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IMPACT NOISE IN MACHINERY: ITS PREDICTION
PART I: ACCELERATION NOISE

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1. Introduction Some ½ to 1 million industrial workers in Great Britain are currently exposed to noise levels which will cause a serious hearing loss in thei working lifetimes. The introduction of legislation regarding the limits of noise in factories has led to the need for prediction of the likely noise levels produced by a machine while at its design stage.

2. Mechanical Noise Sources In most machines, noise arises as a result of an impact of some kind, whether in the obvious case of a drop hammer or the less obvious cases of saws, bearings and gears. Energy is usually accumulated over a relatively long period, then dissipated into the workpiece over a much shorter period. Consider the various sources of noise in an impact machine:

(i) Air may be ejected from between the approaching surfaces.
(ii) The impacting bodies accelerate/decelerate impulsively, causing pressure disturbances in the surrounding air (acceleration noise)
(iii) the workpiece changes shape with similar effect.
(iv) both workpiece and machine subsequently vibrate, dissipating excess energy not used in the work process as heat and radiated sound.

The remainder of this paper is concerned with acceleration noise, while Part II is concerned with the subsequent ringing noise.

3. Acceleration Noise Consider the simplest case of an impact, that between two identical rigid spheres. They will remain in contact for a finite time, typicall fractions of a millisecond, during which a force acts between them and the surfaces experience a high acceleration which may radiate a sound pulse (Fig 1).

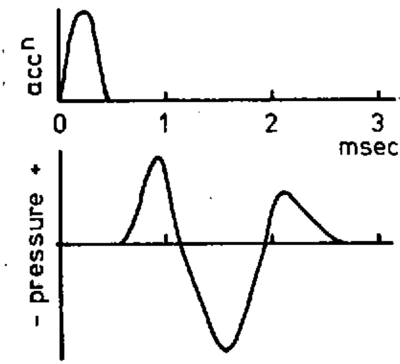

Fig 1: Acceleration and pressure signals for impacting spheres

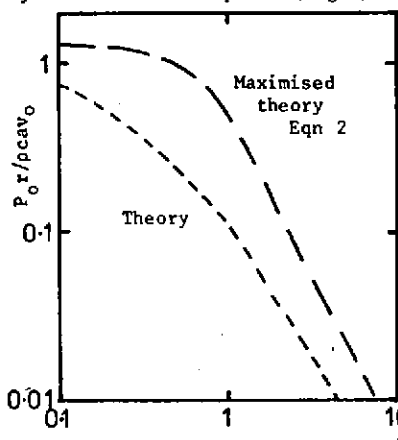

Fig 2: Normalised peak pressure for impact spheres

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It has been shown [1] that the energy required to accelerate a rigid sphere from velocity zero to v_o is given by $E = \frac{1}{2}(m+m')v_o^2$, where m' is a virtual mass $=\frac{1}{2}\rho_o(vol)$, if the acceleration is 'slow' and no compressibility occurs. This energy is carried along in the fluid adjacent to the sphere as it travels at velocity v_o and is recoverable on 'slow' deceleration. If, however, the acceleration is 'high' and compressibility does occur, then an additional amount of energy is needed. If the contact is very short relative to the time taken for a sound case to pass the sphere, it is equal to $\frac{1}{2}$ the K.E. contained in an air bag of the same volume having the same final velocity v_o. This is not recoverable on deceleration and must be radiated as sound, so an upper bound on the energy radiated as sound from an impact of 2 identical spheres is given by $\frac{1}{2}\rho_o(vol)v_o^2$.

Thus the total radiated acoustic energy cannot be greater than ρ_o/ρ_m x input K.E. (1.5×10^{-4} x KE for steel) and an 'impulsive radiation efficiency' μ_{acc} can be defined as $E_{acc}/\frac{1}{2}\rho_o(vol)v_o^2$ (a dimensionless ratio like σ_{rad}) which can never exceed unity. Consider now the effect of increasing the contact duration t_o on the radiated sound energy E_{acc}. As t_o increases for bodies of a given size, cancellation will begin to occur if sound can travel from one side of the body to the other during t_o. Holmes [1] derives an expression relating acceleration noise to t_o for bodies of arbitrary shape in the region of cancellation:

$$E_{acc}=\rho_o L^3 v_o^2 (L/ct_o)^5 \quad \text{where } L = \text{typical dimension}, \; c = \text{speed of sound} \tag{1}$$

It is seen that increasing t_o has a very beneficial effect on the noise radiated.

4. Relationship between the peak pressure and impact duration

The sound radiated from an accelerating body is highly directional being greatest along the axis of impact. Simplifying the equations given by Koss and Alfredson [2] by retaining only the far field terms along the impact axis gives an expression for the pressure P_o in relation to the non-dimensional contact time $\delta=ct_o/\pi a$. This is plotted in Fig 2 together with a maximised law for 2 spheres, given by

$$\frac{P_o r}{\rho c a v_o} = \frac{1+\sqrt{2}}{2} \cdot \frac{1}{\sqrt{4\delta^4+1}} \tag{2}$$

A pendulum rig has been built on which bodies of various types could be impacted. A plot of $P_o r/\rho c v_o \sqrt[3]{vol}$ vs δ is shown in Fig 3 for a variety of spheres, cylinders and cones. Long contact durations were obtained using small metal and rubber inserts. In this Fig the a in the ordinate of Fig 2 has been changed to $\sqrt[3]{vol}$ to allow for variation of shape. An empirical curve may be drawn, approximated by

$$\frac{P_o r}{\rho c v_o (vol)^{1/3}} \begin{cases} = 0.7; & \delta<1.0 \\ = 0.7^{-2} & \delta>1.0 \end{cases} \qquad \delta = \frac{ct_o}{\sqrt[3]{vol}} \tag{3}$$

