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THE USE OF A SCALE MODEL TO INVESTIGATE THE NOISE SOURCES ON A 200 TONNE POWER PRESS

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INTRODUCTION

The noise radiation characteristics and control techniques on a 200 tonne power press are being studied as part of a joint ISVR/CETIM investigation into methods of reducing the noise from industrial presses. Because of the physical size of the press (∿4.5 m high) its capacity and location in a workshop environment detailed measurements present many problems, and all but the most simple tooling and structural modifications are completely impractical. In order to avoid these difficulties a simplified scale model has been built. This allows the investigations to be conducted much more simply in the laboratory and facilitates the effects of extensive structural modifications to be studied.

MODELLING THE PRESS

The full scale press under investigation is a BRET 200-tonne straight-sided, mechanical crankshaft driven press of conventional construction. The main frame structure is fabricated from a number of welded steel panels and was thus relatively simple to model. As the investigation is centred around the noise related to the workpiece material fracture event (i.e., press excitation is related to the tooling and material properties) it was not necessary to model the complex drive mechanism of the press. An alternative simplified operating system consisting of a hydraulic ram and hand pump was used. The model was fitted with an existing tool set to punch a single 20 mm diameter hole. Details of the model are shown in fig. 1.

An overall scale of 1/3 on outer dimensions was used, which gave a model height of approximately 1.5 m. However, in order to attain a practicably low working capacity, a scale of 1/10 was used for material thicknesses. This gave a model capacity of 6.6 tonnes for equivalent stress levels in the side frames.

The 1/3 width and length scales and 1/10 thickness scales are useful scalings to use acoustically as panel modes in the actual and model press occur at approximately similar frequencies. The coincident frequencies of the model are 10 times higher, however, thus giving very different noise radiation characteristics below the critical frequencies.

Damping loss factors were measured on the panels of the completed model and compared to values obtained from the actual press. In the majority of positions, the loss factors were very similar, only on the four columns was there any significant variation. This was anticipated as each column on the model was formed from bends in one piece of sheet steel rather than a welded fabrication as on the actual press. This led to lower values of loss factor on the model which were then matched to those on the actual press by the application of strips of damping tape.

MODEL RESULTS

The initial investigation of the model was to study the noise radiation characteristics of the structure and to compare these with measurements made on the actual press. The noise energy radiated from the individual panels of the press was estimated from measurements of surface vibration. A spatial average of the transient surface velocity ($\langle v\rangle^2$) was measured and the radiated noise energy (E_{rad}) calculated from the well-known relationship:

$$E_{rad} = \rho.c.s.\sigma_{rad}.\langle v\rangle^2.T$$

(where ρc = characteristic impedance of the air; s = surface area of the panel, σ_{rad} = radiation efficiency of the panel; T = measurement record length).

The measurements were made using a signal analyser and the calculations performed by computer, theoretical values of radiation efficiency being computed from the formulae for simply-supported plates quoted in ref. 1. The results showed that the bed, side plates and columns were all major radiators of noise.

Concurrent measurements of near field sound intensity from the model were also performed and these are detailed in the current paper by Watkins. Overall agreement of the total radiated sound energy obtained by both methods was very good but some discrepancies were apparent when more detailed comparisons in frequency bands were made. The two sets of measurements are to be used to obtain values of actual radiation efficiency for the individual panels of the model.

Estimates of noise radiation from the actual press can be obtained from the model by applying the panel dimensions and other scaling factors from the actual press to the surface vibration measurements of the model. Selected results were compared to measurements made at similar locations on the actual press. Very good agreement was obtained as is shown in fig. 2.

A major objective of the model is to study the flow of vibrational energy around the press structure. This is with a view to reducing radiated noise levels by inhibiting transmission of vibrational energy to the major radiating components and also by increasing energy dissipation by increasing structural damping. Increasing the structural damping in machinery components can pose a number of practical difficulties, conventional damping treatments often being of little use. Other more practical treatments are the subject of current research at the ISVR. Filling hollow sections with granular materials has been shown to be a very effective method [2]. Filling the hollow columns of the model with sand increased their damping loss factor by a factor of 30, which is equivalent to a reduction of 15 dB in the noise radiated from these components.

