AN ISOLATOR FOR USE IN AN ACTIVE VIBRATION CONTROL SYSTEM AND MEASUREMENT OF ITS EFFECTIVENESS

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1. INTRODUCTION

There are many practical situations in which the isolation of vibration transmitted through multiple degrees of freedom is an issue. Potential combinations of active and passive isolation techniques for attenuation of the vibration transmission from a source to a receiver have been discussed in reference [1]. Details of active/passive mount designs for single and multiple degrees of freedom attenuation have been given in the patent applications presented by Freudenburg [2] and Ross *et al* [3], respectively. These designs are fundamentally different in that those of Freudenburg use a single secondary actuator in an attempt to cancel vertical vibrations only, whereas the two stage mount design of Ross *et al* attempts to attenuate vibrations transmitted through the three translational degrees of freedom, by using a plurality of inertial shakers acting on an intermediate structure. The work presented in this paper describes the design concept (see Figure 1) and experimental appraisal of a two stage active/passive mount which combines a secondary actuator and a pneumatic mount in an attempt to isolate both translational and shear vibrations of a machinery installation. This is achieved by using a number of two stage isolation mounts in parallel.

2. MOUNT DESIGN AND EXPERIMENTAL EVALUATION

Each mount consists of an active element in "parallel" (see references [1,4]) with a passive isolator, both of which are connected to the top of a pneumatic mount; this is illustrated in Figure 1. The assumption made in this design is that one air mount is highly compliant in the two shear directions and therefore the horizontal motion between the source structure and the "intermediate" plate (Figure 1) will be minimal, provided that the passive isolation stage is chosen to have a relatively high shear stiffness. This averts the problem of excessive shear stress being applied to the rod connecting the output of the actuator to the intermediate plate. Therefore the actuator is able to act in parallel with the upper passive isolator in order to minimise the vertical component of the excitation, whilst the air mounts ensure minimal transmission of shear components of the primary excitation.

In order to investigate the performance of this mounting system, two different experimental rigs were devised: the first one, being laboratory based, is shown in section in Figure 1. This consists of a 0.99 m × 0.66 m × 0.002 m clamped steel receiver plate with the mounting system symmetrically orientated around the plate centre. The mounting system consists of a Ling 409 "primary" vibration unit, trunnion mounted, thus enabling primary excitation in any plane from the horizontal to the vertical. This primary unit is isolated from the receiver by four of the active/passive mounts described previously. The secondary actuators are Ling 201 dynamic units and the physical size of the system is indicated in Figure 1.

The second experiment attempted to isolate a 300 kg Enfield H02 (Horizontally Opposed 2-cylinder) diesel installation from its receiving structure, which consisted of a ribbed steel plate supported on a plastic foam layer [5]. A schematic of this experimental arrangement is shown in Figure 2. In each case the inputs to the secondary actuators are controlled by the ISVR programmable real-time signal processing unit [6], which implements the multi-channel version of the filtered-x LMS algorithm described by Elliott and Nelson [7]. The secondary actuators therefore minimise the sum of the squares of the signals received from a number of error sensors (accelerometers) in an attempt to cause global reductions in the vibrational energy within the receiver.

In each experiment conducted, the effect of the active mounting system is quantified by measuring accelerations before and after the application of active control. In the laboratory case the mounting system is assessed by comparing the "cost function" denoted

$$J_{wL} = \sum_{l=1}^{l=2} |a_L|^2$$

and the "energy function" denoted

$$J_{EN} = \sum_{n=1}^{n=N} |a_N|^2$$

before and after the application of active control, in which L is the total number of error sensors used and N is the number of distributed points over the receiver used to quantify the global performance of the system. The effect of the active control on the diesel installation is quantified by comparing the peak levels of the PSD's at multiples of the firing frequency both before and after control. Measurements are taken at the mounting points and at points distributed over the surface of the receiver.

3. RESULTS FROM THE LABORATORY OF THE MOUNTING SYSTEM

The results from the implementations of active control to the experimental rig shown in Figure 1 are presented in this section. Of particular interest here is the effect of the active control when the primary excitation acts in the horizontal plane and the number and spatial distribution of the input sensors to the controller. Three different sets of locations for the error sensors are considered for each of the vertical and horizontal primary inputs. These are as follows: (1) four sensors on the intermediate plates; four sensors at the bases of the mounts; and (iii) eight sensors distributed over the receiver.

