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DROP HAMMER NOISE 2: PROGRESS OF THE JOINT ISVR/DFRA QUIET HAMMER PROJECT USING A 1/3RD SCALE MODEL

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Introduction: Since little demonstrative work can easily be performed on a real drop hammer, the ISVR and Drop Forging Research Association (DFRA) have jointly constructed a 1/3rd scale model of a Massey Marathon 15cwt friction drop stamp of the type installed in the DFRA's laboratory, Sheffield. The advantages of modelling are well known and need not be repeated here. The programme may be summarised:

(i) To validate the use of a 1/3rd scale model; demonstrating

(a) repeatability
(b) similarity of sound pressure and energy spectra
(c) similarity of directivity

with regard to scaling factors, for all forging conditions.

(ii) To examine the relationship between the noise spectra and vibration spectra on various components of the machine, and identify the importance of each as a noise source, both by direct comparison and later using coherence techniques.

(iii) To establish a base-line for noise from a free impact of the tup onto the anvil with the frames and guides removed, then determine the paths by which energy escapes from the impact area to excite other parts of the machine.

(iv) To systematically modify the model to reduce noise to a minimum by investigating the effects of

(a) tup geometry, using various shapes of constant mass
(b) damping - where and if practical
(c) blocking energy transmission paths between the tup, anvil and frames
(d) die location methods - practical alternatives

(v) The design of a second 1/3rd scale model incorporating the findings from above for demonstrative purposes.

The model is an exact replica except for the headgear which utilises a winch rather than the electric motor/clutch system of the full size machine. It has been installed on a concrete block in the semi-anechoic Westland laboratory of the ISVR (Fig 1).

Results of Phase 1: The first stage of the above programme is nearing completion (January 1981). Prior to assembly the natural frequencies and loss factors of each component were determined in isolation.

Repetition of sound pressure signals for die to die blows is excellent provided that the various wedges in the impact area remain undisturbed; the

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spectra for 20 consecutive blows were virtually identical. Integration of these shows that about 90% of the sound energy radiated is below 3 kHz.

Dimensional analysis shows that model frequencies are inversely proportional to the scaling factor L, thus they should be exactly three times those of the real machine. A comparison of sound pressure spectra and integrals for the model and real machine (with frequency scale suitably modified) is given in Fig 2 and it is seen that the distribution of peaks is broadly similar (although the calibration for the real machine has been lost). Exact agreement cannot be expected since (a) the frequency of much of the energy is determined by the wedge conditions in the impact area and (b) the acoustic environment differs, the real forge being rather reverberent.

These spectra are dominated by a single peak at about 850 Hz, and others between 1500 and 3000 Hz (model frequencies). It was believed that the 850 Hz peak was due to the concrete mounting block, and sound measurements with and without an isolated lead cladding have confirmed this:

(i)	Sound energy from full size machine (die to die, 1090 kg mass, 0.8m drop height ≡ 872 Joules)	~3.9 Joules unweighted
(ii)	Sound energy from 1/3 scale model (die to die, 32.6kg mass 0.3m drop height ≡ 9.78 Joules) Concrete block exposed	$\sim 6.7\times10^{-2}$ J unweighted
(iii)	Sound energy from 1/3 scale model as above, but concrete block shielded	$\sim 4.4\times10^{-2}$ J unweighted

Comparing (i) and (ii) shows that the sound energy ratio ($3.9/6.7\times10^{-2}$) or 17.6 dB compares reasonably well with the input energy ratio (872/9.78) or 19.5 dB. The void around the real machine is covered by steel plates making a direct comparison difficult, but it appears that noise radiation from the foundation at low frequencies, hitherto ignored, requires further investigation.

A comparison of vertical traverses of peak pressure is shown in Fig 3. The pattern in each case is similar and both increase by some 10dB near the impact plane where the peak pressures are probably due to the vibration of the dies and dieholders. Scaling predicts a difference of 11dB in this area whereas higher up where acceleration noise dominates a lesser difference of 6dB is expected.

The decay of peak pressure with distance (through the operator position) shows the source to be a relatively small area (Fig 4) since the inverse square law operates to within 0.1m.

The noise from forging blows is generally less than that from hard die blows since the rate of change of force is reduced. For forging blows on the model (a) the same frequencies appear as for die to die blows but with varying amplitudes and (b) the noise is reduced at all frequencies except the lowest, with the greatest reductions at high frequencies.

Work is currently in progress to identify the causes of the remaining peaks in the spectrum of Fig 2. These cannot be related to any of the measured natural frequencies of the components in isolation and other causes are being sought. In addition to the mass-spring-mass modes of the die-dieholder assemblies, a further system of masses and springs has been identified, that of the combined mass of the lower die and sow acting on the residence of the poppets and associated stress concentrations. The lower die/sow assembly is located on the anvil by four poppets and screws and the shaft of each poppet is wedged into a hole in the anvil (Fig 5). The poppets themselves may be considered as springs, the screws as cantilevers in addition to the stress concentrations beneath the poppet screws and at the load points each end of the poppets. Fig 6 shows measurements of point mobility on the top of the lower die (vertical response) as the poppet screws are tightened from zero to the given torque, but the die/sow wedge kept tight. 'Low' frequency modes appear in which the die/sow assembly vibrates as a solid mass and it is likely that these are the cause of some of the peaks in the same frequency range in Fig 2.

It has not yet proved possible to drop the tup onto the anvil with adequate alignment (essential to prevent damage to the dies and retain a constant contact duration) and so establish a baseline for noise.

