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USING THE ENERGY ACCOUNTANCY EQUATION TO MODIFY PUNCH DESIGN IN PRESSES

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Noise regulations and the design of machinery

Estimations of overall noise efficiency are difficult to give since the measurement of the mechanical energy used by the press is not easy to define, let alone to measure. Flywheel machines take the power from the flywheel, which slows down, but much of this is absorbed in accelerating the mechanism and in other ways. However, we can estimate that, of the flywheel energy which is available, some 0.0001 goes into noise. This is a small fraction and far too low to consider using for any other purpose. Thus, once again, the argument that noise is wasted energy cannot be a valid reason for noise reduction. The reason for reducing noise must hinge on the need to meet deafness limits, and must be related to the 85 or 90 dB(A) limit on $L_{Aeq,8L}$. This is an average radiated acoustical energy reaching the ear of an operator and must be related to the average energy radiated from the whole machine during the same period of time. Knowledge of the frequency of the noise concerns us only in the sense that an A-weighting with frequency must be added to reflect the difference in the subjective acceptability of noise of different frequencies.

How, therefore, can we introduce noise regualtions into design in such a form as to relate easily to the design parameters of the press? The noise is related to the strain energy built up in the press just prior to material fracture and the way that, on fracture, this strain energy is converted to vibration over the whole machine at various frequencies. The total radiated sound energy per impact

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will be given by

$$\omega_{\text{rad}} = \varrho_0 c \sum S \tau_{\text{rad}} \int_0^T < v^2 > dt \tag{1}$$

over the surface of each member.

As seen in Fig. 1, the measurement or calculation of vibration levels in parts of the machine well away from the workplace lead to a poor diagnostic method

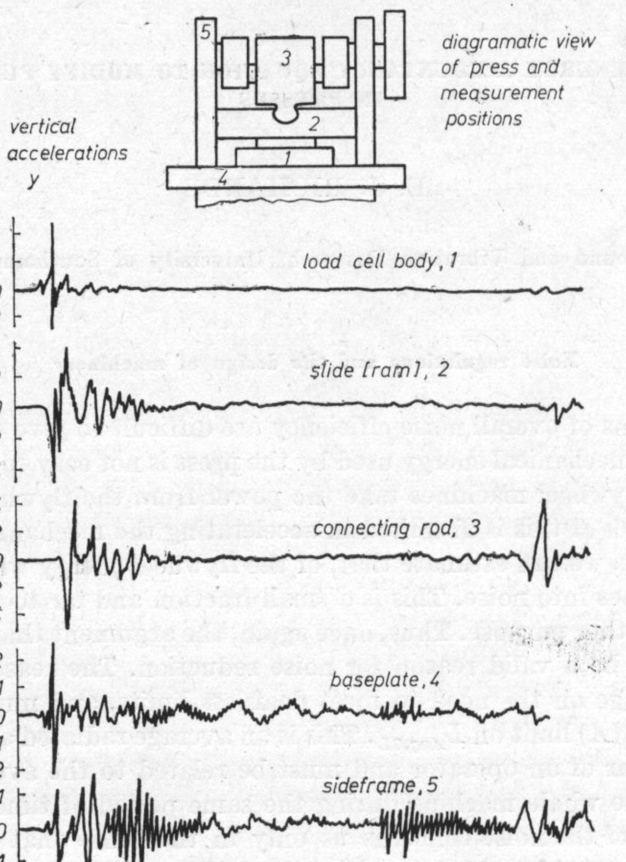

Fig. 1. Progress of vibration through a press

and the use of the energy accountancy equation in its most convenient form carries more promise [1], the impulsive force concerned being the one most easily measured, i.e. the one exerted by the punch itself.

Alternative forms of the energy accountancy equation

The Energy Accountancy Equation [1] is usually formulated in terms of the $L_{A\mathrm{eq}}$ created in a frequency band Δf centred around a frequency f_0 but this is largely because of the need to make allowance for A-weighting of the sound with frequency. There are several other alternative forms which are more convenient to use in practical applications. This is particularly so if the structural modifications being made to the whole of a machine are of a minor character and the noise control measure being taken can be considered as simply an alteration to the force pulse shape $f(t)$ with no alteration to the structural response term nor the acoustic terms.

When this force pulse $f(t)$ is transferred to the frequency (f_0) domain by the Fourier transformation

$$F(f_0) = \int_{-\infty}^{\infty} f(t)\, e^{-j2\pi f_0 t}\, dt \qquad (2)$$

the force pulse $F(f_0)$ becomes a frequency weighted time average of the time average of the time pulse force centred around what we call a frequency (f_0). The term in the Energy Accountancy Equation, $10 \log |F(f_0)|^2$ is not as easy to visualise physically in terms of the operation of the mechine as would a pulse term indicating the variation of the force with time. It is therefore good sense to rewrite the Energy Accountancy Equation in the time domain if this can be achieved.

The noise regulations are drawn up in terms of the total radiated noise energy emitted per day and we can relate this to the noise energy per machine operation by dividing by N, the number of operations per day.

In its original form prior to taking logarithms and expressing it in decibel terms, the Energy Accountancy Equation is written

$$\omega_{\mathrm{rad}}(A, f_0, \Delta f) = \sum_{f_0}^{f_0+\Delta f} \frac{\varrho_0 c \Delta f}{2\pi^2 \varrho_m f_0} N \frac{A\, \tau_{\mathrm{rad}}}{f_0} \frac{1}{\eta_s d} |\dot{F}(f_0)|^2 \mathrm{Im}\, H(f_0) \qquad (3)$$

and the term $10 \log |F(f_0)|^2$ can be taken out of the equation and returned to its time integrating form only if it is constant with frequency, at least in the range of frequencies in which the noise energy is greatest.

This may not be possible, but there will always be some derivative of $|\dot{F}(f_0)|^2$ which will be effectively flat with frequency, and the secret is to examine the impulsivity of the motion and to find a force derivative which is flat.