Results are presented in terms of the reductions in the cost and energy functions (defined earlier) produced by the active control. Tables 1 and 2 show these measured reductions for the systems with both vertical and horizontal primary excitations and with $L = 4$ and $L = 8$ sensor inputs to the controller. These reductions are presented in dB terms where, for example, reduction in

$$J_{w4}(\mathrm{dB}) = 10 \log_{10}(J_{w4}(\mathrm{passive})/J_{w4}(\mathrm{active/passive})).$$

The laboratory experiments employ $N = 12$ equally spaced points over the receiver in order to form the energy function denoted J_{E12}. Table 3 presents the measured reductions for the system with vertical and horizontal primary excitations with the cost function, in this case derived from the accelerations of the four intermediate plates, also denoted J_{w4}. These reductions are plotted in Figures 3-6 along with the measured transfer function between the acceleration of the primary mass and the acceleration of a point distant from any nodal line on the receiver. This transfer function enables the performance of the control system to be related to the modal response of the receiver.

4. RESULTS FROM THE APPLICATION OF THE MOUNTING SYSTEM TO THE DIESEL INSTALLATION

Figure 7 shows the Power Spectral Density, PSD, of the acceleration signal measured from the intermediate plate of one of the four mounts. This shows the frequencies at which the acceleration, and also power are transmitted. The controller aims to minimise the given cost function as in the previous work, with cost function being defined as the sum of the squares of the accelerations from the four intermediate plates on the mounts, *i.e.*, J_{w4}. Minimisation of error sensors distributed over the receiver was also attempted. However in this case the secondary actuators were not able to produce sufficient vibrational amplitudes at the error sensors for the controller to be able to accurately carry out the system identification stage of the control scheme. The effect this minimisation has on the power is indicated by measuring PSD levels at strategic points before and after the application of the active control. These levels are measured using the measuring accelerometer and B&K analyser of figure 2. The measurement points are as follows. In order to assess the effect of the control on the cost function the acceleration PSD's from the error sensors on the intermediate plates are measured before and after active isolation. Also the PSD values below the mounts are monitored along with the eight points distributed around the plate (shown in figure 2) in an attempt to measure the spatial effect of the active isolation on the resulting vibration of the receiver.

Table 4 shows the effect of applying the control on the PSD's measured at these points. Four harmonics are investigated by tuning the active control to the particular frequencies. We see that small global reductions in the PSD levels are obtained at the 1st harmonic, negligible reductions are measured at the frequency of the 2nd harmonic, a slight overall increase (average 1.6dB) produced in the level at the 4th harmonic with good reductions measured at the 6th harmonic. It is also noted that in each case considered the controller has reduced the average level of the PSD signals from the four sensors used to form the cost function. This is expected from the previous experimental work.

5. DISCUSSION OF EXPERIMENTAL RESULTS

The results presented in Section 3 of this paper show that the mounting system is capable of isolating primary vibrations generated in any plane from the vertical to the horizontal. This is achieved by use of a single vertical secondary actuator at each mount and has obvious advantages over systems which employ multiple actuators in order to achieve isolation. The method of evaluating the performance of the system gives information of the reduction of the sensor inputs to the control algorithm and of the reduction in vibrational energy within the receiver. This is important in evaluating the global performance of the system. Figures are presented which show reductions in both cost and energy functions for the different control strategies and different planes of primary excitation (Figures 3-6).

