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GAS FILM DAMPING OF MINING CONVEYORS

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INTRODUCTION

Most mining equipment has to operate within a hostile environment under arduous conditions and consequently is designed for heavy duties. In addition there are stringent safety requirements which, for example, prohibit the use of a large number of conventional, non-metallic noise control products underground.

This paper presents a practical demonstration of damping on a particular piece of heavy mining equipment, using an all-metallic welded construction assembly, in order to reduce radiated noise levels. Whereas other researchers have applied gas film damping to reduce the vibration of relatively thin panels, the current work involved damping a thick steel plate by the incorporation of a thin plate. In addition these plates had much larger spans between fixing points than is commonly encountered with this type of damping technique.

NOISE GENERATION MECHANISM

Armoured face conveyors (afcs) are used extensively within the coal mining industry, principally as the initial means of transporting coal from the coal face immediately after it has been cut, but also as integral parts of continuous-mining machines and roadheaders. At the coalface, the conveyors have to withstand the weight of impact of falling coal as it is cut and also the wear associated with transporting the coal. This transportation is effected by dragging the coal along the afc using flights which are pulled by continuous lengths of chain, Figure 1. It is this method of operation which gives rise to noise radiation from the afc when it is running empty. Noise is generated by impact of the flights and chains on the individual sections of the afc (known as line pans) and particularly by impacts at the pan joints. When loaded with coal, which is often moist due to the use of water sprays to minimise dust levels during cutting, the bulk of the coal prevents these impacts and the fine coal particles act as a lubricant. Consequently under these conditions very little noise is generated.

The main noise radiating surface of the afc is the deckplate upon which the flights and chains rest. Due to the welded method of construction, the all steel assembly is relatively lightly damped and as a result the line pan and especially the deck plate which is typically 20mm thick mild steel, has many resonant modes affecting the overall radiated noise. It will be appreciated, that conventional damping techniques using viscoelastic materials cannot be employed to reduce the amplitude of these resonant modes, because of the deckplate thickness, and therefore a line pan construction which makes use of gas film damping has been devised.

RELATED WORK

Initial studies of structural damping due to thin gas films was concerned with joint damping, investigating the dissipation of vibrational energy of plates which had beams multi-point-fastened to them, for example by rivets, bolts or spot welds. [1-3]

More recently Trochidis[4] extended this to cover the damping of thin plates connected together by multi-point fixing and presented a number of experimental investigations. These showed the effects of layer thickness, the ratio of plate thicknesses, the number and spacing of fixing points, and gas viscosity. Fox and Whitton[5] also considered the damping of a thin plate using a thin gas layer and investigated the effects of air pressure. However, they deliberately restricted themselves to study of the first two normal modes. Moser[6] extended the Trochidis theoretical model to take account of the compressibility of the gas in the layer, but did not produce any additional test data.

ARMOURED FACE CONVEYOR STUDY

The standard (reference) line pan was of the type shown schematically in Figure 1, 1.5m long, 600mm wide and with a 20mm thick deck plate. The damped pan was identical in all respects except for an additional 5mm thick plate attached to the underside of the deck plate. This was connected to the side plates and the deck plate via continuous welds along the two edges and was also connected to the deck plate at each end by welding around small circular cut-outs in the 5mm plate. Although the two plates were rigidly clamped together during welding, the maximum flatness tolerance of each plate was 1-2mm. Consequently the entrained airgap probably varied from 0 to 1.5 mm over the majority of the surface and could possibly have been up to 3 or 4mm in places. Due to this method of construction, two possible damping mechanisms could operate during vibration, one being damping due to the thin gas film and the other being Coulomb or friction damping at the points where the two plates came into contact. The 5mm plate was located on the underside of the standard deck plate such that mechanical strength and wear properties of the line pan were unimpaired. Fixing by welding was used principally for mechanical integrity but also because of the difficulties of adopting other fixing techniques which would not affect operation of the conveyor.

