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ENERGY INPUT, VIBRATIONAL LEVEL AND MACHINERY NOISE; SOME SIMPLE RELATIONSHIPS

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Introduction

The noise energy emanating from a machine can be as little as 10^{-8} or as large as 10^{-2} of the energy used by the machine. In either case, it is small and any improvement we can make is not likely to improve directly the efficiency of production or of operation. It is related more closely to the vibration level of the machine or to that of the air emission associated with the machine, but even so, the relationship is tenuous, and the impression has grown that 'noise' is some sort of mysterious and indeterminate adjunct to vibration and that the best thing to do is "to box the machine in" if its noise is excessive.

Most excessive noise occurs as a result of sharp impacts or discontinuities in the machine system, and in such cases surprisingly simple laws can be enunciated (1)(2)(3) which can be very helpful to machinery designers who do not wish to concern themselves with very elaborate computations (which as often as not just stop short of being realistic) but who nevertheless need basic diagnostic rules to tell them whether they are moving acoustically in the direction of quietness in their machine design work.

The first rule to state is that noise meters are, in most cases, integrators of noise energy over short periods of a second or so, and that the noise level measured from repetitive impact machines will be the same as that measured scientifically by an integrating dosemeter for a single impact and corrected for the number of impacts per second. Thus, from an engineering design point of view it is sensible to predict the noise emanating from a single machine cycle, (since this relates directly to the machine process) and to correct for the repetition rate.

The second rule arises from the fact that the radiated noise energy per event consists basically of two components (Fig 1), that arising from the transient or forced motion associated with the work process and its time history, (acceleration noise), the other (ringing noise) arising from the fact that the vibrational energy left in the machine after it has done its work must be dissipated eitheras heat (internal damping) or as radiated sound (acoustic damping). (The aim must therefore be to maximise the first, and minimise the second).

Dealing first of all with acceleration noise (which often determines the instantaneous peaks) it can be shown(1) that a body coming to rest instantaneously radiates into the far field an amount of energy equal to half that in a bag of air of the same shape travelling at the same initial speed. This is a maximum and it falls off with the time of deceleration. Indeed a master curve (Fig 2) can be drawn of the actual acoustic energy radiated as a fraction of this maximum against the distance this transient sound wave travels during the impact time as a fraction of the body size ($\frac{ct_0}{(vol)^{1/3}}$). It is therefore possible to predict limits of acceleration noise from such bodies as drop stamps, impacting bottles, cavitation bubbles, combustion, gears, billiard balls and so on.

Figure 2 shows the fall in $\mu_{acc}(= \frac{E_{acc}}{\frac{1}{2}\rho_o\, v^2 (vol)}$ with $\delta (= \frac{ct_o}{(vol)^{1/3}})$ for the classical case of two spheres impacting on each other, the theoretical curve agreeing excellently with the actual measured noise energy radiated. Fig 3 also includes a maximised curve which would not be exceeded for bodies of solid but non-spherical shapes[1]. Fig 4(a) and Fig 4(b) show the measured acceleration noise components both from the tup of a drop hammer involved in both soft and hard blows and from two glass bottles impacting each other via small inserts of varying softness (and therefore of varying values of $ct_o/(vol)^{1/3}$. It may be seen that the maximised curve provides a satisfactory guide to the amplitude of the acceleration noise, once the impact time and velocity are known.

It can also be shown[2] that on bodies of less solidity which have one or two dimensions much greater than the third, slow bending waves can take the work energy away from the work area, leaving the machine after impact in a state of considerable vibration. Under these circumstances, acceleration noise is less important than the noise radiated during this ringing phase, and the problem of noise prediction ceases to be related directly to the initial impact, but to the amount of energy stored as vibration and the fraction of this radiated as sound.

For a very poorly damped structure, all this energy E is radiated, (ie) $E_{rad} = E_{escape}$, but on most fabricated machinery the majority of the vibrational energy is absorbed as heat, leaving only a small remainder E_{rad} to be radiated acoustically This must obviously depend upon the efficiency of acoustic radiation σ_{rad}, the structural damping factor η_s, and the bulkiness (d) of the machine (obviously damping depends upon volume, radiation upon surface movements).