An attempt to actively isolate a 300 kg 2-cylinder horizontally opposed diesel installation is also presented, the results of which are given in Table 4. The explanation of these results is as follows. Table 5 presents the peak level of the PSD from the error sensor on mount C, at the harmonics which are to be actively cancelled. The peak levels due to the operation of the machine alone are quantified by observing Figure 7. The levels due to the secondary force are assessed by driving the secondary shaker at the relevant harmonic frequency and measuring the peak in the resultant PSD by use of the B&K analyser. We see that the only harmonic where the secondary PSD level exceeds that of the primary is the sixth, also the harmonic where good reductions are observed. Comparable levels of PSD occur at the 1st harmonic where small global reductions are measured. However the secondary levels are not sufficient to produce significant reductions at the 2nd (firing frequency) or 4th harmonics. Therefore the experimental results suggest that if the necessary secondary force requirements are available then this mounting arrangement has potential for isolating such real machines . In the case considered the harmonic which transmits approximately 95% of the power is the one which occurs at 33 Hz (the firing frequency). Therefore in order to minimise the power input to the receiver this is the harmonic to cancel. The secondary actuators used in this experimental work were not of sufficient rating to affect this harmonic. It therefore follows that two methods could be employed in order to attempt to minimise this harmonic. Firstly employment of much larger secondary actuators may be used to provide the necessary secondary forces. However this may have limited practical use, for example it was calculated that employing four Ling 409 shakers as the secondary actuators would barely meet the secondary force requirement for cancellation of the 33 Hz harmonic. The use of these actuators would however add 120 kg of mass to the installation, almost half the mass of the diesel pump itself. The second method would be to design a practical shear isolator for the upper passive isolation stage. This would maintain the shear stiffness while reducing the vertical stiffness, hence facilitating better attenuation of the primary disturbance and leading to lower secondary force requirements.

6. CONCLUSIONS

In summary, the main issue addressed in this paper has been an investigation of a practical active/passive mount design capable of reducing vibration transmission from a primary source, which is no longer constrained to act in the purely vertical plane. Several conclusions result from this work.

(i) The active/passive mount described is capable of reducing the vibration transmission from both shear and translational primary inputs using a single secondary actuator.

(ii) The high frequency limits of active control due are observed. Greater reductions in the receivers vibrational energy can be obtained by using 8 error sensors as opposed to 4 as frequency increases (> 60 Hz), with particularly useful reductions at resonant frequencies of the receiver.

(iii) The active/passive system is particularly advantageous in comparison to the passive only system at low frequencies.

(iv) The application of the mounting system to a diesel installation gives a preliminary demonstration of its potential use in 'real' situations. However, further experimental work is required utilising either (i) larger secondary actuators or (ii) a practical upper shear isolator, in order to provide an entirely conclusive appraisal of the mounting system's potential.

7. REFERENCES

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Vertical primary excitation			Horizontal primary excitation		
Frequency Hz	J_{w4} reduction (dB)	J_{E12} reduction (dB)	Frequency Hz	J_{w4} reduction (dB)	J_{E12} reduction (dB)
20	27.8	29.2	35	14	3.2
25	14	11	40	13.5	14.5
30	24	25.1	48	30	22.5
35	32	31.5	55	37.2	28.2
40	34.2	32.7	60	27	15.2
48	37.5	38.7	65	20	6.7
55	19.3	23	70	12	0.7
60	29.1	16.5	75	9.7	0.5
65	7	5	80	18.4	-0.8
70	15.5	-2	85	3	-0.9
75	18.1	1.1	90	31.7	12.5
80	18.1	1.6	94	15	8.3
85	16.2	3.1	-	-	-
90	28	13.5	-	-	-
94	37	16.6	-	-	-

Table.1 : Reductions in cost function J_{w4} and energy function J_{E12} for the vertical and horizontal primary excitations using four sensors at the bases of the mounts to form J_{w4}.

Vertical primary excitation			Horizontal primary excitation		
Frequency Hz	J_{w8} reduction (dB)	J_{E12} reduction (dB)	Frequency Hz	J_{w8} reduction (dB)	J_{E12} reduction (dB)
70	1.8	-1.3	70	4.2	-1.7
75	13	7	75	6.7	1
80	13	6.3	80	7.3	3
85	11.5	4	85	7.3	3
90	23	14.5	90	17.7	11.2
94	15.9	7.8	94	23.6	12

Table 2 : Reductions in cost function J_{w8} and energy function J_{E12} for the vertical and horizontal primary excitations using eight sensors as shown in figure 5.6 to form J_{w8}.

