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THE SOUND POWER OUTPUT FROM INDUSTRIAL MACHINES

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Introduction

Noise arises from vibration of a volume occupying a small fraction of the whole of space. This volume can be an industrial machine vibrating either as a 'forced acceleration' motion arising from an impulsive load or from a quasi-steady 'ringing' vibration which will continue until the vibrational energy in it has been absorbed as heat, radiated as sound, or transferred to the floor (which will in turn vibrate). The vibrating volume can consist of a region of combustion or of jet mixing, or it can consist of stationary or turbulent air suddenly put into motion by a force applied to it (eg. fan). In every case, the noise is associated with the compression of the surrounding air associated with this vibration, and the relationships between noise, vibration and machinery energy is extremely tenuous, so much so that source acoustics has taken on a mystique all of its own, and machinery noise control experiments have taken on an ad-hoc nature which has in turn added to the nature of this mysterious 'know-how'.

The fraction of chemical energy going into noise in a smooth burning jet can be as little as 10^{-9}, or as much as 10^{-4} if unstable burning can be established. Aeroplane jet noise can be as little as 10^{-4} of the jet kinetic energy, and the fraction of the kinetic energy of a drop hammer going into noise can be as little as 10^{-5} or as much as 10^{-2} depending upon the way this energy can escape into ancillary structures capable of extensive 'ringing'.

At the same time, noise control, by virtue of the nature of the ear, need only be an 'order of magnitude' science, a halving of the emitted noise energy being only just discernable. Thus although accurate noise prediction of sound power output can only be achieved if the vibration of the relevant air volumes can be completely specified both in amplitude and in phase, the fact remains that rough but useful predictions can be made, in broad terms, once the nature of the acoustic parameters have been identified.

Impact Noise Parameters

Impact machines function by building up energy slowly and distributing it rapidly into the workpiece or throughout the machine. Thus, in a drop hammer or a coining press, the energy is contained in the hammer (tup) and is quickly transferred into the workpiece or into the machine structure. For soft blows, ninety-five per cent of the energy is absorbed in deforming the billet. For the final coining blows, ninety-five per cent of the energy is distributed into the structure as vibrational energy and into the ground. On a punch press, the structure is loaded relatively slowly and a rapid redistribution of energy follows fracture.

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The noise radiated from a drop hammer has a typical signature of the kind shown in Figure 1. The first double pulse is related to the forced deceleration arising from the tup coming to rest, the 'ringing' noise arising for the vibration which follows from that part of the stored vibrational energy which is dissipated acoustically as the structure vibrates. As the latter must depend upon how quickly the internal damping of the structure and the transmission into the floor can dissipate the vibrational energy of the machine, the first 'acceleration' noise and the subsequent 'ringing noise' must be treated as separate contributors to Leq or Leq(A) even though they are related in terms of vibration characteristics.

As the 'quasi-steady''ringing' noise dominates in the determination of Leq (though acceleration noise determines peaks) let us deal with the parameters involved in the energy balance accountancy associated with ringing.

As a large plane plate vibrating with a mean square velocity $\overline{v^2}$ radiates per second a noise power W_{rad} which is equal to $\rho_o cS\,\overline{v^2}$ (where ρ_o is the density of the surrounding air, c is the speed of sound in air and S is the plate surface area) we define a radiation efficiency 'σ_{rad}' for any other body such that, for the body, radiated sound power W_{rad} is $\sigma_{rad}.\rho_o cS\,\langle\overline{v_n^2}\rangle$ where $\overline{}$ represents an average with time of the square of the surface normal velocity v_n and $\langle\ \rangle$ represents the average over the surface. This efficiency becomes unity when the vibration is so fast that each part of the body can be considered as a separate and independent radiator. As energy can be put back into the body in the so called 'near field' which extends for a distance of about half a wavelength from any point of emission, σ_{rad} will depend upon the ratio of the circumferential length of a body to the wavelength, or in the case of a plate to the ratio of the speed of a flexural wave along the surface to the speed of sound in the adjacent air.

The fraction of internal energy in the structure dissipated into heat per radian of vibration is called the damping factor η_s and tends to be constant with frequency. The vibrational energy dissipated per second is therefore $2\pi f.\eta_s \times W_{vib}$ where W_{vib} is the internal vibrational energy at any moment. For a body of mass M, this energy is shared at any moment between elastic and kinetic energy so that it is usual to describe the average vibrational energy in a structural component as $M\langle\overline{v^2}\rangle$ or $\rho_s S\,\langle\overline{v^2}\rangle$ where S is the radiating surface area and ρ_s is the surface density ($\rho_m d$) where d is the average thickness of the component.