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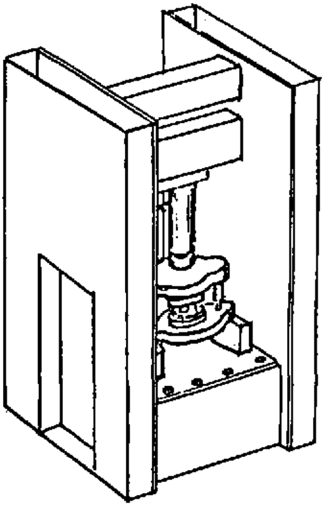
Fig. 1
Model of press



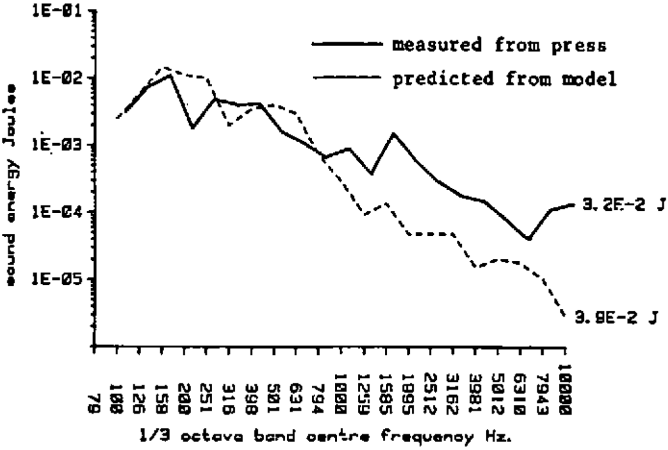


fig 2 Sound energy from press bed

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ACOUSTIC INTENSITY MEASUREMENTS ON A 350kN AUTOMATIC PUNCH PRESS.

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INTRODUCTION

Automatic punch presses continue to be major noise sources contributing to high noise levels in the metal-working industry. Recent advances in instrumentation, such as the acoustic intensity method, as well as in computer data analysis, have enabled studies to be made which may lead to control of punch press noise at the source. The work described here on an automatic punch press follows a similar study on a hand-press study by the present authors [1]. In that study it was believed that it was established for the first time that noise radiation from a C-frame press was stiffness controlled. A noise reduction of 6.8 dB (lin.) was achieved by increasing the frame static stiffness by 2.7 times. In contrast, only small noise reductions of order 2 dB have been reported for automatic presses using increased mass or damping [2, 3]. These findings have given added credence to the automatic press investigation reported here which followed similar methods to that used in [1] but with the added use of acoustic intensity measurements in order to establish if the press frame was the dominant noise radiator and to help verify that increase of frame stiffness is required to make significant noise reduction.

EXPERIMENTAL METHOD

The suspended floor in the anechoic chamber at the Center for Sound and Vibration, N.C.S.U. was removed to accommodate a Bliss 350kN 1831 automatic punch press. Fiberglass panels and foam wedges were replaced on the main floor around the press to minimize sound reflections off the floor. Press operation was at approximately 200 impacts/min. and previous studies had confirmed that changes in tooling and the die area resulted in only small reductions in noise levels order 1-2 dB. A magnetic pickup was mounted on the press crankshaft in order to provide a trigger for transient signal analysis. A typical time-history of the noise pressure is given in Fig. 1 which shows three separate events in the press action, namely 1) blanking and impact of the punch with the draw rings, 2) ram reversal at bottom-dead-center and 3) impact of the die springs against the bolster plate as the press opens. The dominant peaks in the noise spectrum were in the frequency range 100 Hz to 1200 Hz, see Fig. 2. Noise source identifications were made using a variety of methods. Acoustic intensity measurements using a two-microphone probe in conjunction with a Nicolet 660B two-channel analyzer were made over a 25 point hemispherical grid at radius r = 3m around the press. The complex mathematics hardware on the Nicolet allowed corrections for phase errors resulting from microphone mismatch and FM tape-recording of signals.

DISCUSSION

A typical acoustic intensity result is plotted in Fig. 3 for the arc from the back to the front of the press. Interestingly the overall intensity levels measured in the anechoic space were nearly uniform around the press to within ± 2 dB. This suggests that although certain components radiate at dominant frequency peaks the main source of radiation was likely to be the large press frame as it absorbs mechanical energy on the downstroke of the ram and then radiates noise as the C-frame unloads after bottom-dead-center. Two-point transfer functions of surface vibration between the die area and various components were obtained using a PZB impedance hammer and accelerometer kit and frequency comparisons made with the peaks in the acoustic intensity spectra. Near-field acoustic intensity measurements were also taken over several components which might be probable noise sources. Using these methods, Table I summarizes the correspondence between vibration frequencies and noise frequencies at which various sources radiate positive acoustic intensity.

In order to ascertain the importance of a particular component's contribution to the transient radiated sound energy, partial coherent output energy spectra were processed using a four-channel Biomation A/D convertor system and a Data General AP130 Eclipse computer following the methodology described in [4]. Three-input, single-output measurements were made with accelerometer signals as input and the noise at the operator position as the output. For instance, it was found that the tinplate component in the die area contributed over 50% of the noise energy during the third event. However of more significance, as shown by Fig. 4, is that areas of the press frame have significant partial coherence at many frequencies. Interestingly it was found by further signal processing that removal of the dominant peaks in the noise spectrum reduces the overall noise level by only 1-2 dB, which suggests significant noise radiation was produced by the broadband vibrational energy.