Looking at the expression for the Fourier transform of $f(t)$ above, we see that we have the freedom of choosing whichever derivative we use by recognizing that

$$\dot{F}(f_0) = 2\pi j f_0 F(f_0) \quad \text{and} \quad \ddot{F}(f_0) = -4\pi^2 f_0^2 F(f_0), \quad \dddot{F}(f_0) = 2\pi j f_0 \ddot{F}(f_0) \text{ etc.}$$

It follows that we can replace $10 \log |\dot{F}(f_0)|^2$ by $10 \log |\ddot{F}(f_0)|^2 - 10 \log (4\pi^2 f_0^2)$ and we can write the Energy Accountancy Equation in the alternative noise energy form (not logarithmic)

$$\omega_{\text{rad}} = \frac{\varrho_0 c N}{8\pi^4 \varrho_m d} |\ddot{F}(f_0)|^2 \sum_{f_0=0}^{\infty} \operatorname{Im} \frac{H(f_0)}{f_0} \frac{A}{f_0} \frac{\tau_{\text{rad}}}{f_0} \frac{\Delta f}{f_0} \frac{1}{\eta_s} \tag{4}$$

and if $|\ddot{F}(f_0)|^2$ is constant with frequency, we can write $10 \log \omega_{\text{rad}}(A) = 10 \log |\ddot{F}(f_0)|^2 +$ other terms which do not depend upon pulse shape but are dependent upon the structural response, radiation efficiency and A-weighting.

The same would apply if we were to use the third or any derivative of $F(f_0)$ except that the other non-pulse-dependent term would be modified to accentuate the lower frequency terms.

Which form of equation to use depends upon the degree of physical insight required. Transferring back to the time domain we have

$$|\ddot{F}(f_0)|^2 = \ddot{F}(f_0)\ddot{F}(f_0)^* = \int_0^\infty \ddot{f}(t)\, e^{-2\pi j f_0 t}\, dt \int_0^\infty \ddot{f}(t)\, e^{2\pi j f_0 t}\, dt$$

and if $f(t)$ is only significant for a series of short times of length Δt this equals

$$\sum [\ddot{f}(t)_{\max}]^2 (\Delta t)^2 = \sum [\dot{f}(t)_{\max}]^2$$

while

$$|\ddot{F}(f_0)|^2 = \sum [\dot{f}(t)_{\max}]^2 \quad \text{and} \quad |\dot{F}|^2 = \sum f^2(t)_{\max},$$

where Σ implies a summation of the peaks in the values of the relevant function with time.

Since noise is often greatest in the medium frequency range because the modified radiation efficiency is greatest there on machinery type structures, a good compromise is to take the form $|\ddot{F}(f_0)|^2$ or $\Sigma[\dot{f}(t)_{\max}]^2$.

Thus, a useful form of the Energy Accountancy Equation when applied to tool design is

$$L_{A\text{eq}} = 10 \log \sum [\dot{f}(t)_{\max}]^2 + \text{const} = L_{\dot{f}} + \text{const}. \tag{5}$$

More important is the fact that rates of change of force are the highest derivative of the force which can be visualized diagnostically, and that errors arising from its non-constancy in the frequency domain are less important than the loss of conceptuality arising from using the wrong derivative.

Needless to say, the derivative to be used must also depend upon the variation of the structural response term. The higher the derivative used, the greater the fall-off with frequency in the relevant structural response factor, $\operatorname{Im}[H(f_0)/f_0^n]$ and the more the constant term in equation (5) will reflect the low frequency behaviour of the structure: each form must be taken on its merit in the particular case and the instrumentation which is available.

In all cases, the original form of the Energy Accountancy Equation in which everything is related to the frequency domain is the most reliable, though not necessarily the most useful. In practice, as frequency can be related to $(1/2)t_0$, we have found that good diagnostic skill can still be developed in the frequency form of the equation by recognizing that high frequency content can be equated to sharp temporal changes, except possibly in interpreting acceleration noise.

Tool design to reduce noise

While it may be possible to add internal damping (2) to a press at the design stage, the opportunities for doing so on existing presses are not great, especially as the damping has to be increased greatly in the vicinity of the punch as well as in the columns, a region whose design depends upon so many other parameters.

Increasing the stiffness by a useful amount is not very practical either, since the radiated noise level will vary as 10 log k_1/k_2, i.e., the stiffness will need to be doubled for even a 3 dB noise reduction.

Changes int ooling design to reduce the rate of change of force both in the time and the frequency domain bears the greatest promise of practical improvement as such modifications are simple and relatively easy to implement. Tool sets are compact and are not in themselves high noise radiators unless they are badly designed and give rise to ancillary impacts. They can be looked upon therefore as a means of modifying the pulse shapes without alteration to the other characteristics of the machine.

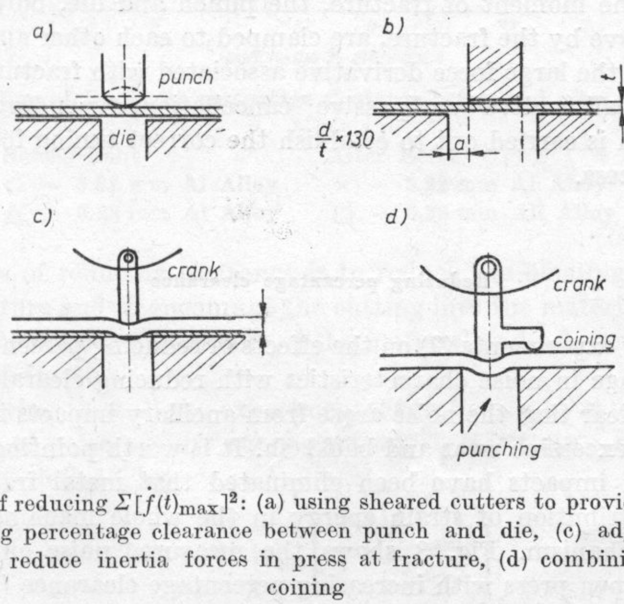

Fig. 2. Methods of reducing $\Sigma[\dot{f}(t)_{\max}]^2$: (a) using sheared cutters to provide longer fracture time, (b) reducing percentage clearance between punch and die, (c) adjusting maximum cutting depth to reduce inertia forces in press at fracture, (d) combining punching with coining

Figure 2 shows some of the many tool modifications that can be made to reduce the force derivatives and therefore the noise radiated. These may be listed as follows:

(1) Reducing the percentage clearance between the punch and die sections of the punch set so as to increase the cutting into the workpiece prior to fracture and thus reducing the size of the fracture force derivative. The disadvantage lies in the necessity for more accurate tooling and increased heating and wear of the punch.

