Proceedings of The Institute of Acoustics

IMPACT NOISE IN MACHINERY: ITS PREDICTION PART II: RINGING NOISE

M E WESTCOTT and R K JEYAPALAN

ISVR,
SOUTHAMPTON UNIVERSITY

Introduction

It was shown in Part I that for impacts of reasonably compact bodies, the radiated noise energy due to longitudinal vibrations was of comparable magnitude to that of acceleration noise, but if flexural vibrations occurred, then the radiated energy from these could be orders of magnitude greater. The vibrational energy escaping from the work area will be dissipated both as radiated sound and in structural damping in the ratio of acoustic loss factor η_{rad} to the structural loss factor η_s. Consider first the steady state situation for continuous excitation.

$$W_{input} = W_{rad} + W_{structural}$$

It is usual to express the radiated sound power by

$W_{rad} = \rho c S \langle \overline{v^2} \rangle \sigma_{rad}$ $\quad$ S = surface area, $\langle \overline{v^2} \rangle$ = mean square velocity, σ_{rad} = radiation efficiency

and the power absorbed by the structure (for simple cases) by

$W_{struc} = \omega \eta_s \rho_s S \langle \overline{v^2} \rangle$ $\quad$ where η_s = structural loss factor, ρ_s = surface density

and therefore $W_{in} = \rho c S \langle \overline{v^2} \rangle (\sigma_{rad} + \rho_s \omega \eta_s / \rho_0 c)$

To predict the radiated sound power W_{rad} it is therefore necessary to know either

(i) W_{in} and the ratio of σ_{rad} to $\rho_s \omega \eta_s / \rho c$, or

(ii) S, $\langle \overline{v^2} \rangle$ and σ_{rad} for each surface.

Both the power escaping from the work process and η_s in the case of a complex structure are difficult parameters to establish, so the first approach is unlikely to be useful except to indicate that noise reduction must consist of reducing W_{in} and minimising the ratio $\sigma_r / \rho_s \omega \eta_s$. If, however, the vibration levels are known, either from finite element methods or from measurements on similar or previous versions of the machine, then the second approach is more useful. This requires a knowledge of σ_r and the prediction of this for simple components is the subject of the remainder of this paper. In the case of transient excitation, it must be assumed that σ_r does not differ to that for the continuous excitation described above.

A series of charts of σ_r have therefore been prepared with the practising machinery designer in mind. Radiation efficiency is a function of wavenumber and dimensions rather than frequency but since the results must usually be A weighted, the charts are presented in terms of frequency. Some of what follows has been presented elsewhere and the results are scattered throughout the literature but not usually in this context.

IMPACT NOISE IN MACHINERY: ITS PREDICTION
PART II: RINGING NOISE

Measurements of radiation efficiency

The value of the charts presented below depends on the extent to which they apply to real cases. For some of these, measured values are shown, derived using the expression

$$\sigma_{rad} = \frac{W_{rad}}{\rho c S \langle \overline{v^2} \rangle}$$

W_{rad} was determined using the standard method for sound power measurement in a reverberent chamber while simultaneous measurements of $\langle \overline{v^2} \rangle$ were made over the vibrating surface.

Radiation Efficiency for whole body vibrations

(i) Pulsating bodies (Fig 1). σ_{rad} for pulsating spheres [1] is given by

$$\sigma_{rad} = \frac{(ka)^2}{1+(ka)^2} \quad \text{where } k = \frac{\omega}{c},\ a = \text{radius} \qquad \text{ie. } \sigma_{rad} = 1 \text{ when ka large},\ \approx (ka)^2 \text{ when ka small}$$

(ii) Oscillating bodies (Fig 2). σ_{rad} for oscillating spheres [1] is given by

$$\sigma_{rad} = \frac{(ka)^4}{4+(ka)^4} \qquad \text{ie. } 1 \text{ when ka large},\ \frac{(ka)^4}{4} \text{ when ka small}$$

(iii) Vibrating segments of spheres and cylinders. A more common occurrence is the vibration of panels, covers etc. which form part of a larger surface. A better representation may be obtained by considering radiation from a pulsating segment in an otherwise rigid sphere or cylinder. This has been examined and the solution is not simple [2]. The value of σ_{rad} varies greatly with details of the configuration.