The standard pan and the damped line pan were compared by measuring the vibrational responses of the deck plates at two corresponding positions, near the centre of the plate and one quarter of the way along the plate at mid width. A transient testing technique was adopted, using impact hammer excitation, this being considerably quicker and simpler than a sinusoidal sweep. The hammer was fitted with a hard steel tip and had a load cell mounted close to the striking face, resulting in a flat force spectrum up to 5 KHz upon impact. No particular difficulties of poor signal to noise ratio on the damped pan or of the transient response of the standard pan decaying too slowly were encountered. Seven impact locations were used, some of these being representative of impact positions during operational tests, and a minimum of five impacts were imparted to the structure at each location. Care was taken to minimise significant variations between successive hammer blows in the input force power spectrum in order to avoid introducing non-linearities in the structural response.

Two separate analyses of the data were undertaken, the first being of the inertance (acceleration/force) frequency response function and the second being of the mobility (velocity/force) frequency response function. A multi-degree-of-freedom curve fitting technique, using a Gen Rad 2505 Signal Analysis system and SDRC Modal-Plus software package, was used with the inertance data in order to identify the structural modes. For this work the input forcing function was normalised for each pan so that frequency response functions could be directly compared. The curve fit confirmed that corresponding equivalent modes were generated in each pan and that the modal density was quite high. Values of the mode amplitudes, phases, frequencies and equivalent viscous damping ratios were obtained over the frequency range 100 Hz - 2 KHz, the values of damping (as loss factors) being given in Table 1. Examples of the frequency response functions, identifying the principal modes, are shown in Figures 2 to 4. For the damped pan there was a significant reduction in amplitude of all resonant modes above about 300 Hz. There was also a downward shift in the frequency of the resonant modes, when compared to the standard pan, and this shift became more pronounced with increasing frequency. The frequency shift was greater than can be accounted for purely from the increase in damping and is most probably a result of the additional 'mass' of the 5 mm thick plate. A further interesting feature was that although the derived damping coefficients for the standard pan steadily reduced in value with increasing frequency, those of the damped pan, up to 2 KHz, remained relatively unchanged. Because of the level of damping achieved at these frequencies, a number of the resonant modes above 1500 Hz were difficult to identify clearly.

The overall increase in level of damping on the line pan is most encouraging and perhaps slightly better than would have been expected from the work of Trochidis.[4] The increase in damping is lower than he achieved but he was using only up to 2mm thick plates. The ratio of plate thicknesses, 4:1 compares with his optimum results, but the average air film thickness is probably higher than his optimum of approximately 0.5mm. The other difference between the test samples was that with the afc line pan there was a higher probability that the plates were in contact between the fixing points and therefore a greather likelihood of Coulomb damping occurring.

The mobility frequency response function data was used to estimate the order of reduction in emitted noise that might be achieved when using the damped pans on an actual installation. Naturally this assumed that operating levels were governed by resonant response of the line pan deck plate, and that each pan type had comparable radiation efficiencies. Predicted reductions are given in Table 2. These were based on space and time averaged values of the surface velocity squared, the frequency content of which is given in Figure 5.

OPERATING TESTS

Fifteen damped pans were fitted into a standard 250 metre long simulation of a coal face afc at the MRDE test site. The conveyor was operated at its normal speed of 1.1 m/s without coal, although a little fine coal and water was present on the pans, typical of a coalface installation. Noise measurements were taken at 0.5 m above the centre of one of the damped and one of the standard pans. The resulting narrowband spectra are shown in Figure 6, with the individual octave band results included in Table 2. It is clear that although there is a 5 dB(A) reduction in the total emitted noise from the damped pan, the individual octave band reductions are much less than those predicted from the mobility data, especially in the 1 KHz region. This could be for a number of reasons:- (i) Levels are controlled by forced rather than resonant response, (ii) Adjacent pans in the line are providing additional damping, which particularly affects the response of the undamped pans. (iii) Noise radiation from other parts of the pan may be important. (iv) There may be significant radiation of noise from the flights and chain. Of these it is believed that it is the additional damping and forced response of the structure which limits the noise reductions. Certainly a number of the small peaks in the standard line pan spectrum have been smoothed, confirming that the resonant response has been reduced, above 250 Hz. However, the data in Table 2 show that the overall dB(A) level from the damped pan is governed by the remaining noise in the 250 Hz and 500 Hz octave bands, and therefore the levels in this frequency region require special consideration.

