DROP HAMMER NOISE 1: THE EFFECTS OF STRESS CONCENTRATIONS IN THE IMPACT AREA

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An experimental analysis of sound energy radiated by a small drop hammer derived from vibration measurements [1] has indicated that a large fraction (-46%) of the total originates in the impact area, i.e. the dies and dieholders, but at frequencies too low to be accounted for by the natural flexural and longitudinal vibration of these components alone. This area must therefore receive priority in any noise reduction programme (in this case reduction at source) and is being considered in some detail.

Typical acceleration time histories in the impact area for forging blows are shown in Fig 1, which, coupled with A weighting, radiation efficiency and surface area, provide a measure of the sound energy radiated by each component. These are characterised by a high amplitude heavily damped transient during the contact period, having a frequency between 1.3 and 1.5 kHz which when integrated is seen to account for most of the radiated energy and is clearly apparent on sound pressure signals. For die to die blows, however, the nature of the vibration signals differs, being of a higher frequency and continuing for some time after contact has ceased.

Several mechanisms have been suggested to account for this low frequency vibration ('low' is used here in a relative sense, 1.5 kHz being at the peak of the ear's sensitivity).

(i) A simple continuous system representing the tup/bolster/dies/sow/anvil in which longitudinal waves travel from the top of the tup to the base of the anvil and back, Poisson's ratio effects causing the measured lateral vibration in the smaller cross section of the impact area. The path length involved indicates a frequency of about 1.2 kHz, close to that measured and would cease on separation of the dies.

(ii) A simple lumped mass - spring system in which the mass of the tup acts on the stiffness of the upper die during the contact period, with Poisson effects as in (i).

(iii) Cantilever, shear and axial modes of the die/dieholder assemblies acting on the resilience provided by stress concentrations in the dovetail joints used for die location, both during and after the contact period.

In practice all three mechanisms (and others?) may occur simultaneously with varying contributions to the noise energy radiated, according to the type of blow. Subsequent measurements have shown that the dies vibrate in oscillation rather than pulsation due to Poisson effects, so mechanism (iii) has been considered in some detail. Investigation of the vibration characteristics of the impact area is complicated by the number of interfaces and components in what is at first sight a simple machine, but the availability of a pendulum impact rig [1] has enabled the vibration of solid components jointed by dovetails to be examined in isolation.

The vibrational behaviour of a solid steel cylinder (150mm dia x 300mm long), halved and jointed by a dovetail and wedge (Fig 2) was examined on the pendulum rig and found to be very different from that of a similar continuous one, both axially and in flexure; the jointed cylinder appears to vibrate as a mass-spring-mass system at low frequencies in addition to the natural modes of the two component halves. These frequencies and loss factors are dependent on the wedge tightness and the system is highly non-linear (hardening spring type) evinced by asymmetric spectra and non-constant decay rates. Similarly a model die block dovetailed to larger masses exhibited the vibration modes shown in Fig 3 (although six modes were possible, only those three could be positively identified).

If these coupled blocks can be represented by the simple systems of Figure 4 then the equivalent stiffness of the connections can be estimated knowing the masses and frequencies, e.g.

Cylinder pair	k_e-3.8x10^9 N/m	Differences due
One third model die on cylinder	k_e-8.4x10^9 N/m	to changes in
One third model die on tup	k_e-5.6x10^9 N/m	geometry?

It remains to establish over what area and volume of the material the equivalent stiffness acts. Consider the joint shown in Fig 2 in which the wedge W is driven into the gap between the die block dovetail and die holder recess. This results in compressive stresses across the sides and base while tensional stresses develop at the corners and parallel to the base, and the whole assembly behaves as a pre-stressed spring. When a direct (compressive) force is applied the area below the dovetail will be subjected to further compression while the volume adjacent to this will be subjected to shearing forces. It may be expected that the overall stiffness is dominated by the area below the dovetail but it has been shown that the stress below a die (cylindrical) acting on a semi-infinite half space is not uniform, but given by the expression [2]

$$q_r = P\Big/2\pi a^2 \sqrt{1-r^2/a^2} \qquad (1)$$

where q_r = stress at radius r, P = applied load, a = radius of die

This in theory, rises to infinity at the edges, and integration over the contact area shows that half the total load is taken outside of 0.867a. Similar distributions can be expected for square and rectangular dies thus the area of stress concentration might be quite small, as indicated by photoelastic figures.

An approximate volume of material over which the stiffness k_e acts can be found using the expression for the stiffness of a block, k=EA/L or L=EA/k – (2) where E = Young's modulus, A = Area and L = length. Assuming first that the effective spring area is that of the whole dovetail base (for the cylinders) then substitution into (2) gives L-0.4m, a result clearly in error since this is longer than the cylinder pair. Taking only the projected area of the tapered sides (Fig 2) gives an equivalent length of 28mm, or 1.6 times the dovetail depth, a much more realistic value. Extrapolating this result to the 1 ton hammer gives the following frequencies:

Upper die on bolster	- 1900 Hz	Axial modes - in reasonable
Upper die on bolster on tup	- 1340 Hz	agreement with the measured
Lower die on sow and anvil	- 1320 Hz	values (1300 - 1500 Hz).
Upper die on tup	- 2076 Hz	Rotational modes.
Lower die on sow	- 820 Hz	

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To gain further understanding of the stiffness characteristics of dovetail joints, static stiffness measurements were made on the die/sow assembly of a one third scale model hammer (to be described in Part 2) with a universal testing machine (compression only). These are shown in Fig 5 for a range of wedge tightnesses and are, as expected, non-linear, following approximately a cubic law $F = kx^3$.