(2) Providing a small degree of shear in the punch or die in order to encourage a progressive fracture around the periphery of the cutter. This provides a longer cutting pulse and a smaller force derivative. The objection to this is the poor finish and the possible need for a further operation to eliminate distortion of the component.

(3) Raising the tool bottom dead centre to give a reduced inertia force following fracture. The ideal would be for the punch to begin to lift off the workpiece at the instant of fracture so that the strain energy which needs to readjust itself in the frame is kept to a minimum. The disadvantage lies in the greater risk of incomplete fracture and the problem of ejecting the blanks.

(4) Coupling the fracture to some amount of coining, e.g., to halt the punch via a relatively slow surface forming (coining) stage. This can be advantageous in certain instances where forming is needed, but there are obvious limitations on when this can be used.

(5) Combining several operations so that the overall rate of change of force is minimized. This needs very careful construction of multiple punch tool sets to optimize the fracture sequences.

(6) If, at the moment of fracture, the punch and die, both of which are made free to move by the fracture, are clamped to each other and then allowed to move slowly, the large force derivative associated with fracture can be minimized. This we call "active" or "passive" cancellation, depending on the degree of sensing which is carried out to establish the correct timing for the clamping or freezing process.

Reducing percentage clearance

In our early experiments (2) on the effects of reducing percentage clearance, little or no change in noise characteristics with reducing clearance was found, and it became clear that the noise arose from ancillary impacts in the bearings associated with excessive wear and backlash. It is worth pointing out that it is only when such impacts have been eliminated that metal fracture and the associated redistribution of strain energy in the whole machine becomes the basic noise mechanism. Fig. 3 shows the measured noise output from the ISVR C-frame open press with increasing percentage clearance (defined as the average punch gap, divided by the material thickness and multiplied by one hundred) before and after the bearings had been replaced. It may be seen that with poor bearings, the noise reduction achieved by reducing the clearance is minimal. What follows describes noise reductions on well-maintained punch presses.

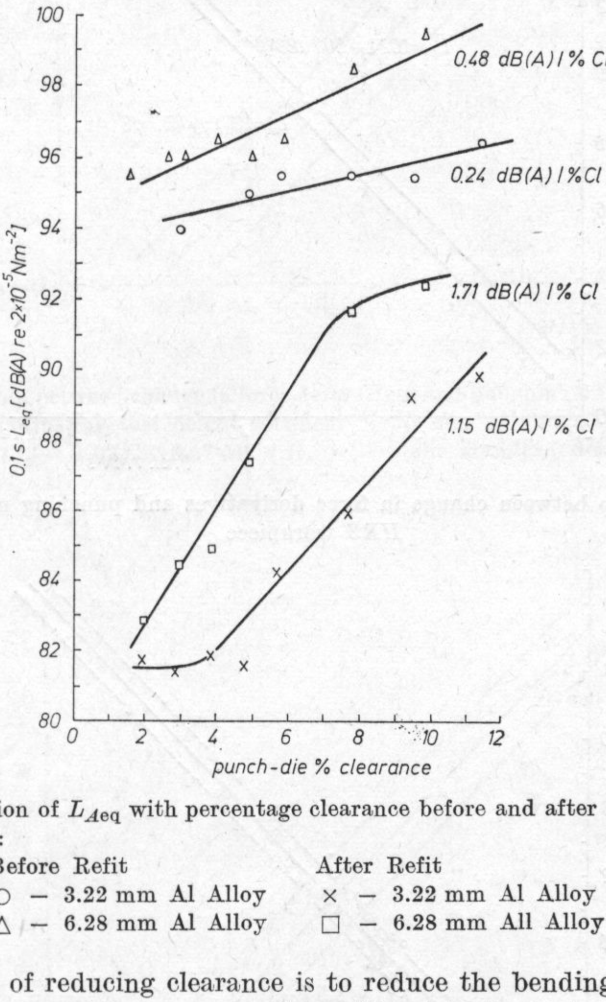

Fig. 3. The variation of L_{Aeq} with percentage clearance before and after elimination of backlash in bearings:

Before Refit	After Refit
○ – 3.22 mm Al Alloy	× – 3.22 mm Al Alloy
△ – 6.28 mm Al Alloy	□ – 6.28 mm All Alloy

The effects of reducing clearance is to reduce the bending moment in the vicinity of fracture and to encourage the cutting into the material by the punch. This reduces the effective thickness of the material and a lower punch force at the moment of fracture.

It is gratifying that all the experimental results can be collapsed on a straight line curve of noise against our parameter $L_{\dot{f}}$ this time expressed in terms of a force level defined by

$$L_{\dot{f}} = 10 \log_{10}\left[\frac{\sum\left(\dot{f}(t)_{\max}\right)^2}{f_R^2}\right],$$

where f_R is the reference force derivative of 1 Mn·s^{-1}.

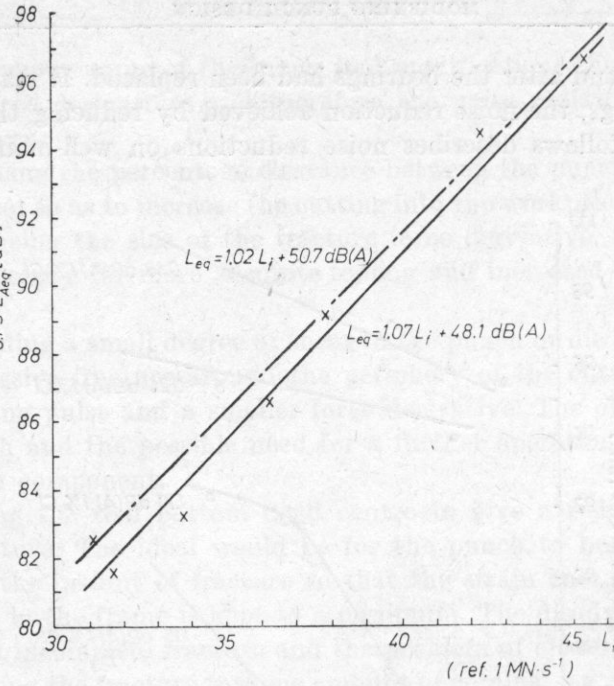

Fig. 4. Relationship between change in force derivatives and punching noise for 3.01 mm *HRS* workpiece