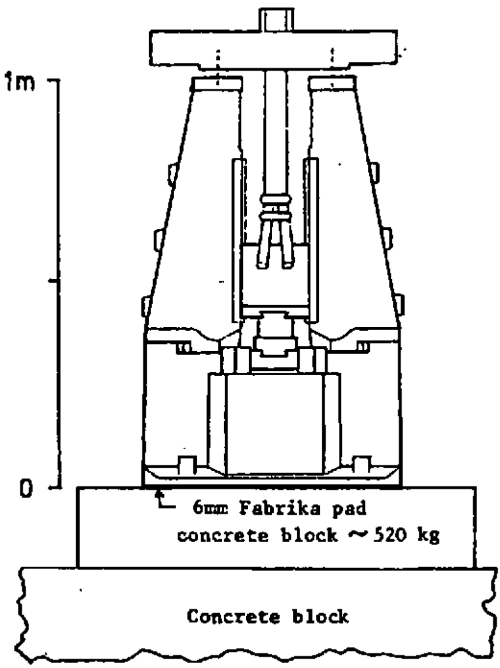

FIG 1: One third scale model drop hammer

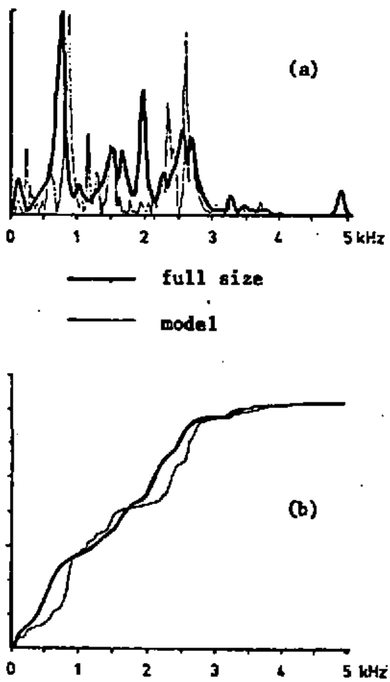

FIG 2(a) Sound pressure spectra for model and real machine
(b) integration of the above.

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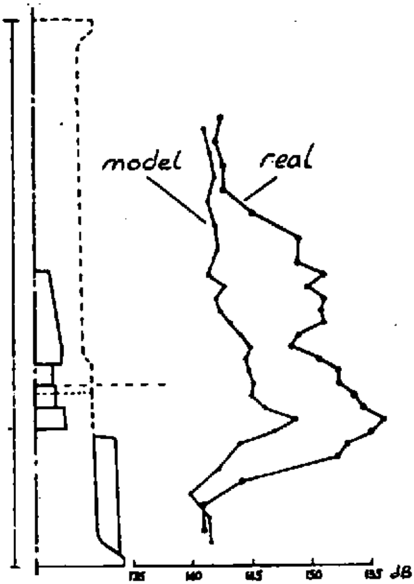

FIG 3: Vertical traverses of peak pressure (0.5mm distance)

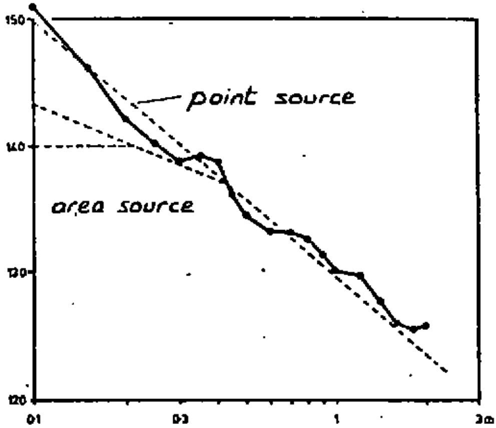

FIG 4: decay of peak sound pressure with distance

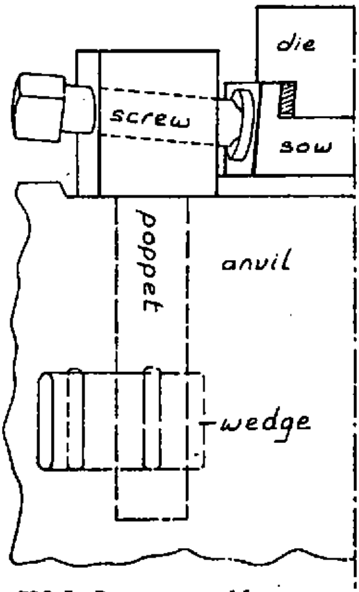

FIG 5: Poppet assembly

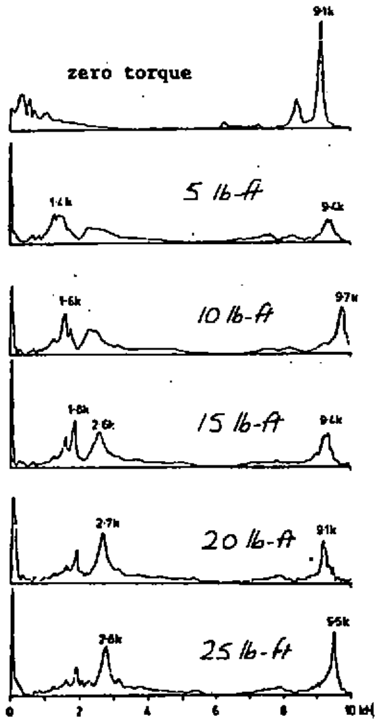

FIG 6: Die mobility as poppet screws are tightened