The ratio of that energy radiated as sound to the energy entering the machine as vibration at frequency f can be written simply for steel structures[3] in the form

$$\frac{E_{rad}(f)}{E_{escape}(f)} = \frac{\sigma_{rad}(f)}{\sigma_{rad}(f) + 1.17\eta_s d.f}$$

where d is an average thickness in centimetres. For undamped structures $E_{rad}(f) = E_{escape}(f)$ and we need only evaluate $E_{escape}(f)$ in any frequency band to establish $E_{rad}(f)$ in the same bandwidth. In most fabricated machines, σ_{rad} is small at low frequencies, and there is no great loss of generality in writing

$$E_{rad}(f) = E_{escape}(f) \times \frac{\sigma_{rad}(f)}{1.17\eta_s d.f}$$

or in logarithmic form and correcting for the A weighting with frequency.

$$L_{eq}(A)(f)(\Delta f) = 10\log E_{escape}(\text{total}) + 10\log(\text{s.c}) + 10\log\left(\frac{A\sigma_{rad}}{f}\right) - 10\log\eta_s - 10\log d + \text{Constant}$$

$L_{eq}(A)(f)(\Delta f)$	$10\log E_{escape}(\text{total})$	$10\log(\text{s.c})$	$10\log(A\sigma_{rad}/f)$	$-10\log\eta_s$	$-10\log d$
Deafness Contribution per event	Total Escape Energy into Structure	Fraction of total in frequency band being considered	A weighted and frequency sensitive joint radiation/damping efficiency	Damping Level	Machine bulkiness or solidity

If the structure is very lightly damped the noise energy and the vibrational energy are the same (ie) we can ignore all but the first two terms, and the problem of noise reduction equates to that of reducing the vibrational energy left in the machine after impact or fracture. Most machinery structures are fabricated and contain a signficant degree of structural damping; under such circumstances noise reduction in any frequency band requires the modification of one or several of the terms in the above equation. Thus noise reduction can result from a reduction in the quantity E_{escape}, in reducing the modified radiation efficiency term $10\log \frac{A\sigma_{rad}}{f}$ allowance being made for both the effects of A weighting and length of ringing, in increasing the damping factor η_s or in increasing the thickness 'd' (ie) in making the structure more 'solid'.

Fig 5 shows the values of $10\log\frac{A\sigma_{rad}}{f}$ for plates, solids and rods or beams of various typical sizes. It is a recognised characteristic in all these curves that they peak at frequencies which depend upon size, or cross sectional dimensions, and that it is obviously wise to keep the vibrational energy in the system to frequencies well below or well above these values. Indeed the process of optimisation of noise control can be illustrated in the diagram shown as Fig 6. As logarithmic additions are equivalent to multiplication, whereas the energy in each frequency band is summed linearly to obtain the total $L_{eq}(A)$, the process of noise reduction must reduce to that of mismatching of frequencies as much as that of reducing E_{escape} and increasing the damping coefficient η_s.

Fig 7 shows a form of heavy duty container or bin in which all these terms have been reduced compared with those occurring in a heavy standard steel container. The reduction in Leq(A) so obtained is 29 dB(A) equivalent roughly to a reduction in radiated noise energy to a one thousandth of the original figure for the same velocity of drop.

The second useful form for this relationship when no metal deformation occurs is obtained by replacing the first two terms, (ie) the expression for $10\log E_{escape}$, by an expression for the work done by the applied external force during the impact. This can be put into the form of a $(\text{force})^2$ times a structural mobility or more usefully from a diagnostic point of view in the form of the square of the rate of change of force multiplied by a relevant point receptance term. In this form, the equation for the A weighted noise radiated per second takes the form

$$L_{eq}(A,f\Delta f)=10\log N+10\log\left[|F'(f)|^2\right]_- + 10\log \, \mathrm{Re}\left[\frac{H(f)}{j}\right]+ 10\log \frac{A\sigma_{rad}}{f}$$
$$- 10\log\eta_s - 10\log d + 10\log \frac{\Delta f}{f} + \text{Constant}$$

N is the number of impacts per second, $|F'(f)|^2$ is an impulse shaping term which will depend in magnitude on the square of the sum of the rates of change of force, and $\mathrm{Re}\left[\frac{H(f)}{j}\right]$ is the imaginary part of the structure response term at a point and is defined in the frequency plane by $V(f)=H(f).F'(f)$. Thus a typical noise spectra will be made up of the various terms typically as shown in Fig 8.

