$ was much greater. The "predicted" underwater sound also differed significantly from the measured levels, at low frequencies by up to 10 dB. Use of all 14 response positions might be expected to improve the situation, but it did not when using a straightforward procedure. The responses at D were still reproduced with a small relative error and these at E – H with a much larger error. This is because the accelerances and responses at F, G and H are much lower than those at D so that the same absolute error corresponds to different relative errors. Instead a "weighting" procedure was applied which was steered by the requirement to obtain a restored response $\{a_j\}_{eq}$ with the same *relative* inaccuracy at all 14 response positions [10, 11].

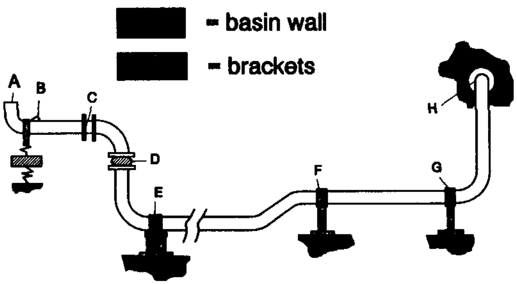

Figure 6 Sketch of fluid-filled pipe system.

Figure 7 shows a typical result of predicted and measured underwater sound at one hydrophone position from the "weighted" calculation.

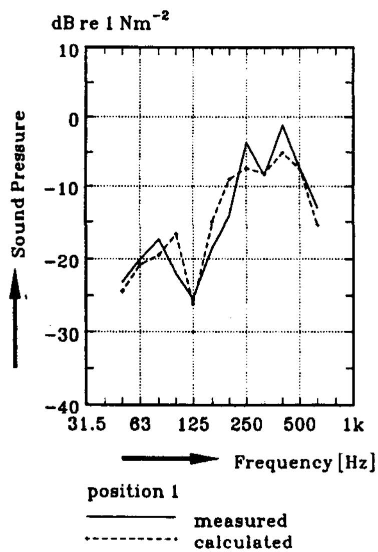

Figure 7 Equivalent forces method. Measured and calculated underwater sound levels caused by sound transfer via the pipe.

3.2 Equivalent intensity

Recently the measurement of structural and structural-acoustic intensity has been explored for sound path quantification on pipes and shafts [12]. A twofold problem has to be solved. The techniques for measuring energy flow in the different wave types should be developed and validated. But if that is successful, in addition a method is needed for relating the energy flow to radiated sound levels caused by such transfer paths. For the latter problem some preliminary investigations have been performed using a method which resembles the F_{eq} method of 3.1. The differences are as follows:

1. Whilst the engine is running m energy flow components $\{P_i\}_{me}$ are measured instead of $\{a_i\}_{me}$. These may be energy flows in different wave types at one cross-section, or energy flows of the same wave type at different cross-sections (across bends, flanges, etc.) or combinations of these possibilities.
2. Whilst the engine is switched off the matrix of "transfer functions" $W_{ij} = P_i/F_j^2$ is determined for n external forces F_j.
3. Analytically a set of $\{aF_j^2\}_{eq}$ is sought (with a = ± 1) which reproduces $\{P_i\}_{eq} \approx \{P_i\}_{me}$. This involves inversion of the m × n matrix $[W_{ij}]$ to solve:

$$\{P_i\}_{eq} = [W_{ij}]\{aF_j^2\}_{eq} \approx \{P_i\}_{me} \tag{9}$$

4. A different procedure has to be followed to calculate an error threshold for rejecting singular values.
5. The radiated sound caused by this combination of equivalent mean square forces is found from

$$\{p_k^2\}_{eq} = [H_{kj}^2]\{aF_j^2\}_{eq} \tag{10}$$

Comparison with F_{eq} method

In the F_{eq} method of 3.1 a set of equivalent forces (including phase) is sought that reproduces the measured acceleration field on a path. All processing is carried out in narrow bands (i.e. phase is retained), but the final results in terms of radiated sound may be converted to 1/3-octave band rms-values. This would reduce random errors and seems most appropriate for many applications. In the present method phase information is lost even in the narrow band case, because "mean" square quantities are used. However, this "error" is probably partially compensated for by the fact that energy flow components have directional information (i.e. positive or negative values). An important feature is that whilst the engine is running, the measurements of $\{P_i\}_{me}$ may reveal whether the supposed transmission direction of the path under study (i.e. from machine to "receiver") is predominant. This information is lacking in the F_{eq} method without additional measurements. Another potential advantage of the present method might be that data reduction can be enforced by converting P_i and W_{ij} data to 1/3-octave bands before the inversion is undertaken. This might improve the condition of the matrix inversion.

Preliminary investigations of the method have been performed on the same pipe system as used in 3.1. Five excitation forces were used in a way similar to that in 3.1. Six energy flow components were measured. At cross-sections between E and F, F and G, G and H (see fig. 6) the bending wave components in two perpendicular planes were measured. Figure 8 shows a result of predicted and measured underwater sound at one hydrophone position. The results seem less accurate than those in figure 7. Further study is needed on a number of aspects, i.e. number of independent forces F_j, statistical accuracy, selection of relevant wave types and the cause of some "negative" mean square underwater sound pressures predicted.

4 CONCLUSION

New substitution source methods for path quantification have become available for a number of applications. The equivalent power point source method seems very practical for airborne sound transfer. The equivalent force method seems most promising for structure-borne sound applications.

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