CONCLUSIONS

It has been shown that significant increases in the structural damping of a piece of heavy mining machinery can be achieved by using a double plate damping technique with a small or zero airgap between the plates. In addition it has been shown that this technique can be used over relatively large spans and that the vibration of thick steel plate members can be reduced by the addition of much thinner plates.

For this type of application it is perhaps most desirable for ease of manufacture and cost to have the plates in contact and welded along two sides. Other fixing systems and different or more controlled spacing between the steel plates may increase damping levels, but such improvements would need to affect the mid-frequency octave bands of 250 and 500 Hz in order to further reduce the dB(A) levels generated by armoured face conveyors during normal operation.

Also, in practice, subsequent reductions in radiated noise, may be controlled by forced rather than resonant response of the line pan.

ACKNOWLEDGEMENTS

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Undamped Line Pan		Damped Line Pan	
Loss Factor	Frequency (Hz)	Frequency (Hz)	Loss Factor
0.0248	156	161	0.0234
0.0150	197	191	0.0152
0.0010	221	203	0.0052
0.0056	362	302	0.0512
0.0176	486	419	0.0570
0.0052	512	466	0.0200
0.0060	531	519	0.0102
0.0018	630	610	0.0400
0.0002	720	680	0.0240
0.0020	839	808	0.0308
0.0012	967	897	0.0176
0.0062	1148	1059	0.0308
0.0076	1196	1102	0.0238
0.0040	1260	1165	0.0056
0.0090	1273		
0.0026	1354	1236	0.0454
0.0086	1468	1364	0.0734
0.0004	1517	1408	0.0152
0.0022	1595		
0.0028	1829	1711	0.0206
0.0006	1969	1781	0.0314

Table 1: Resonant afc deckplate modes up to 2 KHz

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Octave Band Centre Frequency (Hz)	Measured Sound Pressure Level Standard Pan dB (Lin)	Predicted dB Reduction from mobility data on damped Pan (dB(Lin))	Measured Sound Pressure Level dB (Lin) Damped Pan
63	83	2	82
125	92	4	90
250	93	2	91
500	92	5	88
1K	85	18	80
2K	80	10	76
4K	68	13	67

Table 2: Measured and Predicted Noise Levels at 0.5 metre above afc line pan deckplate during operating tests (No coal).