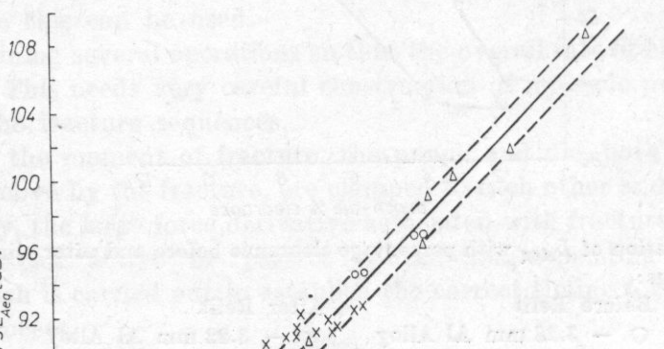

Fig. 5. Relationship between change in force derivatives L_j and punching noise L_{eq} for all measurement conditions. Workpiece material: × — aluminium alloy, ○ — hot rolled steel, △ — bright drawn steel, ——— $L_{eq} = 1.02\ L_j + 50/\mathrm{dB}(A)$, – – – – one standard derivation limits. Best straight line fit to all measurement points (except those correspnding to $L_j < 28$ dB) $L_{eq} = (1.02 \pm 0.025)L_j + (50.7 \pm 1.0)\,\mathrm{dB}(A)$

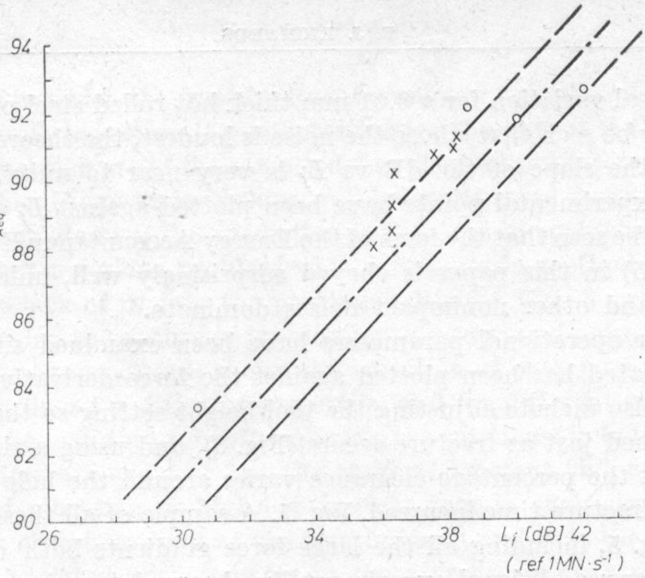

Fig. 6. Relationship between change in force derivatives and punching noise for measurements on the effect of adjusting tool height settings: × 3.2 mm Al alloy; ○ 6.28 mm Al alloy; —·—·— $L_{eq} = 1.02 L_{\dot{f}} + 50.7$ dB (A), — — — one standard derivation limits

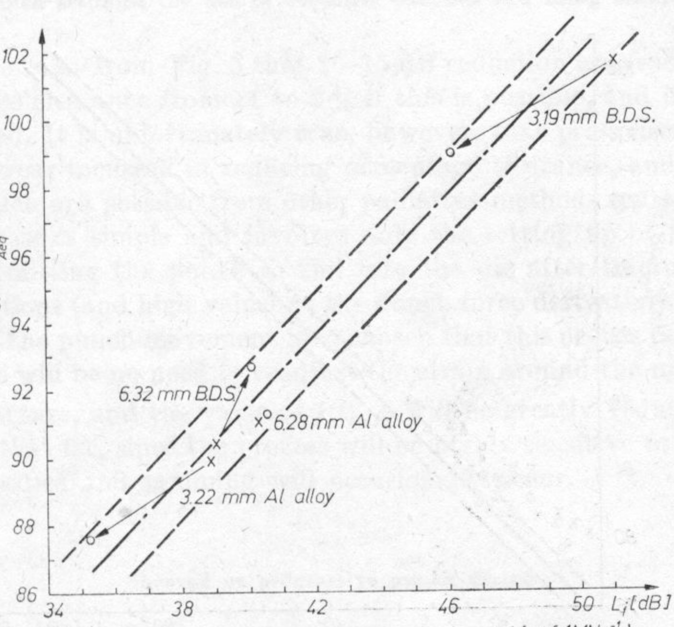

Fig. 7. Relationship between change in force derivatives and punching noise for measurements on the effect of an eccentric die × concetric die; ○ eccentric die; —·—·— $L_{eq} = 1.02 L_{\dot{f}} + 50.7$ dB (A); — — — one standard derivation limits

One typical variation for a 3.01 mm thick hot rolled steel plate is shown in Fig. 4. It may be seen that where the noise is loudest, the theoretical prediction is good and the slope of the dB vs $L_{\dot{f}}$ is very near to unity.

All the experimental points have been plotted against $L_{\dot{f}}$ on one graph in Fig. 5. It may be seen that the form of the Energy Accountancy Equation derived as equation (5) in this paper is obeyed surprisingly well, unless the fracture noise is low and other nonimpact noises dominate.

The other operational parameters have been examined similarly (3) and the noise radiated has been plotted against the force derivative factor. These experiments also include adjusting the tool height setting so that bottom dead centre is reached just as fracture occurs (Fig. 6), and using a slightly eccentric punch so that the percentage clearance varies around the hole to be punched and a longer fracture time incurred, Fig. 7. A sample of all these results is also plotted in Fig. 5, including all the large force gradients both during fracture build-up and any additional experimentally observed transients occurring due to backlash. It may be seen that there is good agreement between the values of L_{Aeq} averaged over a 0.1 s period, and $L_{\dot{f}}$ the standard deviation of one decible from the best fit line which in turn implies a 2 per cent greater slope than the theoretically derived unity, and a standard deviation about this line of 0.025 $L_{\dot{f}}$.

