inter•noise 96
25th Anniversary Congress – Liverpool

INCE: 76.9

NOISE LEVEL PREDICTIONS FOR A LARGE, RECIPROCATING IC ENGINE IN A BROADCASTING STUDO CENTRE

R Walker

BBC Research and Development Department, Tadworth, UK

1. INTRODUCTION

Large machines are, self-evidently, potential sources of severe noise and vibration. In order to determine whether or not it would be acoustically feasible to install a CHP package in a broadcasting studio centre, measurements and calculations were carried out to predict the likely noise levels in studios and other technical areas.

This Paper is concerned only with the airborne noise in a remote area arising from the structure-borne plant vibration, transmitted via the vibration isolation system. Other studies were carried out on the airborne noise components.

2. PREDICTION PRINCIPLES

Ideally, a machine which is mounted on a vibration isolation system and causing airborne noise in some distant space can be represented by three main elements, as illustrated in Fig. 1. These elements are the motion at the machine mounting points ('feet'), the transfer function of the vibration isolation system and the transfer function from the force imposed on the supporting structure to the re-radiated sound level in the remote area [1,2].

A number of simplifying assumptions can be made. The first is that the movements of the machine are unaffected by their means of support. This will be a good approximation when the machine is very stiff and/or massive in comparison with the vibration isolation system. The second is that the vibration isolation system can be represented by a relatively simple stiffness. This is true for relatively light mountings operating well away from the system fundamental resonant frequency and in a regime where the mounting has no internal modes. The third assumption is made that the underlying supporting building structure (foundation) is relatively stiff and/or massive. This means that most of the machine displacement appears as deflection of the vibration isolation system.

The prediction method relies on the separate identification of the three elements in the overall system -

a. The vibrations of the machine can be easily measured if a closely

similar machine can be found which is already in operation.
b. The intended or idealised vibration isolation system may be represented by a calculated stiffness.
c. The transfer of vibration energy from the force exerted by the machine to airborne sound in the remote area can be measured directly, by applying known dynamic forces to the intended supporting structure and measuring the resulting airborne noise levels in the receiving area.

All of these measurements and assumptions can be combined to produce a prediction for the airborne sound level which would occur in the target area if the machine were to be installed.

The machine vibrations, measured using an accelerometer, can be converted to displacement magnitude by dividing by ω^2 (ω is the frequency in radians/s). That displacement is assumed to distort the vibration isolation system directly, causing a force to be imposed on the foundation through the stiffness of the vibration isolation system. The effective force is the product of the displacement and the isolation system stiffness. The result can then be combined with the measured force-sound level transfer function to produce the predicted sound level in the target area.

3. MEASUREMENTS AND CALCULATIONS.

The vibration measurement equipment consisted of B&K type 4393 and 4379 accelerometers. Signal processing was by a MLSSA analyser. The electromagnetic shaker used to measure vibration-sound level transfer functions was a Pye-Ling 409. It was spring-mounted in a frame to provide an output force with respect to the inertial reference frame.

4. RESULTS.

The force to sound level transfer function was measured from the plant room floor at the proposed location of the CHP set to a microphone in a studio. The floor slab was, as expected, very stiff and the achievable noise level in the studio was quite imperceptible, as expected. Signal-noise ratio improvements had to be applied to reject extraneous noise [3,4]. It required a signal-noise ratio improvement of 33 dB. Fig. 2 shows a force to sound level transfer function obtained.

Measurements of machine vibrations were carried out an existing installation. Fig. 3 shows the product of the force-sound level transfer function and one of the acceleration spectra. The plot includes conversion from acceleration to displacement and the scaling term for a total static stiffness of 21×10^6 N/m (+146dB re. 1N/m).

This entire process was repeated for five engine frame accelerometer positions. The five predicted studio sound levels after conversion to the more familiar one-third octaves. are shown in Fig. 4, together with the drama studio background noise criterion.

5. SUMMARY AND AND CONCLUSIONS.

The results show that the predicted worst-case levels were about 5 dB below the background noise criterion. This shows that, in principle, the structure-borne vibration could be adequately isolated. However, the margin was very small and there were some other factors which had to be considered:-

a) The calculations were based on perfect vibration isolators, perfectly installed. No allowances were made for isolator imperfections or installation shortfalls.

b) All ancillary supports were assumed to be perfect. No allowance was been made for extraneous flanking paths via pipework, including the inevitably substantial exhaust system.

c) No account has been taken of the static to dynamic ratio of the vibration isolators. This would add between 3 and 6 dB to the predicted levels, depending on the choice of isolator material.