For comparison a solid block of similar dimensions was tested (Fig 5) and was also non-linear but less so than the assembly, following approximately a square law $F = kx^2$. Also shown is the stiffness line for the solid block calculated from equation (2), $k = 2 \times 10^{10}$ N/m. An initial non-linearity might be expected since the load is first taken by the high spots only, the contact area increasing as the load increases giving a higher stiffness. A similar effect might be caused by the lateral force of the wedge arching the sow block but no clear relationship between stiffness and wedge tightness was observed (Fig 5). It seems likely therefore that the much reduced and non-linear stiffness which leads to mass-spring-mass modes in dovetailed components is due to the effects described above (Equation 1) i.e. the effects of stress concentrations at the edges of two bodies of dissimilar cross-section area in contact.

In conclusion, while the importance and nature of the sources in the impact area have been indicated (excluding acceleration and air ejection noise), the prospects for noise reduction at source without recourse to enclosure appear to be limited. Consider the general expression for noise radiated from an impact [3]

$$L_{eq}(A, f_o, f) = 10\log N + 10\log E_s + 10\log A\sigma_{rad}/f - 10\log \eta_s - 10\log d + C$$

E_s (the initial energy of vibration) might be reduced but the mechanism for this is not yet fully understood. The effects of A weighting and lower σ_{rad} at low frequencies might be utilised by reducing the frequency of vibration, and finally an increase of damping η_s might be considered. Some of these solutions may be achieved by changes to the dovetail geometry, but must be considered within the restraints imposed by the requirements of forging efficiency and safety. Modifications to the time honoured practices found in the forging industry may be difficult to introduce.

REFERENCES

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2. Timoshenko and Goodier. Theory of Elasticity. McGraw Hill, 1969.
3. E.J. Richards. JSV (to be published). On the Prediction of Impact Noise Part III; Energy Accountancy in Industrial Machines.

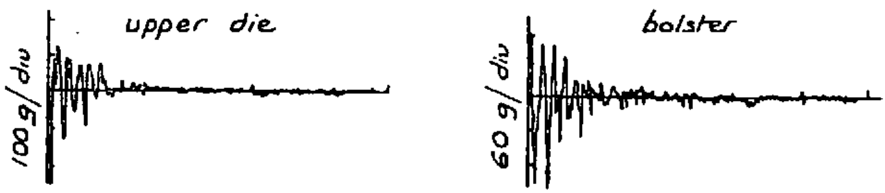


FIG 3: Typical acceleration time histories in the impact area

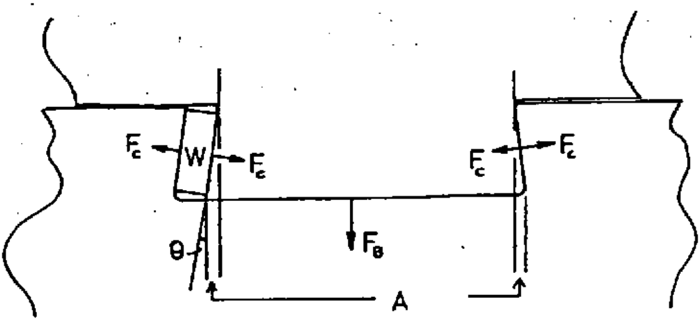
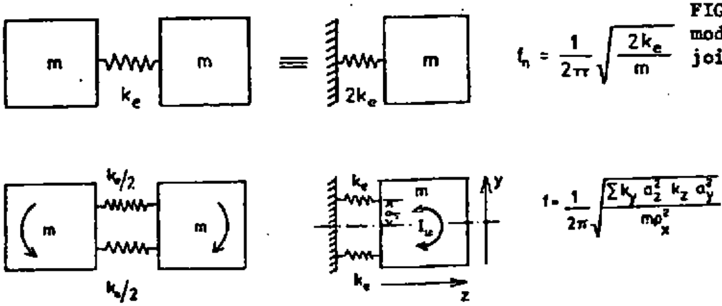


FIG 2: Dovetail joint used for die location



$$f_n = \frac{1}{2\pi}\sqrt{\frac{2k_e}{m}}$$

$$f = \frac{1}{2\pi}\sqrt{\frac{\sum k_y a_z^2\ k_z a_y^2}{m\rho_x^2}}$$

FIG 4: Simple models for dovetail jointed components

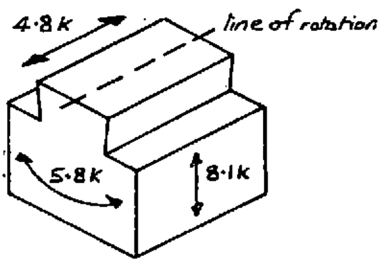


FIG 3: Modes and Frequencies of a model die dovetailed to a model cup (Of 6 possible modes only those 3 could be positively identified).

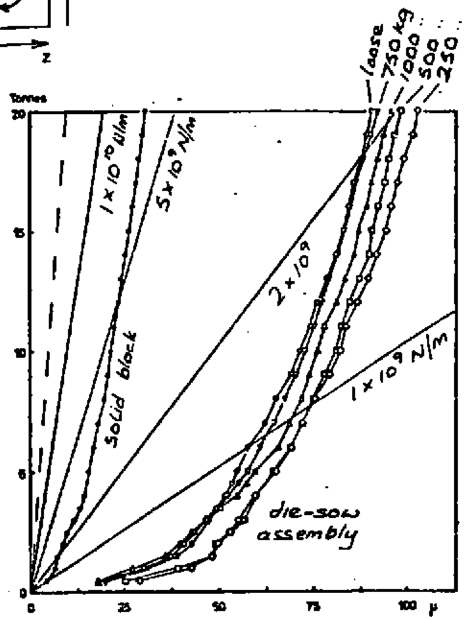


FIG 5: Measured static stiffness for model die/sow assembly.