5. Relationship between Energy radiated E_{acc} and contact duration

Noise legislation in the UK is firmly linked to Leq so a prediction of total radiated energy is also required. The total radiated noise energy may be obtained by summing and integrating the acoustic pressures over a surface

$$E_{acc} = \iint \frac{(P_1+P_2)^2}{\rho c} \, ds \, dt \tag{4}$$

which includes any cross-coupling terms. These expressions have been evaluated and it is seen from Fig 4 that the cross coupling terms are only significant at high values of δ, when μ_{acc} has already fallen by some 20 dB. In practice hammer/anvil configurations are somewhat different so the energy radiated has been measured for impacts of simple bodies by integration of microphone signals over a spherical surface. Fig 5 shows these to be in good agreement with theory, and an empirical approximation can be made

$$\frac{\mu_{acc}}{2} = \frac{E_1}{\frac{1}{2}\rho_o(vol)v_o^2} \quad \begin{array}{ll} = 0.7 & \delta<1 \\ = 0.7\delta^{-3.2} & \delta>1 \end{array}$$

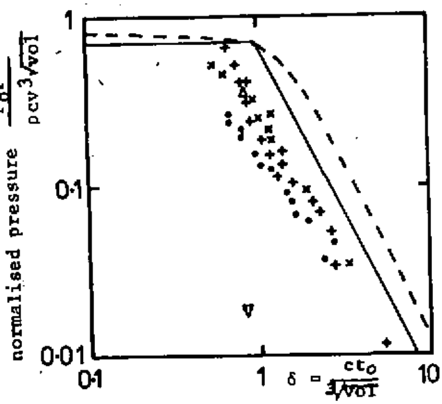

Fig 3: normalised peak pressures for cylinders and cones

100mm dia cylinders +
150mm dia cylinders ×
cones point to point ▲
cones base to base ▼

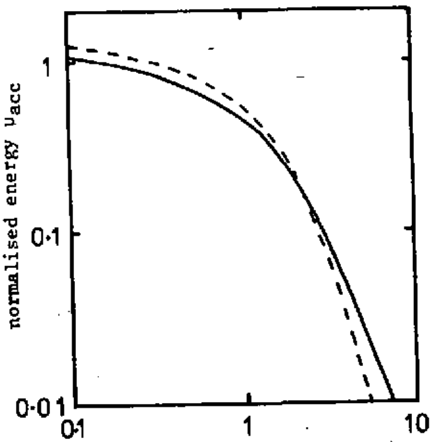

Fig 4: normalised energy vs δ

$\delta = \frac{ct_o}{\sqrt[3]{vol}}$

with cross coupling - - - -
without cross coupling ———

6. Acceleration Noise from a Drop Hammer

To establish the importance (if any) of acceleration noise in a real machine, pressure measurements were made around a drop hammer over a range of forging conditions. (Fig 6).

7. The Relative Magnitude of acceleration noise energy

It has been shown that E_{acc} cannot exceed about 1.5×10^{-4} times the input kinetic energy, and its value is predictable. The energy radiated by the subsequent vibration is less predictable, as the vibrational energy is dissipated in internal damping and as sound. It can be shown [1] that

$$\frac{E_{acc}}{E_{vib}} \propto \frac{\eta_T}{\eta_R}$$

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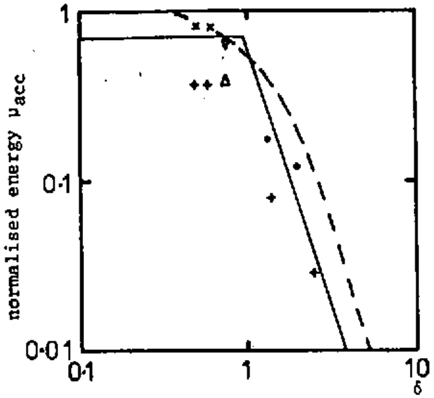

Fig 5: normalised energy vs δ
- spheres •
- 100mm dia cylinder +
- 150mm dia cylinder ×
- cones base to base ∇
- cones point to point Δ

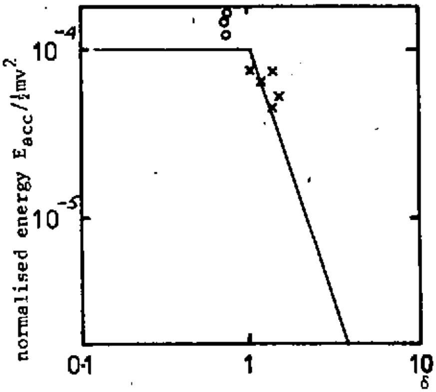

Fig 6: normalised energy μ for friction drop stamp
- die to die blows o
- forging blows ×
- maximised equation ———

where E_{vib} = energy radiated as sound
η_T = total loss factor
η_R = acoustic loss factor

for the simple case of impacting spheres.

Using the pendulum rig, a series of cylinders and cones were impacted and the ratio E_{acc}/E_{vib} measured.

8. Conclusions

(1) There is a limiting value to E_{acc} for short contact durations.
(2) Above a value of δ=2 this energy falls off rapidly.
(3) The dominating noise mechanisms which produce acceleration noise must be those with the shortest contact times and may be associated with backlash, bearings, gears etc.
(4) It is unlikely even in the case of a drop hammer that acceleration noise causes an Leq > 90 dBA.

REFERENCES

(1) D G HOLMES, Jan 1976, Journal of Sound and Vibration Vol 44 No.1 Impact sounds due to sudden accelerations.

(2) KOSS and ALFREDSON, March 1973, Journal of Sound and Vibration Vol 27, no. 1, Transient sound radiated by spheres undergoing an elastic collision.