Vertical primary excitation			Horizontal primary excitation		
Frequency Hz	J_{w4} reduction (dB)	J_{E12} reduction (dB)	Frequency Hz	J_{w4} reduction (dB)	J_{E12} reduction (dB)
20	33	35.5	35	5	9
25	35.4	30	40	12.6	4.6
30	35.6	36	48	16	14.8
35	35.5	28.2	55	16.5	14
40	26	34.6	60	20	6.9
48	36	33	65	12	-2
55	23	28	70	8.8	1
60	34	14	75	13	-2
65	26	2.2	80	10	-1.4
70	17	2.2	85	14.5	2.4
75	17	-1.5	90	7	-3
80	10	-4.4	94	6.1	-1.5
85	10.8	-0.7	-	-	-
90	9	0.3	-	-	-
94	8.8	0.8	-	-	-

Table 3 : Reductions in cost function J_{w4} and energy function J_{E12} for the vertical and horizontal primary excitations using four sensors located on the intermediate plate (see figure 1) in order to form the cost function J_{w4}.

measuring position	1st harmonic (r.p.m)	2nd harmonic (firing freq)	4th harmonic (≈66Hz)	6th harmonic (≈99Hz)
above mount A	4.5	0.9	6.5	15.9
above mount B	3.3	0.8	4.1	18.9
above mount C	6.7	1.6	7.2	10.6
above mount D	6.1	1.4	14.0	-0.8
Average	*5.2*	*1.2*	*7.9*	*11.2*
below mount A	4.3	1.4	-4.8	19.1
below mount B	0.8	2.0	19.4	14.7
below mount C	5.8	0.5	4.3	11.8
below mount D	4.2	4.9	-8.0	12.8
position A	4.0	1.3	-5	19.9
position B	9.5	2.0	-12	12.7
position C	1.1	0.8	-2.5	8.6
position D	1.9	3.6	-2.1	10.8
position E	6	3	-2.3	10.6
position F	6.1	0.9	3.5	12
position G	5.6	0.5	0.5	10.0
position H	4.2	0.1	-13.4	7.0
Average	*4.8*	*1.75*	*-1.6*	*12.5*

Table 4: Reductions in the acceleration PSD levels (dB) at various positions on the structure for the system with four sensors located on the intermediate structures. The negative values indicate an increase in level.
(dB re arbitrary units)

Harmonic	peak in PSD due to primary	peak in PSD due to secondary
1st (16.5Hz)	-39 dB	-40 dB
2nd (33Hz)	-16 dB	-29 dB
4th (66Hz)	-10 dB	-14 dB
6th (99Hz)	-15 dB	-9 dB

Table 5 : Showing the peak PSD levels due to primary and secondary excitation measured on the intermediate structure of mount C.The secondary actuator being driven at maximum current, (dB re. arbitrary units)

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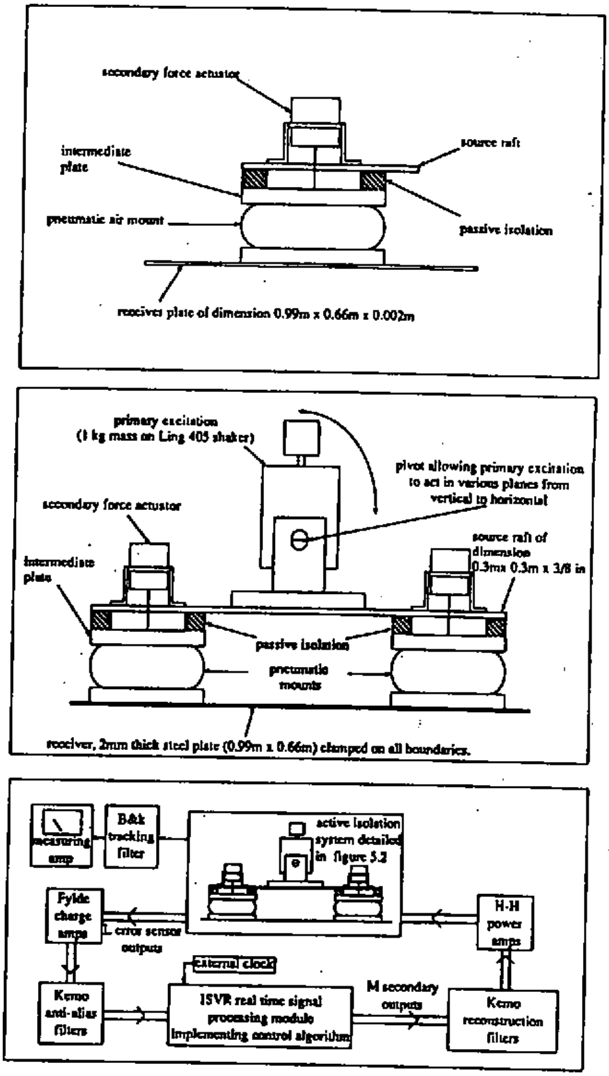
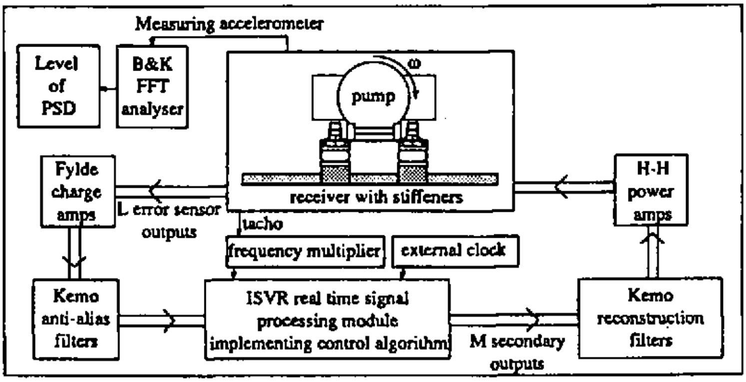


