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ESTIMATION OF RADIATED NOISE USING STRUCTURAL RESPONSE METHODS

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INTRODUCTION

The noise radiated from a structure can be estimated by equating the input and output energies. This is very useful in impact situations, because the noise energy radiated per blow can be expressed in terms of the structure parameters, like the structure radiation efficiency, the structural loss factor, the shaping of the force pulse and the response of the structure at the point of impact [1]. This dependence of the noise radiated on the structural response is important for noise control. Large reductions in the noise energy radiated per impact can be obtained by modifications of the structure to give a lower response. The advantage of this method of noise control over other methods is that the shape of the impact is not altered, that is if the structure is part of a machine, and the impact is due to the machine operation, the latter is not slowed down. Also, in some circumstances a further increase in the structural damping is not possible.

ENERGY BALANCE

The energy that escapes into the structure can either be radiated as noise or dissipated as heat due to the structural damping, i.e.,

$$E_{escape} = E_{rad} + E_{loss\ factor} \tag{1}$$

where E_{escape} is the total energy that escapes into the structure. The noise energy radiated per impact is given by:

$$E_{rad} = \int_0^T \rho_o c_o A\sigma_{rad} \langle \overline{V(t)^2} \rangle dt \tag{2}$$

or

$$E_{rad} = \int_{-\infty}^{\infty} \rho_o c_o A\sigma_{rad} \langle |V(f)|^2 \rangle \, df \tag{3}$$

where $\rho_o c_o$ is the acoustic impedance of the air, σ_{rad} the radiation efficiency and A the surface area of the structure. $V(t)$ and $V(f)$ are the normal velocity of vibration in the time and frequency domain, respectively, and T is the time of ringing of the structure. $< >$ denotes spatial averaging. The energy dissipated due to the structural damping is given in the frequency domain by:

$$E_{loss\ factor} = \int_{-\infty}^{\infty} 2\pi f \eta_s \rho A \ <|V(f)|^2> df \tag{4}$$

The noise energy radiated can be obtained by combining equations (1), (3) and (4)

$$E_{rad} = E_{escape} \cdot \frac{\sigma_{rad}}{\sigma_{rad} + \frac{2\pi f \eta_s \rho}{\rho_o c_o}} \tag{5}$$

E_{escape} is given by the integral over the impact duration of the impact force and the velocity at the point of impact in the same direction as the force, i.e.,

$$E_{escape} = \int_0^{\tau} f(t) v(t) dt \tag{6}$$

which can also be expressed in the frequency domain. The escape energy in a bandwidth Δf, with central frequency f, is thus given by:

$$E_{escape}(f, \Delta f) = |\dot{F}(f)|^2 \ Im[H(f)] \ \frac{\Delta f}{2\pi f} \tag{7}$$

where $|\dot{F}(f)|^2$ is the force derivative spectrum and $Im[H(f)]$ is the structure response, defined by the ratio of the velocity normal to the surface at the point of impact to the impact force derivative, i.e., $[V(f)/\dot{F}(f)]$. Thus the noise energy radiated from a structure per impact can be expressed in terms of the structure parameters, and the level of noise radiated can be controlled by reducing the structural response level.

The physical interpretation is that by reducing the level of the structure response, less energy escapes into the structure. In impulsive situations, more energy is left in the impacting hammer, which may result in bouncing of the hammer. If multiple impacts result, then the same amount of energy will be transferred. The only difference being that the transfer is over a number of impacts rather than a single impact. If the noise energy radiated is expressed as an average level over a period of time, then the same level of noise will be radiated.

EXPERIMENTAL INVESTIGATION

This concept to estimate the radiated noise is investigated in the case of a plate structure. The noise radiated from the plate is controlled by putting lumped masses to reduce the level of the response. The changes in the response can be obtained by structural modifications and these can be of either structural or material changes, like a change in the structure cross-section; resilient interlayers; etc. At medium and high frequencies, lumped masses are very effective to restrict the flow of vibrational energy. Thus for the tests on the plate the response is altered by having a block of mass fixed directly onto the plate at the point of impact. To further reduce the structure response, the attachment of the mass onto the plate is via a resilient interlayer. The noise reduction obtained for the same impact force, with a lower response is very high [2], especially at high frequencies. Also, good agreement is obtained between measured and estimated noise levels. From this comparison of the estimated and measured results, it can be concluded that using this method to estimate the noise energy radiated from a structure excited by an impact will give good results.

Most structures have a high modal density in the frequency range of high noise levels, thus an average value for the structural response can be used. In the case of beam and plate-like structures, this frequency averaged response is the same as that of a similar structure but of infinite extent, that is the structure has no resonance. This condition is approached for highly damped structures. In this case the bandwidth associated with each mode would be larger than the bandwidth separating successive modes, or, the stress waves set up in the structure are sufficiently attenuated before they reach the boundary of the structure, such that the reflections are very weak. This is an added advantage of using this average frequency response to estimate the broadband noise radiated from a structure, because the exact shape of the response is not necessary. However, for narrow band analysis the same approach can be used provided that the exact details of the response with frequency are known.

LOW FREQUENCY RESPONSE

The estimation of the structure response is therefore very important, to estimate the radiated noise. A frequency averaged level of the response is usually acceptable. However, this only applies when the modal density is high. In some cases, the first ringing frequency of the structure is within the frequency range of interest, and therefore an average level of the response associated with the modal density cannot be obtained. Therefore, an alternative estimate is needed. From the summation of modes solution to the response of a structure, it can be shown that [3] the response below the first fundamental frequency is dependent on the static stiffness of the structure, and is given by

$$\mathrm{Im}\left[H(f)\right] = \frac{\eta_s}{\pi^2 K} \tag{8}$$

where K is the static stiffness and η_s the structural loss factor associated with the first mode.

An example where the first ringing frequency is in the range 1-2 kHz is the ringing of bottles in bottling lines. Bottles have their fundamental frequency at about 2 kHz, therefore it is important in the estimation of noise from bottle clashing to estimate the response below this frequency. Experiments on the imaginary part of the imaginary part of the response of a bottle excited on the side show that there is good agreement between the estimate using equation (8) and the measured response at low frequencies. This also shows in the comparison of estimated and measured radiated noise [3].

Above the fundamental frequency, the modal density is very low and one would expect that the estimated noise will be different from the measured noise in broad bands. However, although there is a slight fluctuation, the total noise radiated can be estimated to within ±2 dB.

CONCLUSION

The two main conclusions are first that the noise from a structure can be estimated using an energy accountancy concept, where the noise is expressed in terms of the structure parameters. The second is that the noise energy radiated from the structure can be controlled very effectively by modifications in the structure to reduce the level of the response. This latter method of noise control is useful when other methods are not used because of other consequences on the operation of the structure.

REFERENCES

1. E.J. Richards (1981) Journal of Sound and Vibration 76, 187-232. On the prediction of impact noise: III. Energy accountancy in industrial machines.

2. J.M. Cuschieri and E.J. Richards (1983) Journal of Sound and Vibration 86, 319-342. On the prediction of impact noise: IV. Estimation of noise energy radiated by impact excitation of a structure.

3. E.J. Richards, A. Lenzi and J.M. Cuschieri (1983) Journal of Sound and Vibration (to be published). On the prediction of impact noise: VI. Distribution of acceleration noise with frequency with application to bottle impacts.