In Conclusion

The lessons to be learnt from this formulation are many, but can be listed for convenience as follows:-

(a) the vibration level left in the structure will vary as some function of the rate of change of force, so that the strongest noise reduction technique available to us is to reduce $f'(t)$ to a minimum.

(b) if the machine process requires a large rate of change of force, this should be limited, by the use of resilient inserts, to reach as little of the highly radiative surfaces as possible so as to decouple in frequency the peaks in $10\log|F'(f)|^2$ and $10\log \frac{A\sigma_{rad}}{f}$.

(c) The damping term $10\log\eta_s$ should be maximised, but in view of the considerable damping already occurring in fabricated structures, large increases are necessary to achieve significant reductions.

(d) Depending upon size 10 log $\frac{A\sigma_{rad}}{f}$ will peak between 800 and 2000 Hz. It is important to prevent high level vibrations occurring at these frequencies. Thus softening the impact to excite frequencies out of the range of such maxima should be aimed at.

(e) Wall thickness is often determined by requirements of stress levels and stiffness requirements. It can be shown that some configurations, while providing adequate stiffness for work accuracy, are nevertheless too thin and frail for good noise control.

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2. E J RICHARDS, M E WESTCOTT, R K JEYAPALAN. On the Prediction of Impact Noise; Part II, Ringing Noise, JSV Vol 65, No.3 p419-451.

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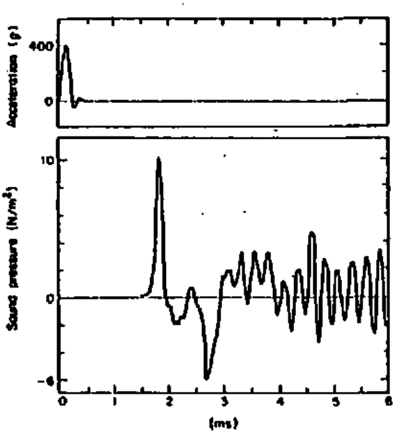


Figure 1. Typical acceleration and sound pressure signals from colliding cylinders.

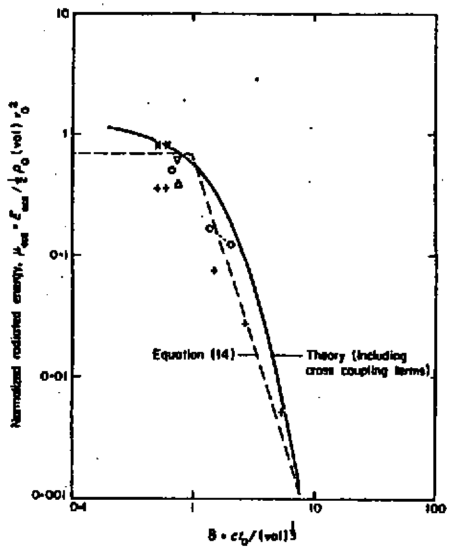


Fig 2: Normalized sound energy μ_{rad} *vs.* δ. O, 75 mm diameter cylinders; +, 100 mm diameter cylinders; ×, 150 mm diameter cylinders; ●, 100 mm diameter spheres; Δ, cones point to point ∇, cones base to base.

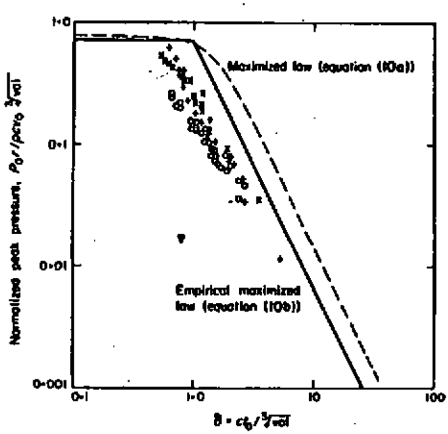


Fig 3: Normalized peak pressure *vs.* δ for cylinders and cones. ●, 75 mm diameter cylinders; +, 100 mm diameter cylinders; ×, 150 mm diameter cylinders; Δ, cones point to point; ∇, cones base to base.

