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AN ESTIMATION OF THE NOISE RADIATED FROM A MODEL DROP HAMMER USING SURFACE VELOCITY TECHNIQUES

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INTRODUCTION

The drop hammer is a classical example of an impulsive noise generating system that radiates high L_{eq} noise levels when in normal operation. By far the noisiest operations performed are die-to-die blows and finishing blows, when little or no workpiece distortion occurs and extremely high force levels are experienced between the dies. The noise generating and radiating systems are complex, and as work on a full size hammer is difficult, a one-third scale model of a Massey Marathon 1-ton friction drop stamp was constructed in a semi-anechoic chamber within ISVR. The geometric modelling was made as exact as possible, as can be seen from fig. 1, and the materials selected for each component were the same as for a full size hammer.

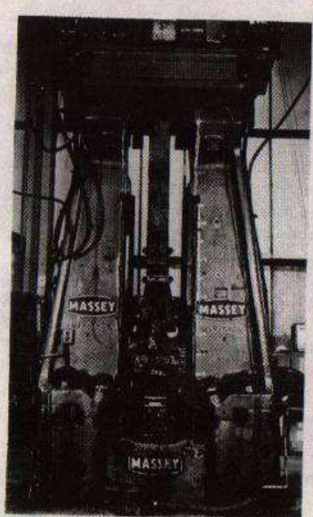

Fig. 1(a) Full size 1-ton friction drop stamp

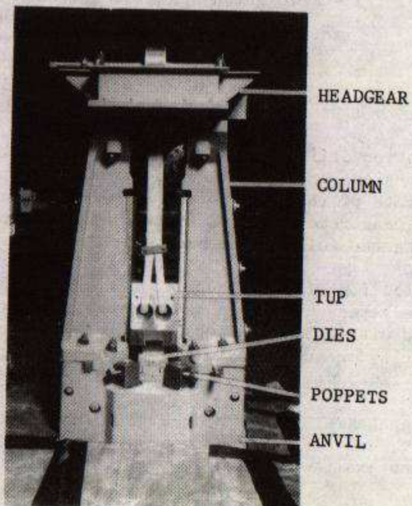

Fig. 1(b) Model friction drop stamp

NOISE GENERATING MECHANISMS

Noise radiated from an impulsive system can be divided into two main categories: (i) Acceleration noise, occurring only during the impact, caused by the rapid accelerations of the impacting bodies; (ii) the subsequent ringing noise emanating from the components of the system set into free vibration. In most machines, ringing noise is the major source, with acceleration noise a secondary factor [1,2,3].

In this case, we are considering acceleration noise from the tup and anvil, and ringing noise from the whole structure. Other sources, e.g., billet expansion, air ejection, headgear mechanism noise, are not considered as they have been shown elsewhere to be insignificant (or, in the case of headgear mechanism noise, a separate and easily dealt with problem) [4].

ESTIMATES OF RADIATED NOISE FROM SURFACE VELOCITY MEASUREMENTS

Noise energy radiated from the drop hammer per blow, A-weighted to represent deafness contribution, can be written as [2]:

$$L_{eq}(A,\ f,\ \Delta f) = \sum_{\substack{\text{all} \\ \text{components}}} 10 \log \int_{o}^{T} A.S\sigma_{rad}\overline{\langle v^2\rangle}\ dt + \text{Constant}$$

where A = A-weighted spectral content

σ_{rad} = radiation efficiency of component

$\overline{\langle v^2\rangle}$ = short term time-space average surface velocity (normal to component surface) squared

T = time over which averaging is performed

Radiation efficiency curves for individual components can be calculated from their sizes and known modes of vibration [5]. Therefore, in order to obtain an estimate of radiated noise per blow from each component of the drop hammer, the only measurement needed is the time and space averaged surface velocity squared during and following a blow.

Examination of microphone traces around the hammer indicate that non-reverberating noise levels have high decay rates, and at least 95% of the noise energy is produced within 35 ms of the blow. Therefore, a 40 ms integration time for surface velocity measurements was used, and an impulse sound level meter (35 ms integration time) was used to measure the sound energy radiated per blow.