Figure 1. Details of the mount design and laboratory experimental rig



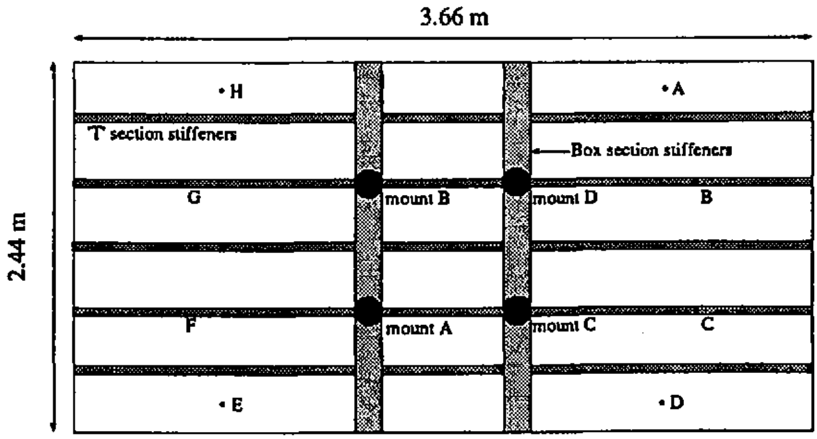


Figure 2. Section of the experimental system used to measure the reductions in PSD for the diesel installation, also shown is a plan view of the receiver and measurement locations

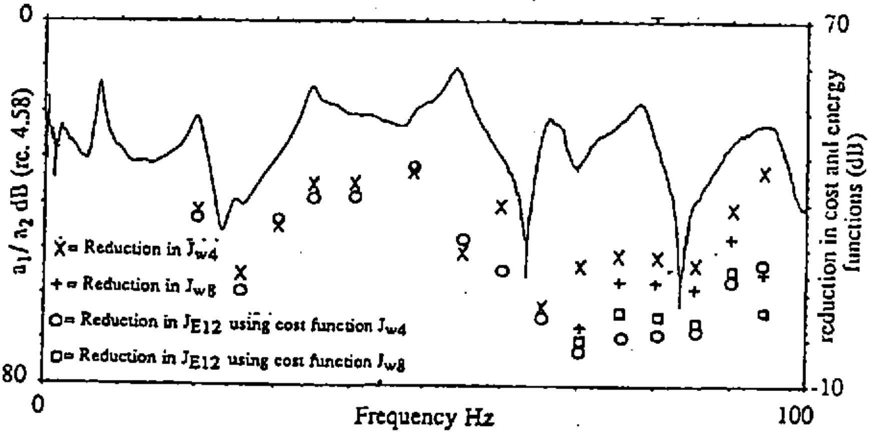


Figure 3. Reductions in cost and energy functions for the system with vertical primary excitation.