Needless to say, the agreement is less acceptable, particularly for thin materials and low values of $L_{\dot{f}}$. It may be seen that while agreement is good for

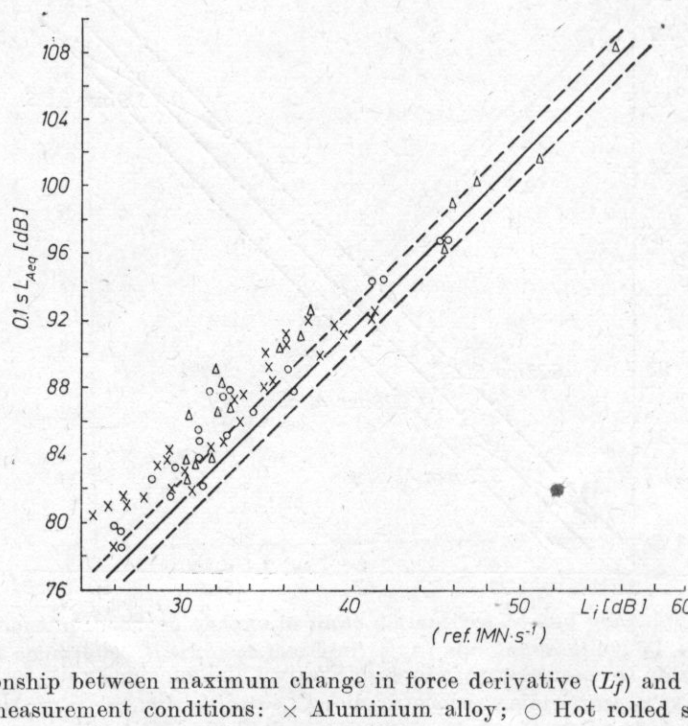

Fig. 8. Relationship between maximum change in force derivative ($L_{\dot{f}}$) and punching noise (L_{eq}) for all measurement conditions: × Aluminium alloy; ○ Hot rolled steel; △ Bright drawn steel, ——— $L_{eq} = 1.02\ L_{\dot{f}} + 50.7$ dB (A), – – – – one standard derivation limits

noisy processes in which the fracture forces dominate the noise output, the noise prediction formula (equation (5)) underestimates the output if the punch force build-up and backlash are of the same order as the fracture forces. This situation often occurs when the press is working well below its rated maximum force, and under such circumstances, the prediction is poor. As in such circumstances the total noise is not a serious problem, this deficiency in method is not serious.

In fact the lack of fit into the prediction line of Fig. 8 provides a useful method of checking the nature of the dominating source of noise for that condition. Table 4 (not reproduced) shows the variation of both best fit gradients of such prediction lines for different materials, and for different percentage clearances. It may be seen that for soft materials the relationship between noise and fracture force derivative is poor but the residual constant "b", representing the general level of vibration, is high. It is clear that in such circumstances the impact point is elsewhere than at the fracture area and that the structural response terms associated with impacts in these areas are greater.

Predictions are still possible, if the full strength of the energy accountancy equation is incurred, just as they have been using the transfer mobility methods used in (4) to deal with the noise radiated from the surface of a diesel engine frame subject to combustion forces on the piston.

Altering punch settings, the use of eccentric punches and using sheared cutters

It may be seen from Fig. 5 that 10–15 dB reduction are feasible by reducing percentage clearance from 11 to 2 % if this is possible, and if the press is well-maintained. It is unfortunately true, however, that press users to not like the excessive wear incurred in reducing percentage clearance, and the scope of reductions which are possible from other palliative methods must be explored.

One of these is simple and involves only the setting up of the punch; if instead of permitting the punch to run into the die after fracture, incurring sharp accelerations (and high values of the punch force derivatives), the bottom dead centre of the punch movement is so chosen that this occurs just as fracture happens, there will be no need to readjust the strain around the machine at the moment of fracture, and the value of $\dot{f}(t)_{\max}$ will be greatly reduced. It is not feasible to go that far, since the process will be highly sensitive to variations in material properties and jamming will occasionally occur.

Sheared or progressive punch design

Having established firmly that in their most noisy conditions punch press noise is directly related to the time rates of change of punch force and that once such large rates have been designed into tooling, reduction is extremely difficult to achieve, consideration needs to be given to minimizing such force derivatives at the design stage by deliberately lengthening the fracture process by encouraging progressive fracture around the hole to be punched or by overlapping sensibly and integrating the punching of several holes and coining operations.

This can be done (a) by making less uniform the percentage clearance between the punch and its cavity around the hole by fitting the punch eccentrically: (b) by shearing the cutter by a small amount so that the fracture travels progressively around the hole: or (c) by selecting the progression of cutting times of different holes in such a way that the total load is less impulsive.

Experiments on all these possibilities have been carried out at ISVR and are reported here, not so much to provide a detailed record, as much as to indicate the promise to be obtained using such pallative methods, and to raise suggestions regarding tool design.

Experiments on the effects of sheared punches have been more successful than any of the previous methods. Shear cutting is often used as a means of reducing the maximum punch load both on a single cutter and on a multi-punch tool involving progressive cutting of a series of single holes in the same action.

It is interesting to examine the theoretical design associated with shear, since the need is for a uniform reduction of the punch force derivative rather than to space the fractures and individual events. Such calculations have been made