A drop hammer in normal operation has a maximum repetition rate in the order of 1 sec, and this, combined with the high decay rates mentioned above, means that we can examine single blows rather than continuous operation of the drop hammer.

Surface velocity measurements were taken on the model using Brüel and Kjaer miniature high-g accelerometers through B & K 2365 Charge Amplifiers, and analysis was performed on an HP 5420A signal analyser. Measurements were taken at closely spaced intervals on sectors of components to ensure accurate averaging, e.g., 78 equally spaced measurement positions on one half of one column, 38 on one quadrant of the anvil. Measurements were taken on all components, including the moving tup, and the results are shown in the following table, as energy per component in one-third octave bands with a scale A-weighting factor applied. A particular advantage of this tabulation is that it shows components and frequencies of importance. Those items which contribute more than one hundredth of the total noise energy have been marked with a tick.

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Table 1: Radiated energy for drop hammer components (estimated from v^2 measurements).

Third Octave Bands	Small concrete block	Large concrete block	Anvil	Column	Sandblock and lower die	Upper heads	Tup	Upper die	Total
250	.001	.001	.001	.001	.000	.000	.003	.000	.007
315	.001	.003	.003	.001	.000	.000	.007	.000	.012
400	.013	.007	.001	.003	.009	.009	.001	.004	.021
500	.020	.017	.004	.001	.000	.000	.081	.000	.125
630	.0619	.006	.021	.033	.001	.000	.2154 ✓	.000	.357
800	.029 ✓	.005	.015	.099	.004	.002	.303 ✓	.001	1.270
1000	.221 ✓	.107	.018	.041	.010	.011	.542 ✓	.003	1.060
1250	.314 ✓	.010	.072	.030	.018	.021	.633 ✓	.002	1.060
1600	.318 ✓	.067	.093	.302 ✓	.049	.036	.601 ✓	.005	1.746
2000	1.610 ✓	.006	.416 ✓	.406 ✓	.131	.091 ✓	.471 ✓	.008	4.120
2500	.357 ✓	.008	.199 ✓	1.590 ✓	.334 ✓	1.170 ✓	.326 ✓	.007	3.910
3150	.296 ✓	.006	.400 ✓	.669 ✓	.343 ✓	.121	.126	.087	1.990
4000	.149	.001	.083	.790 ✓	.174	.014	.036	.029	.769
5000	.036	.000	.122	.733 ✓	.079	.013	.135	.016	.508
6300	.045	.002	.146	.137	.071	.013	.017	.074	.434
8000	.099	.001	.264 ✓	.166 ✓	.209 ✓	.020	.154	.061	.959
Total	4.43	.28	1.81	4.23	1.39	1.26	3.76	.18	18.3

= 102.6 dB (ref = 10^{-12} J)

Radiated energy measured by microphone = 24.7 mJ = 103.9 dB (ref. = 10^{-12} J)

We can compare this estimate with the total noise energy from microphone measurements. An imaginary cylinder was defined around the hammer, and 72 equally spaced measurement positions were used on the cylinder surface, for accuracy. Impulse sound level measurements, averaged over 3 die-to-die blows were taken at each position, and the total energy radiated per blow was calculated. This was repeated using a third octave filter. Figure 2 shows the comparison in third octaves between microphone and $<\overline{v^2}>$ results. It can be seen that there is good agreement between the two methods, both in total and third octave noise. The discrepancy at high frequencies is most likely due to this energy emanating from higher order modes, and the spatial averaging of the velocity measurements becomes insufficient for accuracy.

The whole process was repeated on a full size Massey Marathon friction drop stamp, and the agreement between microphone measurements and $<\overline{v^2}>$ estimates was again very close, within 1 dB in total with the same high frequency discrepancy.

Examining again, Table 1, all except one of those regions of high energy content can be accounted for by some resonant system, either as a vibrational mode of a single component, or as two or more components connected by a springing device. The exception is the large amount of energy from the tup at frequencies below 4 kHz. This energy cannot be accounted for by any vibrating system, and so must be due to acceleration noise.

