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THE MEASUREMENT OF TRANSFER FUNCTIONS FOR NOISE CONTROL APPLICATIONS.

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1. INTRODUCTION.

In principle, the prediction of airborne noise resulting from machine vibration requires knowledge of the source of the vibrations, of the structural transmission path between source and receiver and of the efficiency of the conversion of vibration to airborne noise. In practice, the transmission path and the conversion efficiency may be combined into one overall transfer function. Practical values of these parameters have been measured experimentally. Two real cases have also been investigated in the field. All of the measurements were carried out using an FFT analyser giving 400-point spectra over the range 0-500Hz.

2. BASIS OF THE METHOD.

Fig. 1(a) shows an impedance analogue of a complete system representing the transmission of vibrational energy from a machine to airborne noise in a nearby room. In general, the machine will have an output impedance, X_m, the antivibration mountings (avms) and the building structure will both consist of complicated equivalent arrangements of series and parallel elements. The element representing the building structure will, additionally, contain the component representing the radiation of the sound as airborne noise.

In practice, this complete equivalent circuit may be considerably simplified. Firstly, there is no need to consider the details of the 'structure' block at all because measurements may be carried out which characterise both the transfer function for vibration velocity to sound level and the input impedance. Secondly, if the 'anti-vibration' system can be represented by a single perfect compliance which has an impedance very much less than any other impedance in the system (particularly, less than the machine output impedance) then the avm block can be replaced by a single element and the machine impedance neglected. Fig. 1(b) shows the resulting, simplified circuit.

If T is the transfer function from velocity to sound level,
d is the machine displacment at its foot,
I is the structure impedance,
S is the stiffness of the avm system,

then the predicted sound pressure level, L, is given by:

$$L = d \, . \, S \, / I \, . \, T$$

The calculation is carried out for all of the frequencies in the 400-point spectra. The resultant narrow-band prediction may be converted to 1/3 rd octave by summing the contributions in each idealised frequency band.

3. PRACTICAL CONSIDERATIONS.

Although the principle of the calculation is straightforward, there may be practical difficulties in the evaluation of the four parameters required. The transfer function is most conveniently measured using a two-channel analyser with one accelerometer and one microphone. This will give the (complex) transfer function directly, provided that sufficient signal–noise can be obtained. In most cases, this is likely to be the most difficult aspect of the method. It is necessary to achieve, in all frequency bands of interest, a receive–area signal which is usefully above the existing background noise level. Generally, a source of vibration is required which will generate large forces at single frequencies or over a band of frequencies. Electromagnetic shakers or rotating–masses are feasible but are not usually portable enough for field measurements. The most convenient source has been found to be a hammer (usually, a large sledgehammer). This can be made to generate very large forces but the spectrum needs to be controlled. On a hard surface, a hammer will produce a very short impulse with most of the energy concentrated a high frequencies. By interposing suitable resilient pads, the duration may be extended and the peak force reduced, thereby producing more energy at lower frequencies. Some damping of the pad is desirable to provide a reasonably smooth spectrum. About 100mm thickness of expanded polystyrene packaging or 50mm of carpet tile have been found to be reasonable, although the former usually survives only for a few measurements. This method shares the common disadvantage of all impulse measurements, that is, a large peak-to-mean ratio which places serious additional demands on the instrumentation dynamic range. The use of analysers with more than 12–bit digital coding is advantageous.

The machine's dynamic displacement is not difficult to measure but it does require access to the actual machine, or a similar example. To carry out the measurement, the machine must be functioning under normal operating conditions and supported on reasonably compliant mountings. Under these conditions, the vibration of the machine mounting points will be representative and will be essentially unaffected by the support structure. On the assumption that the structural impedance of the machine is much greater that of the supports (as is also assumed for the calculation), the machine will be operating effectively in 'free space'. The displacement at the mounting points is simply the acceleration integrated twice with respect to frequency, that is, divided by $(j\omega)^2$.

The structure impedance can be easily measured using the two–channel analyser with a combined force transducer/ accelerometer. Commonly available impedance transducers are generally too delicate for the measurement of building structure impedances. (It will be seen later that this impedance measurement can, in any case, usually be eliminated from the calculation.)