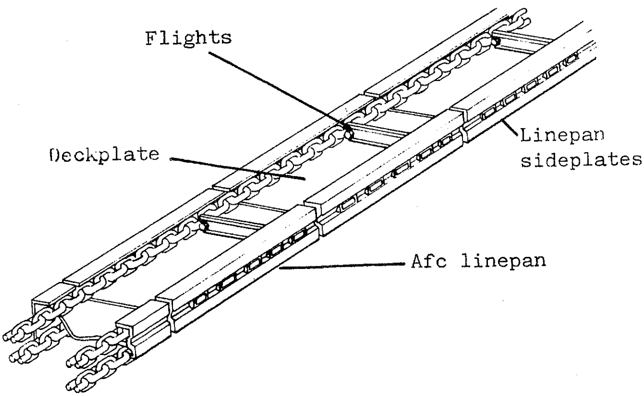

FIGURE 1. SCHEMATIC OF ARMOURED FACE CONVEYOR

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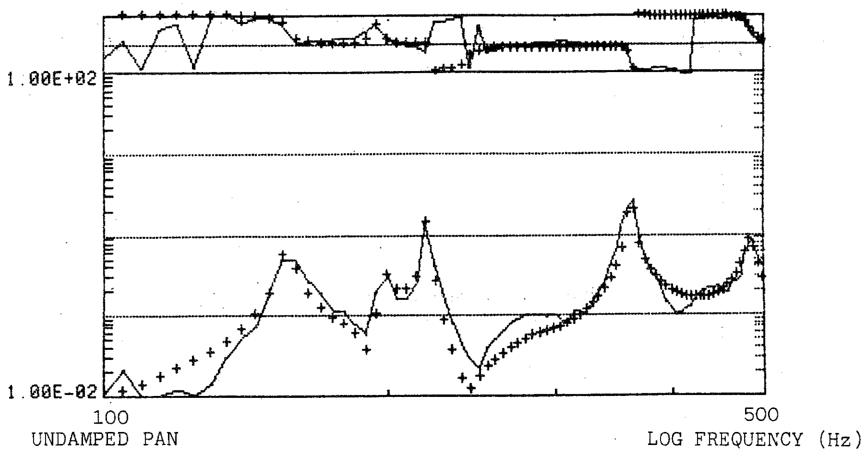

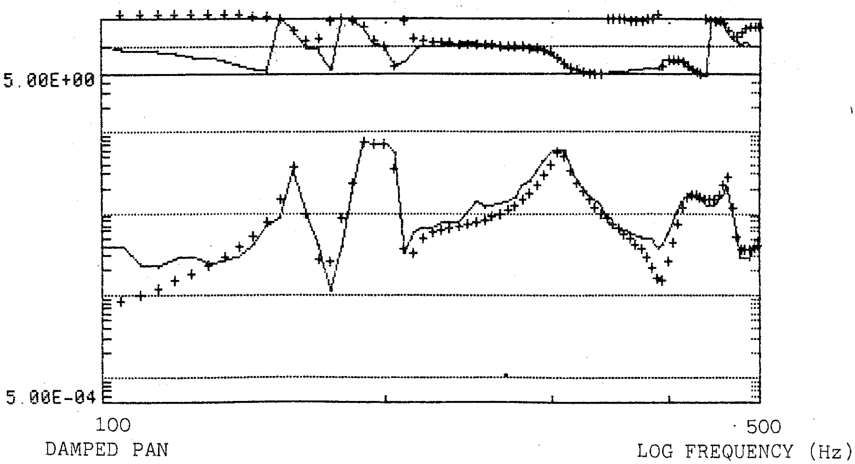

FIGURE 2. INERTANCE FREQUENCY RESPONSE FUNCTIONS. (100 Hz - 500 Hz)