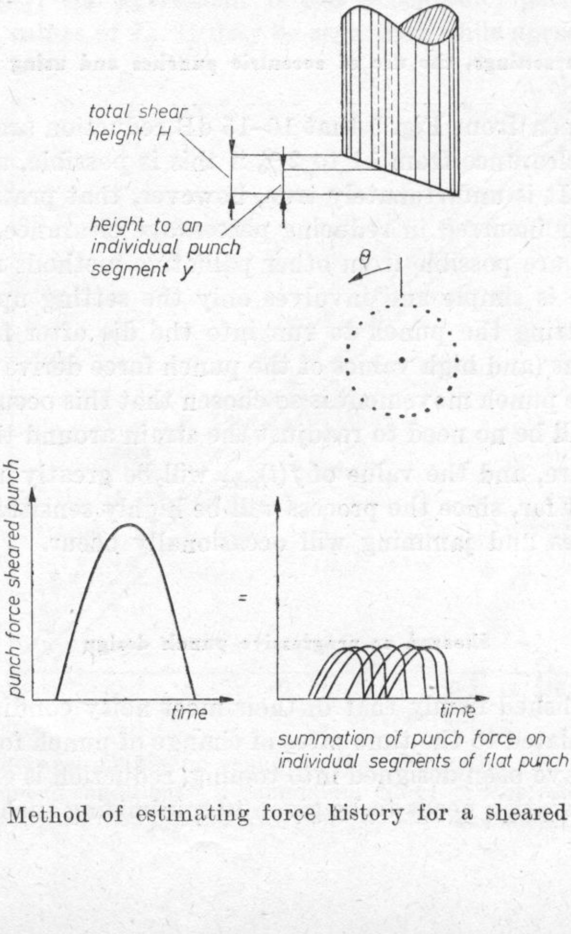

Fig. 9. Method of estimating force history for a sheared punch

for the sheared cutter shown in Fig. 9, each fracture around its periphery being assumed to occur at delayed times determined by the displacement. Fig. 10 illustrates the resultant calculated force history for varying shear angles. It may be seen that there is little point in going beyond three or four degrees of shear for the case being considered. This is illustrated more clearly in a study of the estimated values of the force derivative parameter L_j shown in Figs 11(a)-(d).

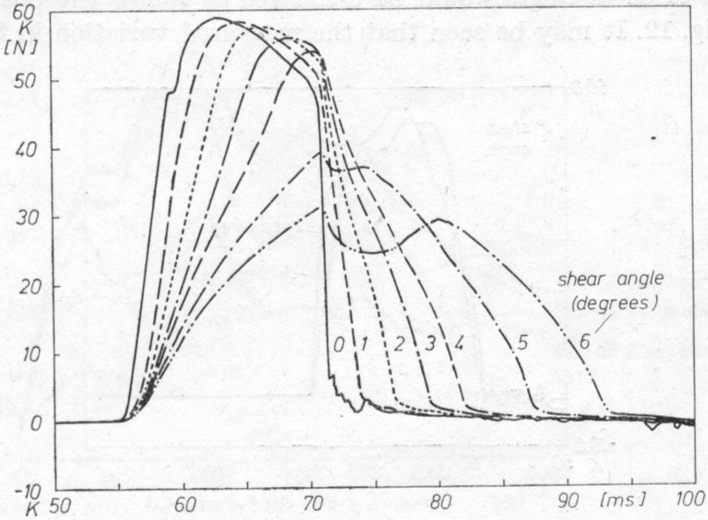

Fig. 10. Derived force histories for various punch shear angles

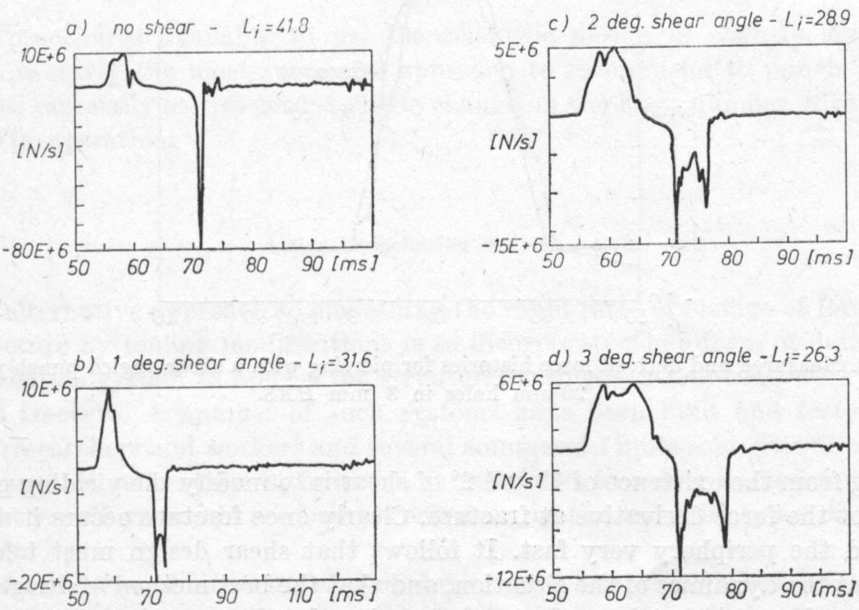

Fig. 11 (a)–(d). Estimated punch force derivatives for sheared punch

The estimated value of L_f falls from 41.8 to 31.6 for one degree of shear angle with very little worthwhile additional reduction below three degrees of shear. As the use of such small angles gives no great workpiece distortion it is thought that small shear cutting tools may be used with no further work needed.

If fracture were to occur according to the calculated local displacement of the tool around its periphery, the expected force-time diagrams with one and two degrees of shear angle would be estimated to follow the estimated paths shown in Fig. 12. It may be seen that the measured variation in force pattern

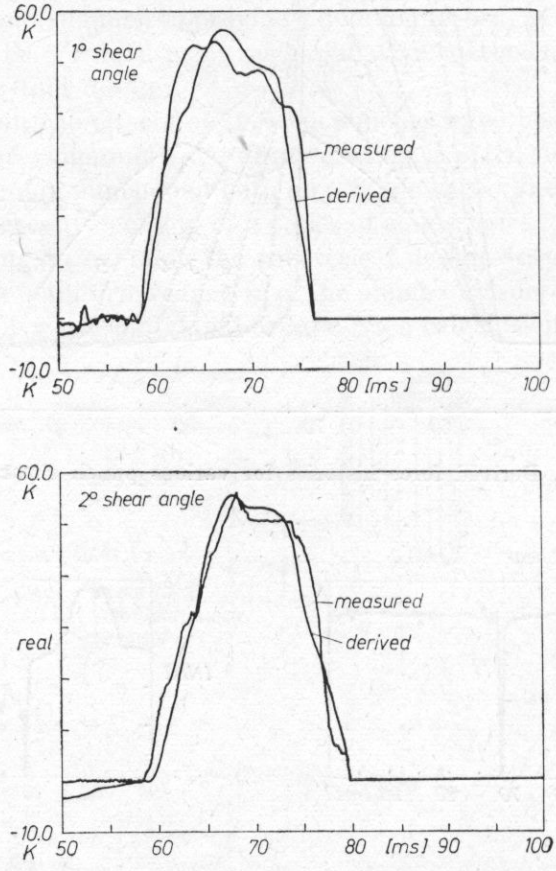

Fig. 12. Measured and derived force histories for piercing with a shear angled punch piercing 20 mm holes in 3 mm *HRS*