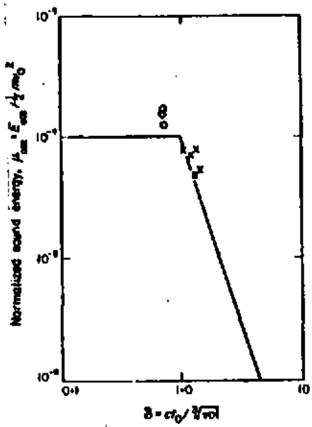


Fig 4a: Normalized sound energy $\mu_{acc} = E_{acc}/\frac{1}{2}mv_0^2$ vs. δ for friction drop stamp. O, Die to die blows; x, forging blows; ——, equation (14).

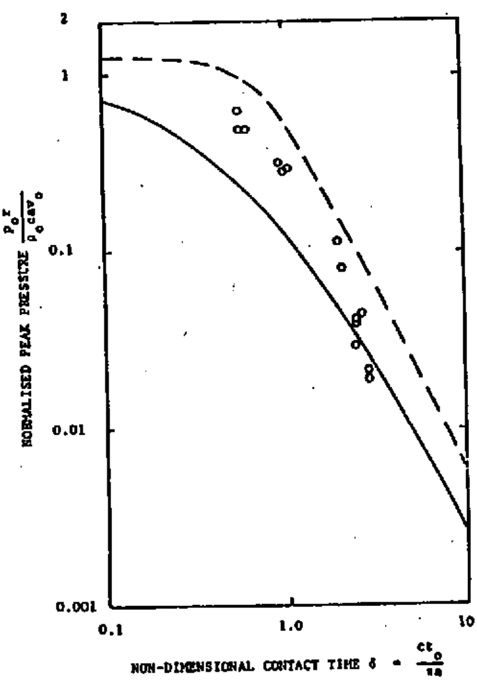


FIGURE 4b : NORMALISED PEAK PRESSURE vs CONTACT DURATION FOR BOTTLE TO BOTTLE IMPACTS

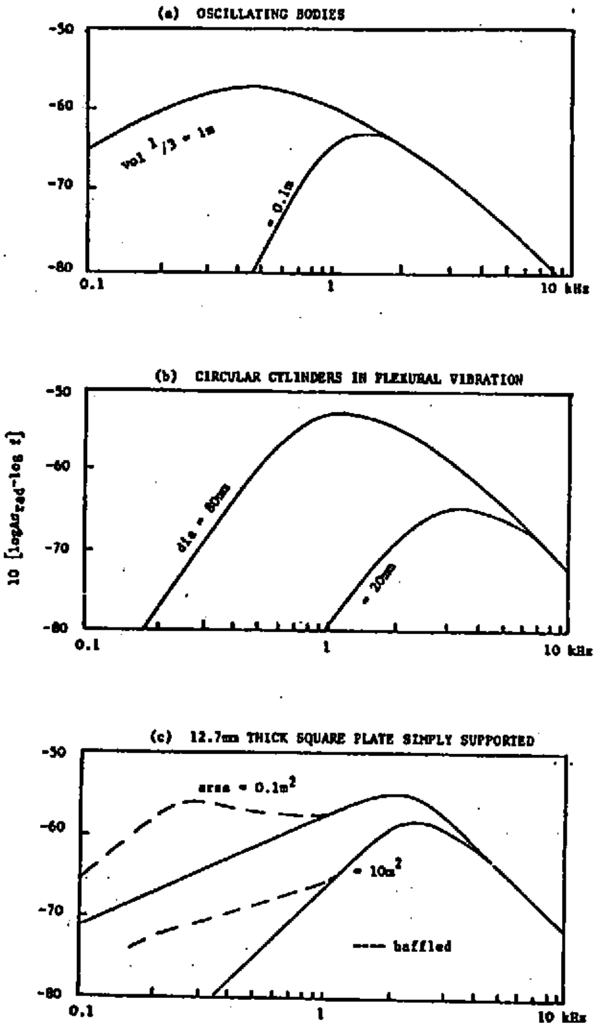
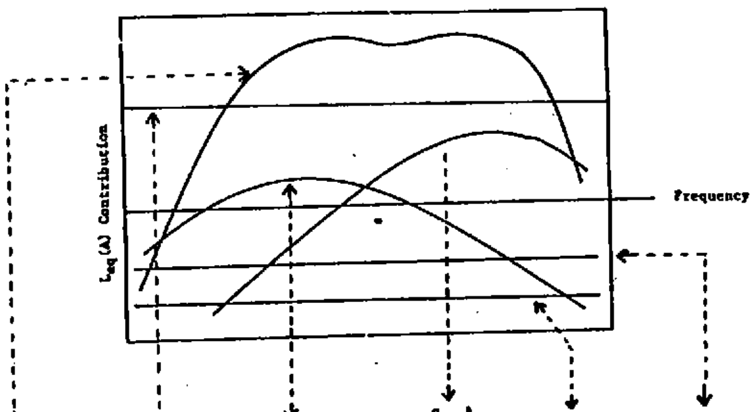