As confirmation of the fact that the press frame was the likely dominant noise radiator, finite-element studies were made of the press which showed that the cross-members between the side-plates were likely to have strong vibrational energy. The input forcing function as measured on one of the two press rams is shown in Fig. 5 together with its energy spectra in Fig. 6 which is in the radiated noise frequency range of 100 Hz to 1200 Hz. Following the methodology described in [1], the transient response to this measured forcing function was calculated treating the press as a single degree of freedom system. Fig. 7 shows the effect of increasing the press mass and stiffness by 2.5 times on the response to the input-forcing function of Fig. 5. This figure demonstrates that the press response is more stiffness controlled than mass controlled. The stiffness change increases the fundamental resonance frequency while reducing its amplitude by 6 dB. This suggests that like the hand press one method of punch press noise reduction in the future is the redesign of new presses with greatly increased stiffness. It is probable that such redesign could be efficiently achieved by modern computer-aided design and finite-element methods.

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Table I.

Frequency Comparisons of Spectrum Peaks

	Noise Spectrum Frequency (Hz)																																									
	175	195	270	295	325	345	370-5	440	465	485	500	510	525	560-5	575	585	595	610	620-5	660	715	730	745	755	770	785	800-5	815	825	830	840	855	865	885	935	945	960	975	995	1005	1020	1030	1050
Exit Chute Related Linkages				•	•			•					•	•	•	•	•	•	•	•													•		•	•	•	•	•			•	
General Spring Area	•									•	•								•			•	•	•	•	•	•		•	•	•	•					•	•				•	•
Flywheel Cover (Two-Mic.)	•		•	•	•	•	•	•	•	•	•		•		•		•	•	•	•																							
Flywheel Cover (Mic./Accel.)	•	•	•	•	•	•	•					•	•						•														•										
Side Plate (Spring Area)							•																																				
Side Plate (Exit Area)							•											•	•		•																						
Bolster Plate	•		•		•		•	•	•	•	•					•		•					•	•	•		•	•	•	•		•								•			

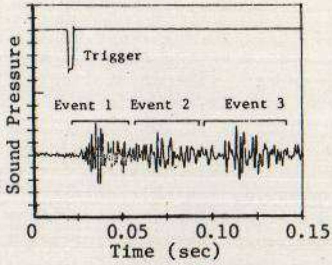


Fig.1. Noise Time History

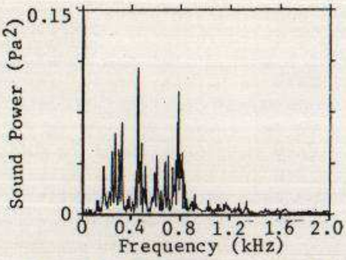
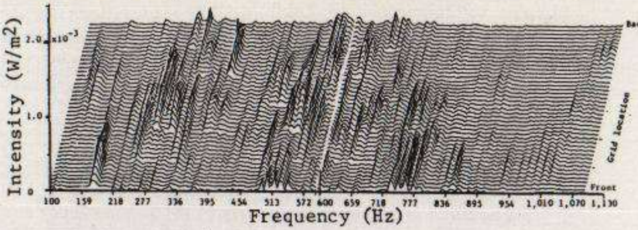


Fig.2. Noise Power Spectrum

Fig.3.
Contour Plot
of Acoustic
Intensity
at r = 3m



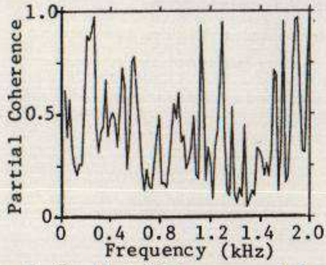


Fig.4. Partial Coherence of X-member

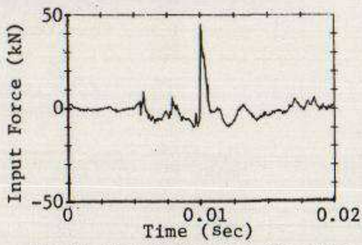


Fig.5. Input Forcing Function

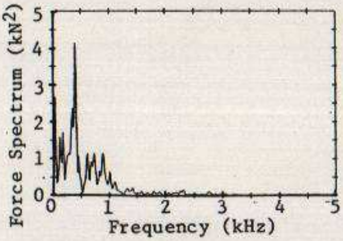


Fig.6. Input Force Spectrum

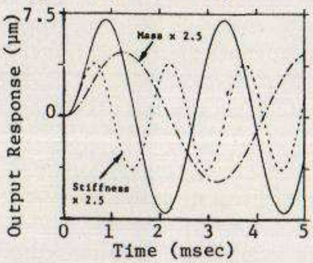


Fig.7. Single DOF Response