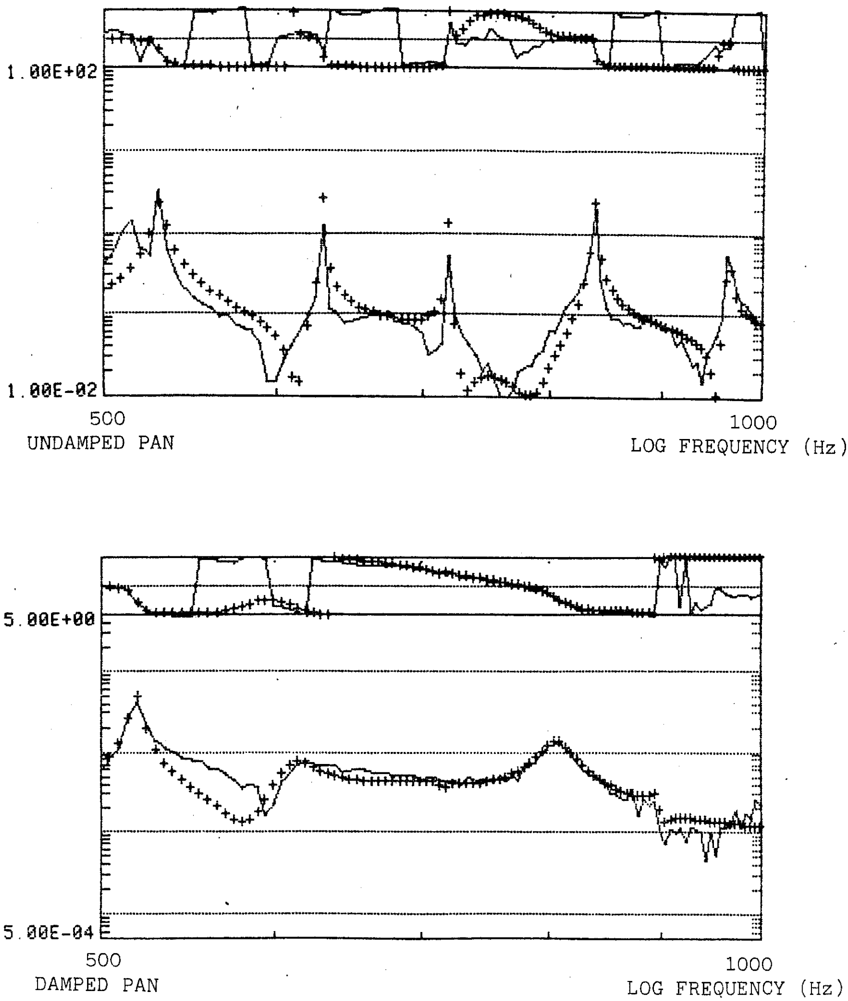

FIGURE 3. INERTANCE FREQUENCY RESPONSE FUNCTIONS. (500 Hz - 1 KHz)

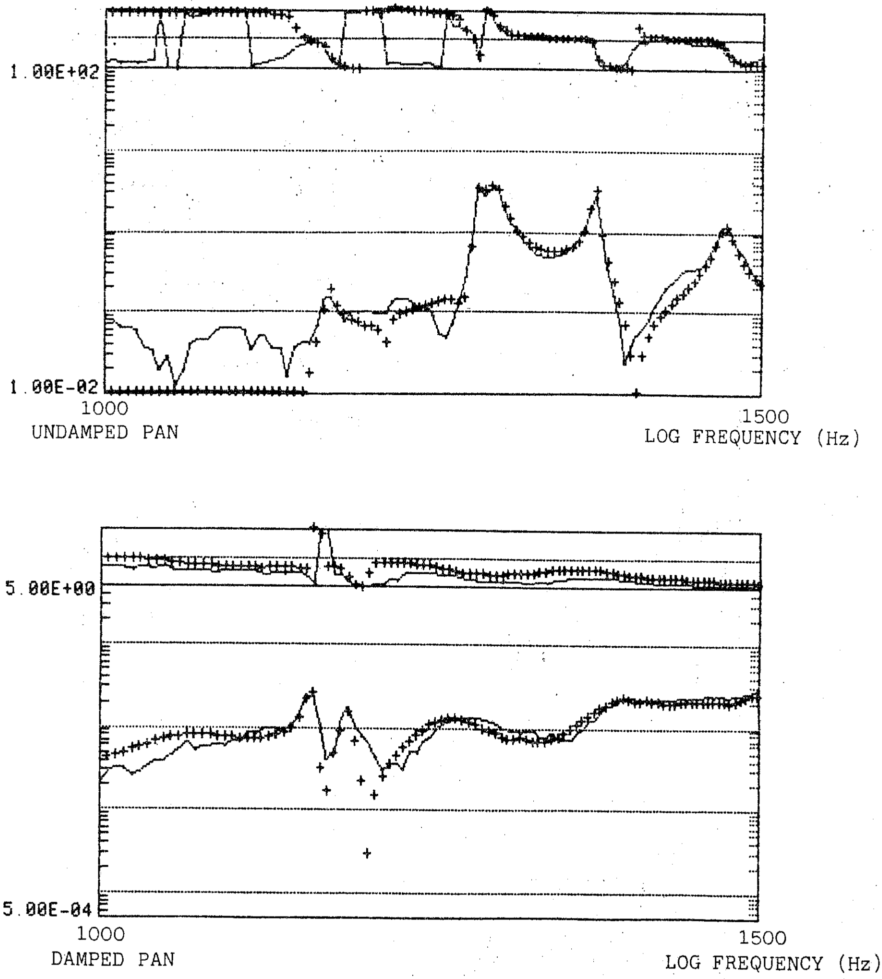

FIGURE 4. INERTANCE FREQUENCY RESPONSE FUNCTIONS. (1 KHz - 1.5 KHz)