There is no reason to assume that σ_r differs greatly from the spherical case for compact non-spherical bodies.

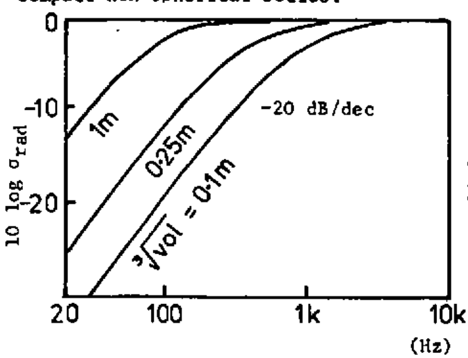

Fig 1: σ_r for pulsating spheres

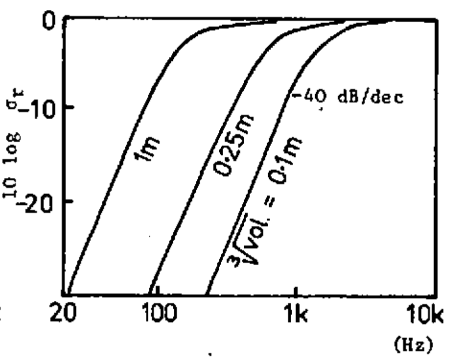

Fig 2: σ_{rad} for oscillating spheres

IMPACT NOISE IN MACHINERY: ITS PREDICTION
PART II: RINGING NOISE

Radiation Efficiency for Platelike Structures

For the infinite undamped surface supporting progressive waves it has been shown that no sound can be radiated until the progressive wavespeed is greater than that of sound in air [3]. At speeds greater than this, σ_{rad} becomes high (Fig 3). Plates and panels are not infinite and radiation can occur below the coincidence frequency for 2 reasons (i) Radiation from uncancelled edge and corner modes (ii) incomplete cancellation due to damping.

Radiation from baffled panels has been considered before [4] so only unbaffled, free-free plates have been measured. Fig 4 illustrates the effect of size and shape for baffled panels. Fig 5 shows measured values of σ_{rad} for an unbaffled free panel compared to theoretical values for a baffled simply supported panel. Agreement is good above the coincidence frequency.

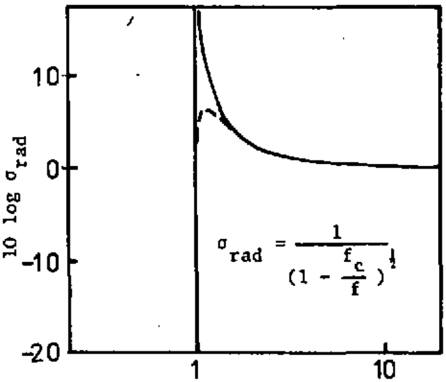

Fig 3: σ_r vs f_c/f for infinite plates

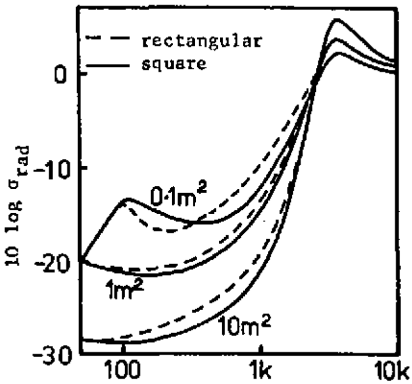

Fig 4: σ_r of 3.17mm thick square and rectangular plates (5:1)