FIGURE 5: THE VARIATION OF 'MODIFIED' RADIATION EFFICIENCY OF VARIOUS BODIES PLATES AND BEAMS WITH FREQUENCY

FIGURE 6

THE MAKE-UP OF MACHINERY NOISE AT ANY PARTICULAR FREQUENCY (f)



$$L_{eq}(A)(f) = E_{structural}(total) + S.D. + \left[10 \log \frac{\sigma_{rad} A}{f}\right] - 10 \log \eta_s - 10 \log d + \text{Constant}$$

- $L_{eq}(A)(f)$: TOTAL A WEIGHTED NOISE ENERGY RADIATED AT FREQUENCY (f).
- $E_{structural}(total)$: TOTAL VIBRATORY ENERGY LEFT IN MACHINE AFTER IMPACT.
- S.D.: SPECTRAL DISTRIBUTION OF THIS VIBRATORY ENERGY.
- $10 \log \frac{\sigma_{rad} A}{f}$: ACOUSTIC EFFICIENCY OF RADIATION OF STRUCTURE ALLOWING FOR BEHAVIOUR OF THE EAR AND LENGTH OF RINGING TIME.
- $10 \log \eta_s$: LEVEL OF STRUCTURAL DAMPING.
- $10 \log d$: BULKINESS FACTOR, TYPICALLY EXPRESSED IN AVERAGE THICKNESS OF MEMBERS.
- Constant: CONSTANT DEPENDING UPON UNITS USED.

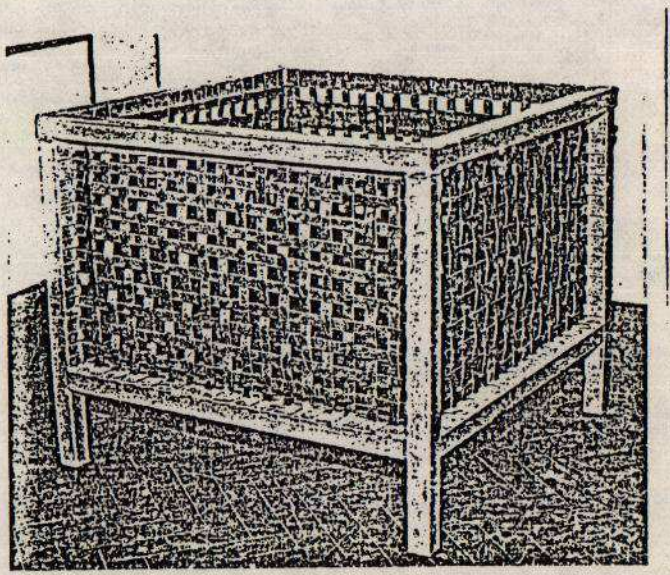


FIGURE: 7 WOVEN STILLAGE

FIGURE 8 SUMMATION OF CONTRIBUTIONS TO L_{eq}