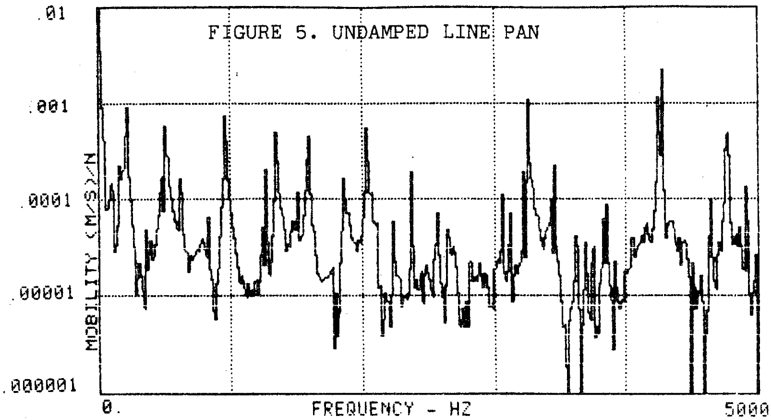

FIGURE 5. UNDAMPED LINE PAN

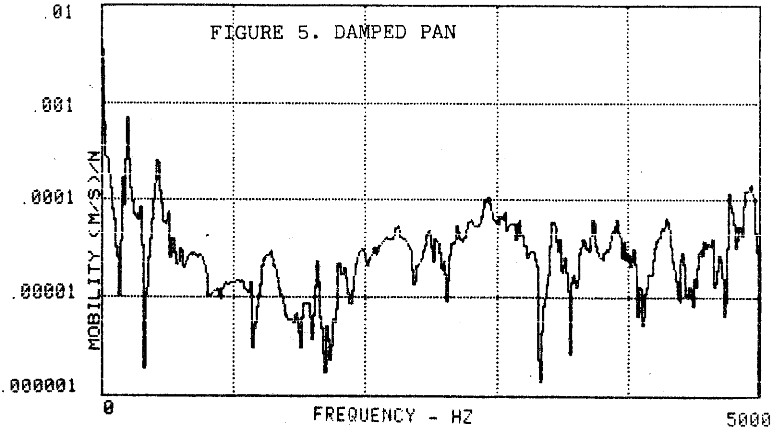

FIGURE 5. DAMPED PAN

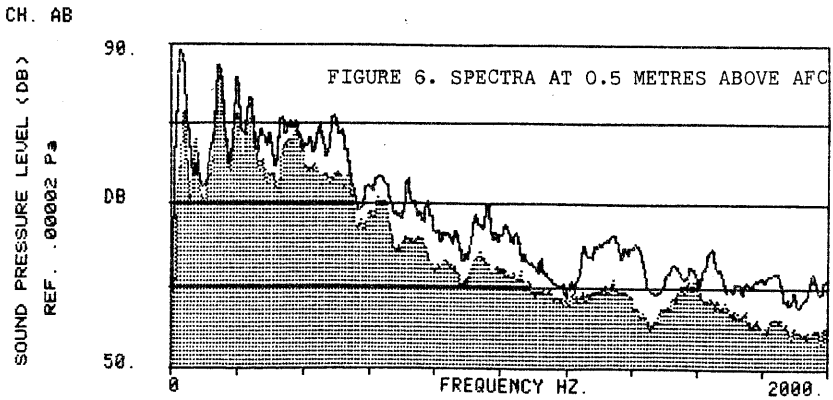

FIGURE 6. SPECTRA AT 0.5 METRES ABOVE AFC