arising from the existence of 1° and 2° of shear is to modify the yielding pattern but not the force derivative at fracture. Clearly once fracture occurs it spreads around the periphery very fast. It follows that shear design must take into account the dynamics of the situation, and that the best method of design is via the use of experimental purchase: that such a development is well worthwhile on long run tool assemblies is indicated in Fig. 13, which shows the measured noise spectra (L_{eq}) for 0.1 and 2 degrees of shear. The reduction in the linearly weighted noise level is 12 dB: since so much of the noise energy is around frequencies 200–500 Hz, an additional 2 dB reduction would be recorded if A-weighting of the spectra had been carried out.

To conclude this section, it is probably true to say that of all the noise

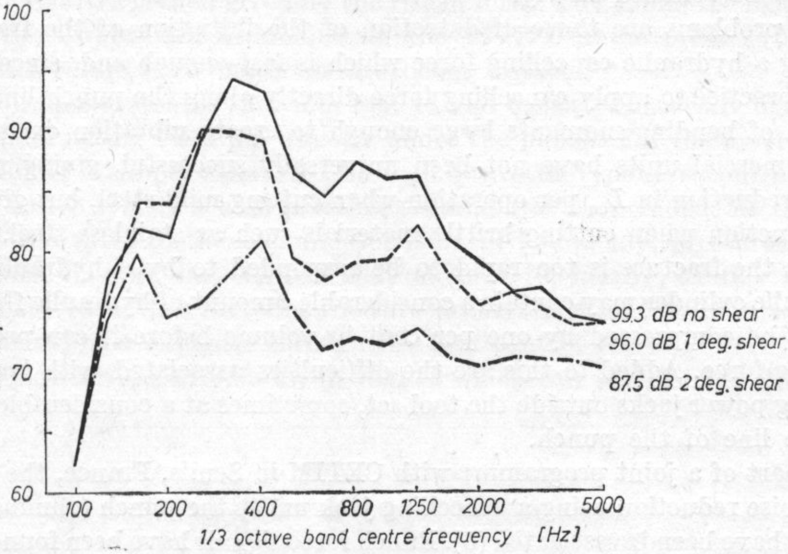

Fig. 13. *SPL* spectra — sheared punch

control procedures available to us, the scientific design of slightly sheared punches is surely the most successful approach to recommend to punch press designers, especially as this needs so little change on the large number of presses already in operation.

Active cancellation devices

An alternative approach to smoothing the rapid rates of change of force on the structure by tooling modifications is to incorporate some form of damping or cancellation system to oppose the structural springback following workpiece material fracture. A number of such systems have been built and tested by various researchers and workers and several commercial units sold. The effective performance of such systems, however, is restricted to a few limited applications where soft materials are being worked or on the less stiff types of press structure.

Two categories of force cancellation device exist, one which is best described as a passive or damper system, the other an "active" device which provides an electronically timed, though hydraulically powered, cancelling force to replace the tooling forces at the instant of material fracture. Both aim at reducing the sudden springback of the structure following fracture, thus smoothing the sudden release of strain energy around the whole machine. It is the sudden release which gives rise to the high frequency vibrations replacing it by a subsequent "leak" system, which cuts down these high frequencies and allows the strain energy to be released slowly.

The problems are those of detection of the initiation of the fracture, of obtaining a hydraulic cancelling force which is fast enough and, since it is difficult in practice to apply cancelling force directly along the punch line, the elimination of bending moments large enough to create vibration excitation.

Commercial units have not been universally successful, giving up to ten decibels reduction in L_{eq} per operation when cutting mild steel, but giving very little reduction when cutting brittle materials such as stainless steel. In these instances the fracture is too rapid to be responded to by a hydraulic device. A hydraulic cylinder may contain a considerable amount of hydraulic fluid which needs to be compressed by one per cent in volume before it can provide the concelling force. Added to this are the difficulties associated with housing the cancelling power jacks outside the tool set, sometimes at a considerable distance from the line of the punch.

As part of a joint programme with CETIM in Senlis, France, the possibilities of noise reduction using a cancelling jack under the punch (eliminating bed bending) have been investigated (5). Punch force signals have been found in practice to be surprisingly repetitive and similar to each other and consistent enough to use a simple time delayed pulse triggered at a fixed point in the press cycle to operate the cancellation device.