The static stiffness of the avm system can be obtained from the manufacturer's data. However, it is necessary to use the dynamic impedance, which may be very different and which may also be a function of frequency. In avms with significant modal behaviour (most of those based on metal resilient elements), it is necessary to allow for the very large changes in dynamic impedance at some frequencies. With elastomeric mounts or with metal mounts below the frequency of the first mode, a single figure for the impedance is usually a sufficient approximation. The most convenient ways of quantifying the avm impedance are either to measure the input impedance at one end with the other end on a inert support or to measure the transfer function from one end to the other with a small inert load [1,2,4,5].

When using a hammer as a source of excitation for the velocity/spl transfer function, the measurements and calculations can be simplified if the hammer is calibrated as a force transducer. This was done for most of the practical work described in this paper. A large sledgehammer (5 kg) had a small accelerometer attached to one of its faces. The assembly was calibrated, using a normal force transducer, and found to behave as a reasonable force transducer, up to at least 1 kHz. There were minor effects from transverse vibrations of the wooden handle and it was necessary to allow the actual impact to occur freely, rather than under the positive control of the handle as would be more usual in the use of a sledgehammer. These effects were small in the context of the other uncertainties. The equation for sound level can be simplified by combining the velocity and impedance terms:

$$L = d \,.\, S \,/\, T'$$

where T' is the transfer function from force to sound level,

4. EXPERIMENTAL VERIFICATION.

The measurements outlined above were carried out in a laboratory situation and the predicted noise levels compared with measured values. The source was simulated by mounting a large electromagnetic shaker on a set of suitable commercial vibration isolators. The 'machine' displacement was measured directly with an accelerometer. The transfer function and 'foundation' impedance were measured together using the calibrated force generator (hammer) on the mounting points and a microphone in the receiving area. The avm stiffness was deduced from the system fundamental resonant frequency. Fig. 2 shows one set of results.

5. FIELD MEASUREMENTS.

The practical problem investigated was that of the installation of building services plant very close to a broadcasting studio. In this case, the building exterior formed the upper part of the studio. On an adjacent roof, alongside and structurally connected to the studio wall at a distance of less than 0.5m was the proposed location of a pair of 300kW chillers, each containing a 75kW electric motor and multi-cylinder reciprocating pump, together with four powerful (12kW), high-speed fans for the cooling air.

The three factors required for the prediction of potential noise levels were reasonably easily measurable - a nominally identical unit was already working nearby and the proposed structural mounting points for the plant already existed. The measurement of the plant vibration displacement presented no problems. However, those of the transfer function and impedance were more difficult. At the time, the calibrated force generator was not available. The measurement was therefore carried out in two steps, using the sledgehammer to measure the velocity-to-sound level transfer function and a small force transducer to measure the mounting-point impedance. The tailoring of the excitation spectrum to give sufficient low frequency energy was achieved by the selection of different types of resilient pads between the hammer and the hard concrete upstands.

Fig. 3,4,5 show the results for velocity to spl transfer function, structure mounting point impedance and machine mounting point displacement respectively. Fig. 6 shows three of the predicted spl spectra calculated from the measured data for different combinations of mounting point and microphone positions. Also shown is the design criterion for backgound noise level in that studio environment.

Following these predictionsof potentially high levels of interference in the studio, the proposed installation was modified to include additional stages of vibration isolation. This greatly complicated the prediction, to an extent where the uncharacterised, additional components rendered further prediction impossible. Measurements were carried out after the completion of the installation to enable a retrospective 'prediction'. Fig. 7 shows the result, compared with the original prediction for the same combination of positions. The purpose of this retrospective work was to obtain information about the additional stages of vibration isolation, to test the principles of the prediction method in the field and to allow the final result to be interpreted more meaningfully.

6. RESULTS, DISCUSSION AND CONCLUSIONS.

Fig. 8 shows (a) the measured studio noise levels from all sources, including the plant, (b) the structure-borne noise levels predicted as described above and (c) the calculated studio noise levels based on the measured airborne noise levels and airborne sound level difference.