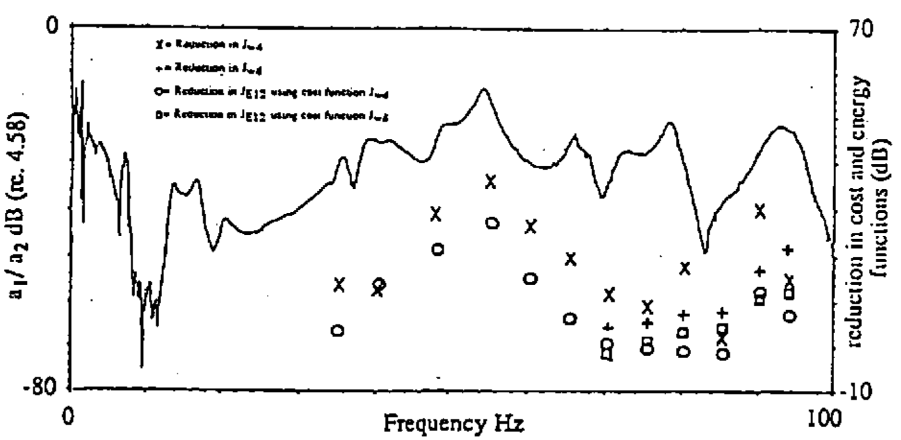


Figure 4. Reductions in cost and energy functions for the system with horizontal primary excitation.

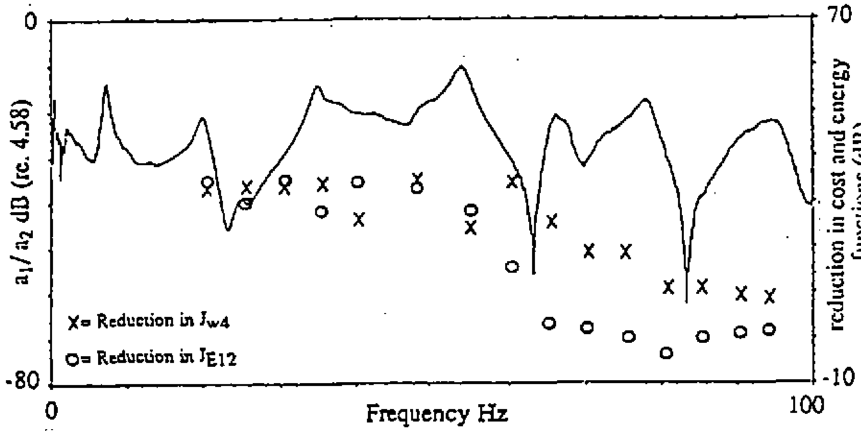


Figure 5. Reductions in cost and energy functions for the system with vertical primary excitation and the error sensors located on the intermediate plates.

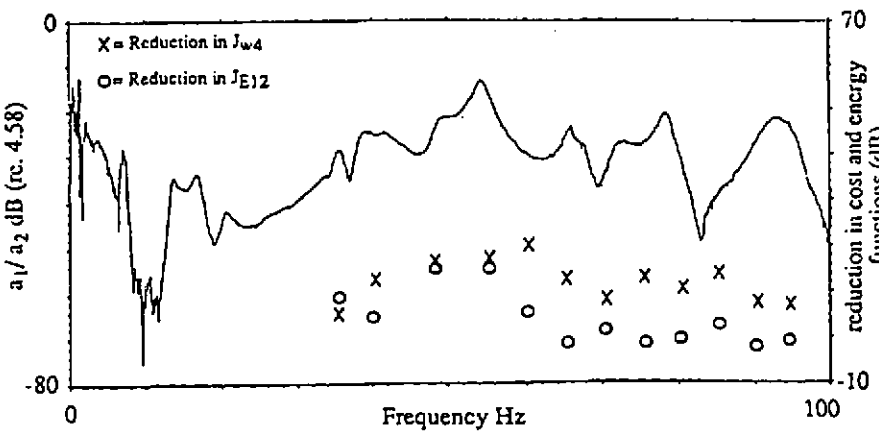


Figure 6. Reductions in cost and energy functions for the system with horizontal primary excitation and the error sensors located on the intermediate plates.