Earlier work on the full size hammer |5| suggested that acceleration noise was relatively unimportant, and that significant reductions in total noise could be achieved by treating only the ringing of the hammer components. These new estimates from $<\overline{v^2}>$ measurements, however, suggest that acceleration noise is significant. Measurements on the full size hammer give acceleration noise as 40% of total noise.

To validate the $<\overline{v^2}>$ technique for measuring acceleration noise energy, results have been compared with two other acceleration noise estimation techniques.

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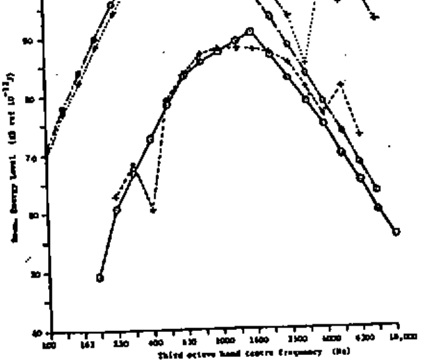
(i) By integrating pressure^2 signals with time around clashing cylinders on a pendulum rig. That portion of the resulting trace due to acceleration noise can be identified by its rapid rise time. Total acceleration noise energy can be estimated from traces measured on the surface of an imaginary sphere around the clashing cylinders, and compared with acceleration noise estimated from $\langle \overline{v^2} \rangle$ measurements. Acceleration noise energy from these two methods are within 0.5 dB of each other in total (no frequency-related comparison is possible with this method). This pressure^2 integration method is accurate in simple experiments but is not useful for the complex pressure^2 time signatures that emanate from a drop hammer, and so cannot be used for direct comparison.

(ii) Acceleration noise can be estimated from the force-time history of the blow (see reference |6|). Figure 3 shows the estimates for acceleration noise from $\langle \overline{v^2} \rangle$ and force-time data for both model and full size hammers. Agreement between each estimate is very close, with ringing noise becoming apparent on the $\langle \overline{v^2} \rangle$ curves above 3 kHz. Agreement is within 1.5 dB(A) overall.



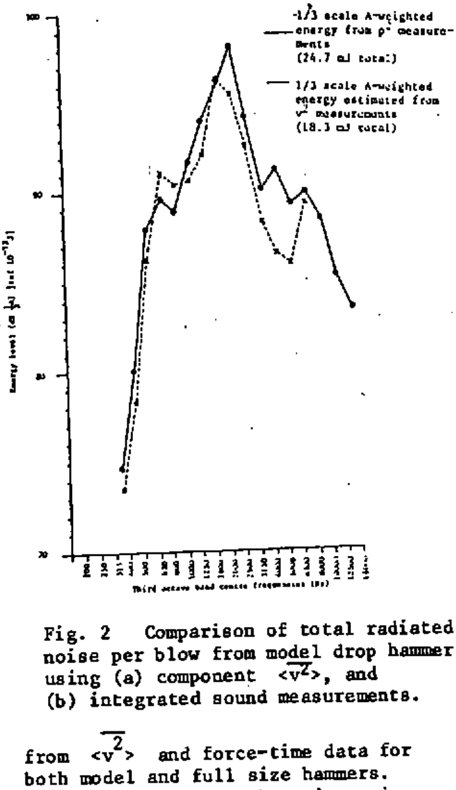


Fig. 2 Comparison of total radiated noise per blow from model drop hammer using (a) component $\langle \overline{v^2} \rangle$, and (b) integrated sound measurements.

CONCLUSIONS

Unlike the majority of impulsive machinery, acceleration noise from drop hammers is a significant source of noise energy, particularly for die-die blows (measured for a full size hammer as 40% of the total energy coming from acceleration noise). These measurements suggest that very modest reductions in overall noise energy (4 dB(A) maximum) can be achieved by the relatively easy task of reducing ringing noise, while significant reductions can only be gained by reducing tup acceleration noise.

Acceleration noise is directly related to the forging operation of the hammer, as the high force levels necessary for workpiece distortion are obtained by high acceleration levels. The problem of reducing acceleration noise without affecting forging efficiency becomes an area for future research.

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