The work was aimed basically at investigating the performance and noise

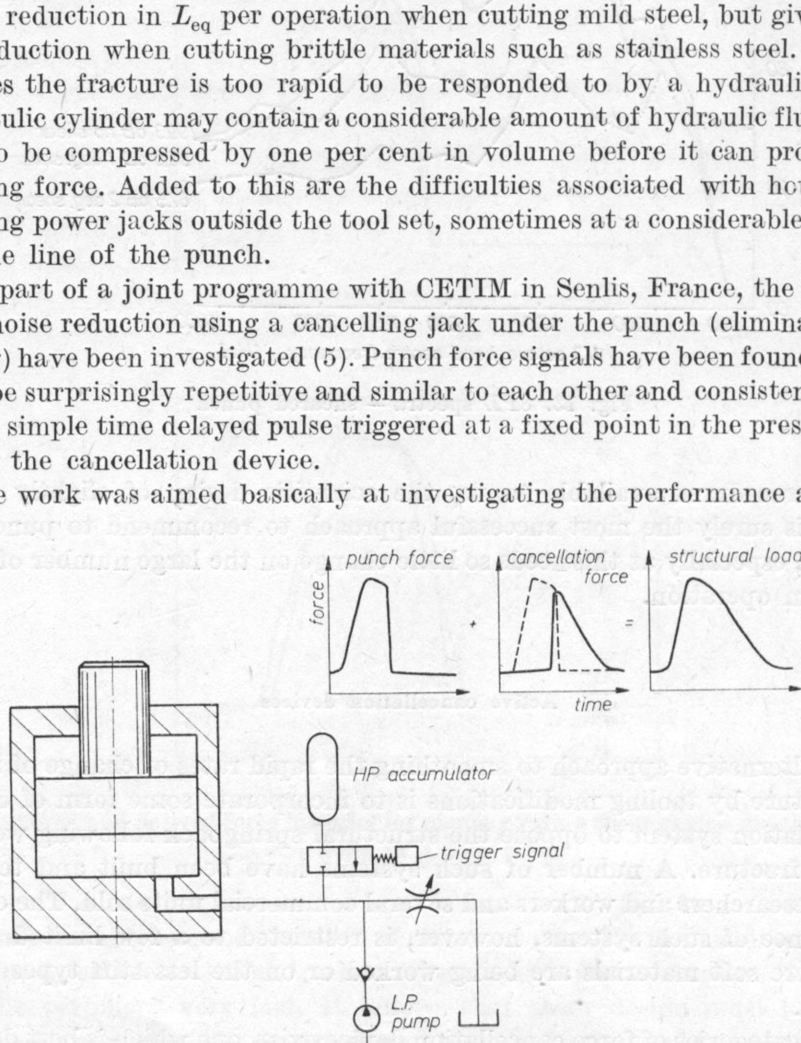

Fig. 14. Force cancellation system using *HP* accumulator

reductions possible with such systems. There is of course a limit to the noise reductions possible by smoothing the release of strain energy from the structure. This is related to the rate of rise of force. This in turn is determined by the punch velocity on contact with the workpiece. Smoothing the release of strain energy below that of the rate of rise will not give any further noise reduction as the peak force derivative is then given by the rise of force. This limits the maximum noise reductions which are attainable on the ISVR C-frame press to around 10 dB when medium hard materials are being worked.

The hydraulic system is shown in Fig. 14 and consisted basically of an hydraulic cylinder acting via a pin directly under the punch. For the passive system this cylinder is simply exhausted through a throttle-type flow control valve but for the active system a high pressure accumulator is included. At the correct instant the valve to the accumulator is opened suddenly, pressurising the cylinder which then applies the cancellation force to, ideally, exactly replace the force removed as the material fracture occurs. The load is then released slowly as the cylinder exhausts through the flow control valve.

Two typical results are shown in Fig. 15 which can be taken to illustrate

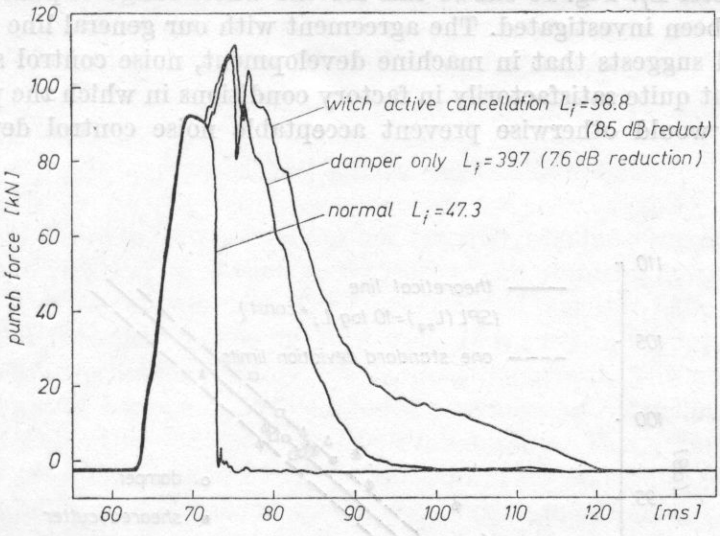

Fig. 15. Effect of active cancellation – 3 mm *BDS*

the end of this particular investigation. In this work, noise was not measured as such, since the measurement of

$$10 \log \frac{\sum [f(t)_{\max}]^2}{f_{\text{ref}}^2}$$

was found to be simpler and a reliable measure of machine noise output. It may be seen that ultimately a noise reduction of 8.5 dB was estimated to occur with active cancellation, 7.6 dB reduction with the passive damper. On the other hand, at the beignning, the introduction of the damper gave only 1.7 dB reduction from that from a normal system. The decrease can therefore be traced more to the timing of the punch "freezing" process rather than to the details of the pulse cancellation.

This is generally true in all our work, and provides a clear warning in the use of such methods. Fig. 15 shows clearly the sharp succession of high force gradients which occur if timing is not right. If this is not kept in mind, the total effect of the cancellation device will be to add additional vibrational energy, risking distortion and possibily damage to the press, and a possible increase in the total noise radiated.

Noise measurements against values of L_f

Throughout the latter part of the work described in this paper, spot checks have been made of the measured noise levels L_{eq} (0.1 s) against our punch design noise parameter L_f. Fig. 16 shows this for the whole range of punch variables which have been investigated. The agreement with our general line is very satisfying, and suggests that in machine development, noise control studies can be carried out quite satisfactorily in factory conditions in which the presence of other noises would otherwise prevent acceptable noise control development.

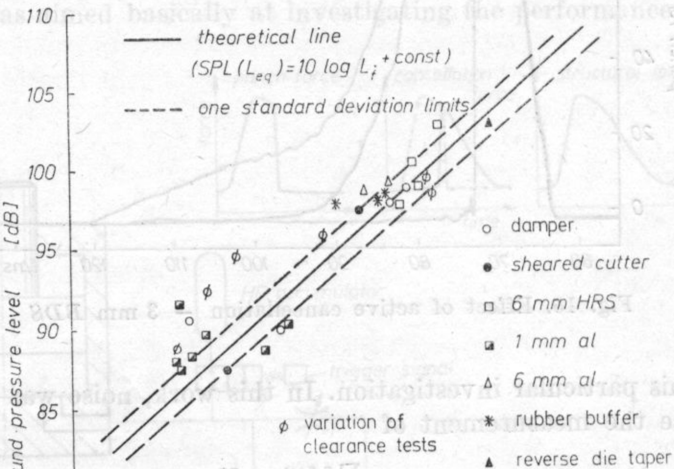

Fig. 16. Sound pressure level versus punch force derivative parameter for various piercing conditions

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