For any single impact, we therefore have two equations which describe the energy radiated as sound and the balance between it and the energy absorbed in structural damping, viz,

$$W_{rad} = \sigma_{rad} \times \rho_o cS \int_o^\infty \langle\overline{v^2}\rangle\, dt$$

and

$$W_{vib} = W_{input} - W_{work} - W_{ground} = W_{rad} + W_{struct}$$

$$= |\sigma_{rad}\rho_o cS + \rho_m S\eta_s d2\pi f| \int_o^\infty \langle\overline{v^2}\rangle\, dt$$

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and hence $$\frac{W_{rad}}{W_{vib}} = \frac{\sigma_{rad}}{(\sigma_{rad} + \frac{\rho_m}{\rho_o c} .d.2\pi f.\eta_s)}$$

If d, the typical plate thickness is measured in centimetres, c = 340m/sec, and $\frac{\rho_o}{\rho_m} = 1.55 \times 10^{-4}$ for steel, this can be written:

$$\frac{W_{rad}}{W_{vib}} = \frac{\sigma_{rad}}{\sigma_{rad} + 1.23.d.\eta_s.f}$$

If there is no structural damping in the machine or components, η_s is zero and $W_{rad} = W_{vib}$, (ie) the radiated sound energy is independent of the radiation efficiency of the surfaces. Under such circumstances, W_{rad} can only be reduced by isolating such components from the source of vibrational energy (ie) reducing W_{vib}. Examples which come to mind are gearbox casings, flywheels, some guard-rails, hammer anvils, machines mounted on girders and to a greater extent than is generally realised concrete floors themselves. In such cases, even small amounts of damping can be beneficial if the radiation efficiency is low, (ie) $\sigma_{rad} << 1$.

In most practical fabricated machinery structures however, it is seldom that a figure of η_s less than 0.01 occurs, with figures as high as 0.05 being measured in punch presses. Under these circumstances (η_s= 0.01) and for a typical structural thickness of one centimetre, the value of the expression $1.23d.\eta_s.f$ will be 1.23 at 100 Hz, 6.15 at 500 Hz and 24.6 at 2 kHz. As σ_{rad} is likely to be significantly less than unity below 500 Hz, no great loss in accuracy will occur and a far easier understanding of the general laws of noise emission will be obtained if we ignore σ_{rad} in the denominator and write

$$\frac{W_{rad}}{W_{vib}} = \frac{\sigma_{rad}}{1.23\ d.\eta_s.f.}$$ or in logarithmic A weighted decibel terms.

$$Leq(A) \text{ per event at any frequency} = 10\log W_{vib}(f) + |10\log(A\sigma_{rad}) - 10\log f$$

$$-10\log\eta_s - 10\log d + \text{Constant}$$

This method of writing the energy balance equation has the advantage that the contributions arising from changes, respectively, in the vibrational escape energy W_{vib}, the time history of this input (ie) the impulse shape, or the spectral content of the vibration, the acoustically frequency sensitive terms, the damping factor, and the bulk factor or average thickness (internal energy is related to the volume of material, the radiated sound to the surface area) can be added linearly and studied separately. Thus we have:

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$$Leq(A)(f) = 10\log W_{vib}(total) + 10\log(S.c) + 10\,|\log A\sigma_{rad} - \log f| - 10\log n_s - 10\log d + C$$

Deafness contribution per event =	Total Escape Energy into Structure	Spectral content at any frequency f	A weighted & frequency sensitive joint radiation/damping efficiency	Damping level	Machine Bulkiness or Solidity

For any given component structure geometry, these terms are independent of each other, and the effects of machinery change can be examined term by term and summated linearly. Thus the first term represents the total energy escaping into the structure per impact, the second term depends upon the impulse shape of the vibrational energy escape and of the structural response, and can be described in the form of a spectral distribution curve, the third term includes the radiation efficiency of any component of the structure, allowance being made for the sensitivity of the ear and for the shorter ringing period of the higher frequency vibrations, the fourth term represents the structural damping associated with the component and its interface with other parts of the machine, the fifth term represents the ratio of the radiating surface area to its volume or mass, (ie) the average thickness of webs, flanges, plates etc.

Noise Control Techniques

The rest of the given paper discussed what reductions in Leq(A) per event can be achieved under each of these headings. In view of the existence of a fairly high damping factor in any prefabricated machine, the amount of noise reduction to be obtained by the addition of damping is shown to be quite small. The two best methods of noise control lie in impulse reduction so shaping it that the vibrational energy is as small as possible in the frequency range where the 'acoustic radiation' function is at its highest. Figures are shown in this function for some typical machinery configurations, and it is deduced that the over-riding requirement is to soften impacts and to make work processes as uniform and progressive as possible. While this may sound self evident, the separation of the variables can be used to indicate in detail the gain to be obtained from any machinery modifications, without the use of complicated computer analyses which as often as not fall just short of reality in their representation of practical impact machinery.