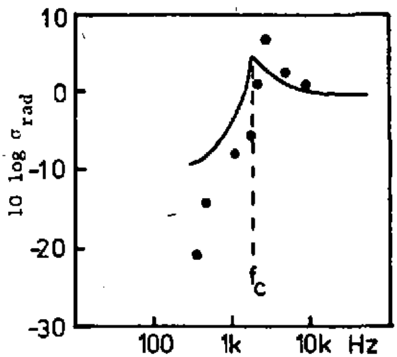

Fig 5: measured σ_{rad} of freely suspended plate 0.8m x 0.3m

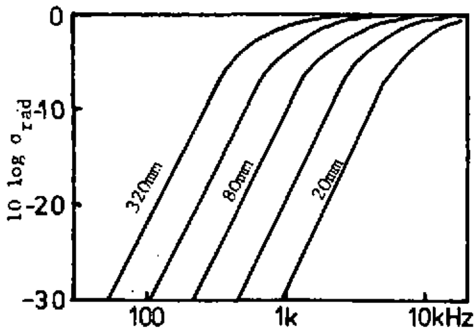

Fig 6: σ_r of long cylinders in flexural vibration

IMPACT NOISE IN MACHINERY: ITS PREDICTION
PART II: RINGING NOISE

The Radiation Efficiency of Circular Rods and Pipes

(i) Pulsating Pipes [2]

$$\sigma_{rad} = \frac{2}{\pi(ka)|H_1(ka)|^2}$$

where H_1 = Hankel function of order 1

ie. $\sigma_{rad} = \frac{\pi ka}{2}$ when $ka < \frac{2}{\pi}$; = 1 when $ka > \frac{2}{\pi}$

(ii) Oscillating Rods and Pipes [2]

$$\sigma_{rad} = \frac{2}{\pi(ka)|\overline{H}_1(ka)|^2}$$

where $\overline{H}_1$ = 1st derivative of H.

(iii) Circular Pipes and Rods in Bending

In this case the total radiation depends not only on cancellation around the cylinder but also along its length. ie, both on ka and the coincidence frequency. Fig 6 illustrates this for a number of long cylinders. The effect of finiteness of length is small above coincidence. Fig 7 shows some measured values for straight and bent rods.

The Radiation Efficiency of Beams of Complex Section (I beams, U beams)

No analytical methods exist by which the σ_{rad} of complex section can be predicted. The theory for circular beams has expressed in cylindrical co-ordinates and a 'master curve' drawn for beams of various width/depth ratio. σ_{rad} for an I beam vibrating in its various modes has been measured for comparison with cylindrical and elliptical models.

Conclusions

The value of the above approach is seen only when applied to a real machine, and an energy balance obtained. A comprehensive vibration survey of a small drop hammer has recently been completed from which the energy radiated from each component is being calculated for comparison with that measured using a microphone. The results, which it is hoped will indicate the major sources of radiation, will be presented.

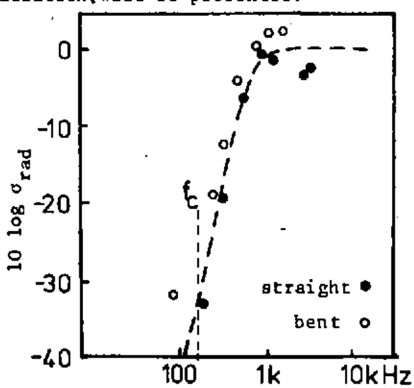

Fig 7: measured σ_{rad} for straight and bent cylindrical rods

REFERENCES

(1) E J RICHARDS and D J MEAD (Eds) 1968, Noise and Acoustic Fatigue in Aeronautics, Wiley & Sons.

(2) P M MORSE, 1948, Vibration and Sound, McGraw Hill.

(3) M HECKL, L CREMER and E E UNGAR, 1973 Structure Borne Sound, Springer, New York.

(4) L BERANEK (Ed) 1971, Noise and Vibration Control, McGraw Hill.