It has been shown that the structure-borne vibration energy from the CHP plant could be adequately controlled, in principle. However, even based on a theoretically perfect simple model and perfect installation, the margin for error would be small. It is not thought that such an ideal result could be achieved in practice. Subsequent re-calculations, based on more complex isolator models and different values for engine masses and static deflections, as well as some additional data, suggested that the theoretical worst-case noise margin could be about 6 - 7 dB. However, this prediction was still based on a theoretically perfect installation and would, in practice, be a very small safety margin.

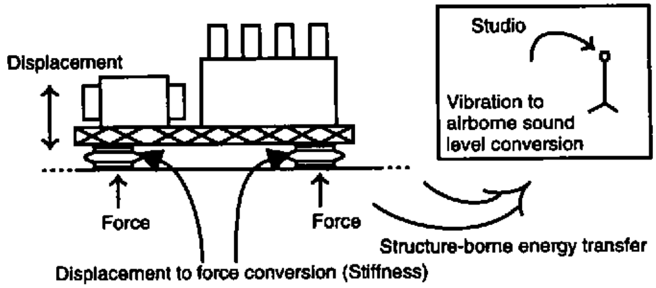
References

1. Walker, R., Mathers, C.D. and Fletcher, J.A. The prediction of noise from machine vibration. 13th ICA, Belgrade,1989, Paper 3.8.
2. Walker, R., Mathers, C.D. and Fletcher, J.A Vibration isolation - the prediction of regenerated levels, a case study. Proc. I.O.A.,10, Part 8,1988, p137.
3. Walker, R, Maximal-length sequence methods : acoustic measurements in the presence of high levels of extraneous noise. BBC RD Report No. RD 1994/5
4. Walker, R. The measurement of large values of airborne sound insulation. BBC RD Report 1986/8.

Acknowledments

The assistance of Ray Turford and Mike Stokes of BBC Environmental Services for making arrangements for measurements in BBC premises, and of Mark Anderson of Energetica and Keld Skerbek of Ulstein-Bergen for arranging the engine measurement session are gratefully recognised. The Paper is published by permission of the British Broadcasting Corporation.

Fig. 1 Sketch showing principles of noise prediction calculation.



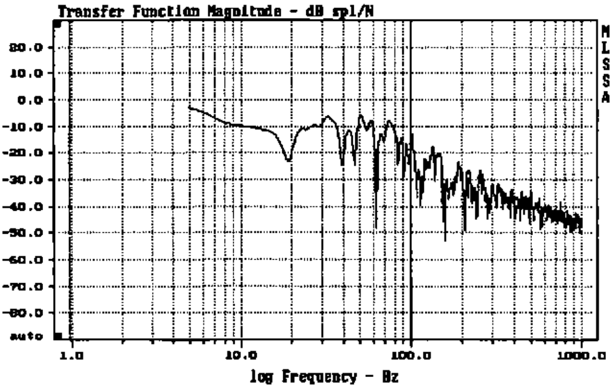


Fig. 2.

Force to spl transfer function, measured from proposed plant location to nearby studio. Units are dB relative to 20μPa/N.

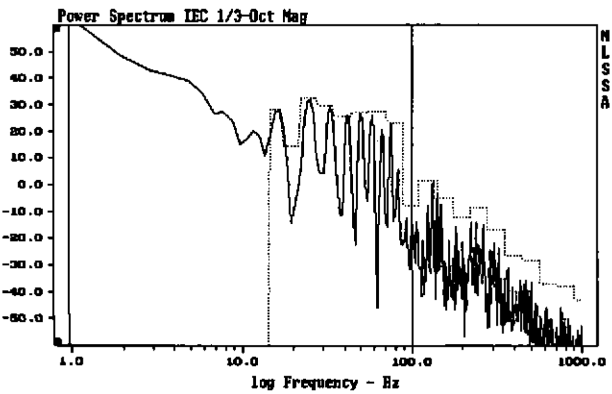


Fig. 3.

Predicted studio noise levels, based on accelerometer position 1, with one-third octave conversion.

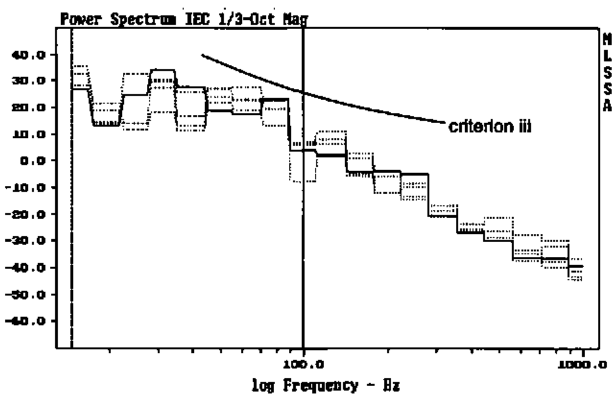


Fig. 4.

One-third octave studio noise level predictions, calculated for accelerometer positions 1 - 5, with part of drama studio background noise criterion.