Comparing first the 'predictions' of the noise via the two paths. At frequencies of 63, 80 and 100 Hz there is a slight dominance of the structure-borne noise of between 2 and 14 dB. At 125 and 160 Hz, the two predicted contributions are about equal. Above 160 Hz, the predicted structure-borne noise falls very sharply to become insignificant in comparison with the airborne component.

Comparing both predictions with the measured studio noise levels, the agreement is sporadic at best. At low frequencies, where the noise was predicted to be structure-borne, the difference between the prediction and the result takes a range of -2.5 to11 dB and an average value of 4 dB. At higher frequencies, where the airborne component was predicted to be dominant, there is more correlation but a probably significant, consistent error of 4.1 dB average. This error was most probably due to the change in the acoustic fields as a result of the installation of the plant.

All of the field measurements are based on very few sample positions because of the short timescale allowed for the investigation and the awkward situations in which the measurements were made. They are therefore subject to large statistical errors. In the much-better controlled environment of the laboratory, the method shows reasonable agreement, although with errors of up to about 10dB.

The overall conclusion which may be reached is that the measurements are practicable in the field and that the prediction method is accurate enough to form the basis of informed design decisions. Potential errors exist in the difficulty of generating measurable signal levels, especially in the presence of existing noise and in the possibility of other coupling paths such as pipework, conduits, etc. The performance of real vibration isolation systems, in particular, modal behaviour, is also a potential source of serious errors.

7. REFERENCES.

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8. ACKNOWLEDGEMENTS.

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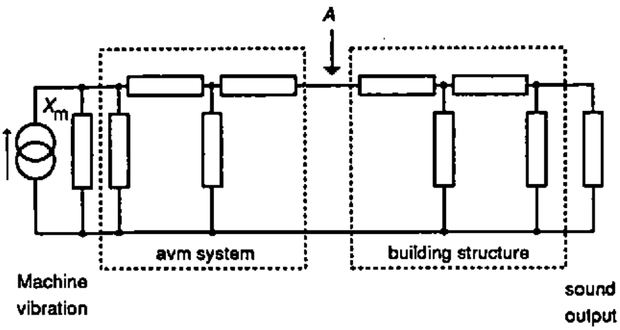

Fig. 1(a) 'Complete' system equivalent circuit.

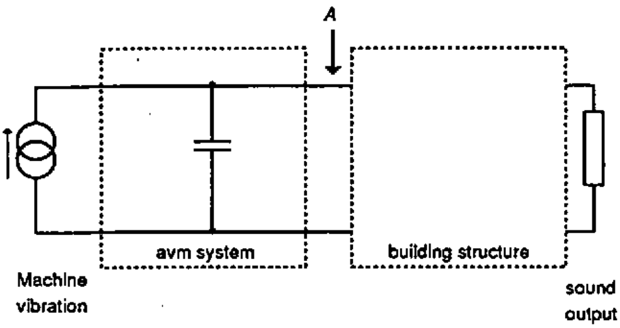

Fig. 1(b) Simplified system equivalent circuit.

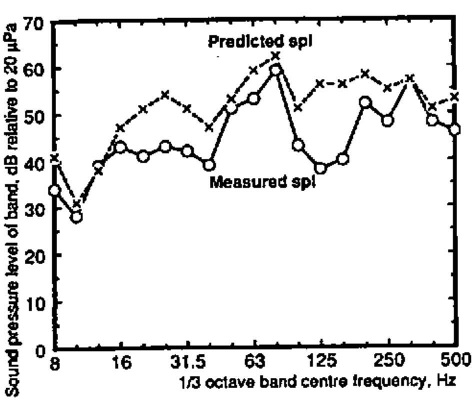

Fig. 2 Experimental results.

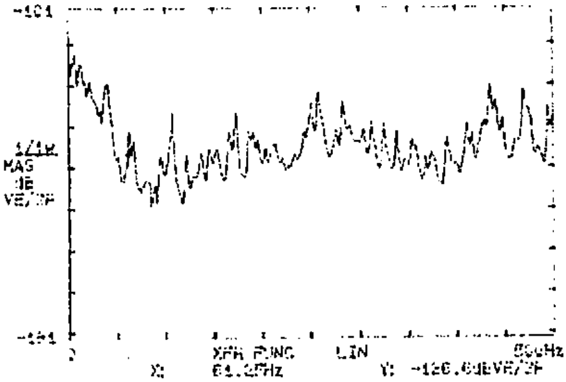

Fig. 3 Sample of transfer function (velocity to spl).

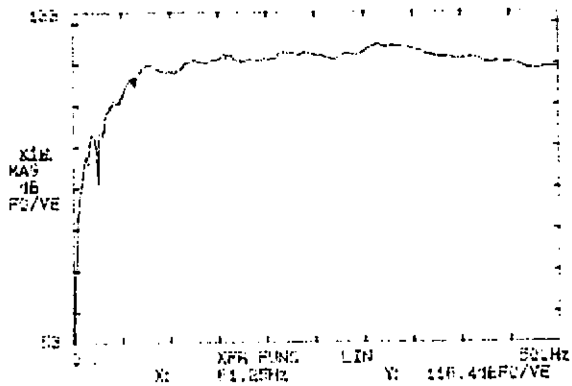

Fig. 4 Sample of mounting point impedance.

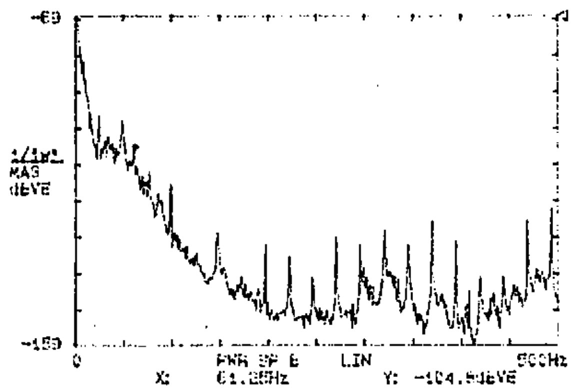

Fig. 5 Sample of plant vibration (displacement).

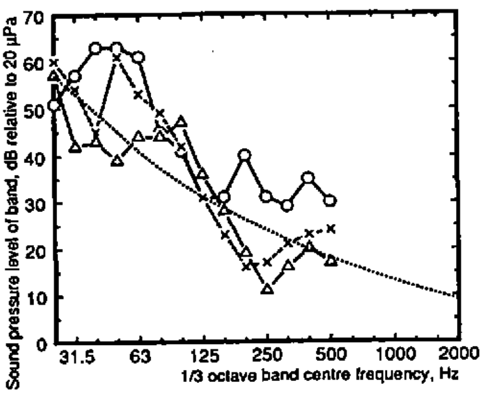

Fig. 6 Examples of original predictions.

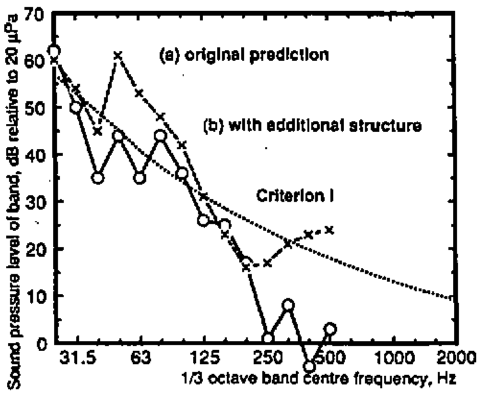

Fig. 7 Predictions of structure-borne noise levels.

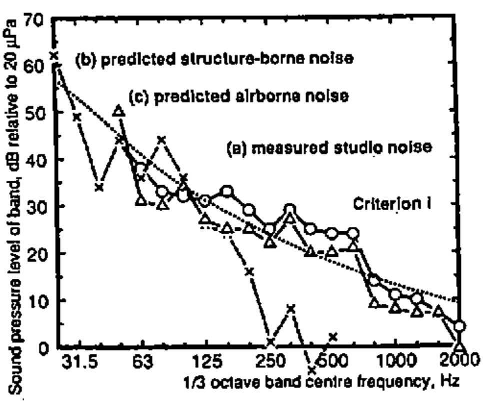

Fig. 8 Predicted and measured noise levels.

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