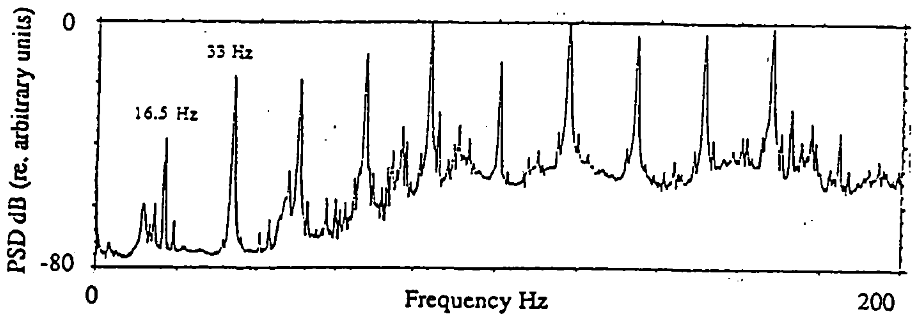


Figure 7 : PSD of the accelerometer signal from the intermediate structure of one of the four mounts (mount C in figure 2).

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PROBLEM DEFINITION AND INITIAL MEASUREMENTS FOR THE FREE FIELD ACTIVE NOISE CONTROL OF A SIMPLE MECHANICAL DEVICE

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1. INTRODUCTION

A promising approach for free field active noise control (ANC) in a three dimensional free field is to feed a measurement of the noise to be controlled to a nearby set of secondary sources downstream in the acoustic propagation path. The signal is passed through an adaptive algorithm which acts to minimise the acoustic pressure at a set of error sensing microphones some distance out into the field [1,2,3]. See figure 1.

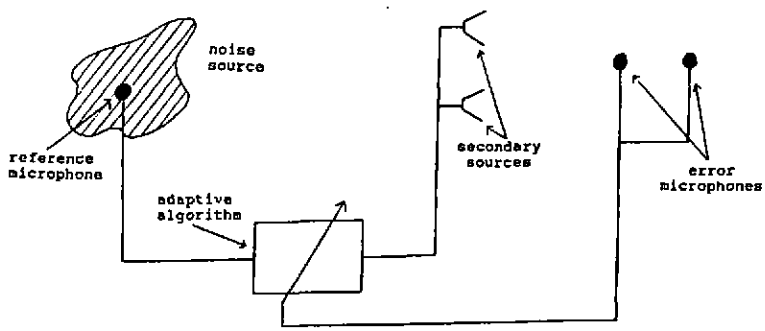

figure 1.

For the case where the noise source is mobile it is impractical to locate a set of microphones remote from the source. It might be possible to overcome this by using a set of microphones close to the noise source, and adapt the feedforward algorithm to a condition that will minimise the far field acoustic pressure. This will not, of course, minimise the acoustic pressure at the error sensing microphones.

The problem we address is the active quieting of a mobile vehicle for a limited azimuth angle. This is a complex problem so a step by step approach has been adopted starting with a much simpler problem. Initially the feasibility of achieving cancellation in the far field will be established, by using error sensing microphones in the far field. The microphones will then be moved as close as possible to the noise source, while still quieting the vehicle sounds over the region of interest.

2. PROBLEM DEFINITION

2.1 bounds of problem

Many potential difficulties are reduced by constraining the initial goals of the work. We will discuss several of these constraints.

Noise cancellation will be achieved over a limited azimuth angle, power reduction over the whole field is not a concern, thus, the secondary sources will be allowed to add to the noise in the region outside the desired quieting zone. This removes the restriction that the secondary sources must be within half a wavelength of the noise source [4], a requirement which is very difficult in the case of the distributed noise source we wish to consider.

The noise source will consist principally of engine noise, which tends to be a slowly time varying, periodic signal with distinct frequency components. The periodic nature reduces the constraints on causality, again allowing more freedom in the placement of the secondary sources, and allows the analysis to be carried out in the frequency domain. Each frequency component can be tracked and individually controlled, thus reducing the problem to one of controlling a set of single frequencies.

Noise cancellation is to be achieved on a scale of 100's of meters. Since the atmosphere attenuates high frequencies more than low frequencies, there is no need to attempt canceling at high frequencies. This focuses the work at frequencies below 500 Hz, where ANC works best [5]. Also the principal spectral lines from vehicles are generally below 500 Hz.

2.2 approach

The general approach is shown in figure 1, where a reference signal from the noise source is fed through an adaptive algorithm to the secondary source, that minimises the acoustic pressure at a microphone. The reference signal is measured using an accelerometer on the engine block, which eliminates any acoustic feedback from the secondary sources. The transfer function relating the acceleration to the acoustic pressure can be included in the feedforward path. As the source is distributed the secondary source will not be directly coupled to the noise source, therefore canceling will be achieved through superposition [1].

Several choices exist for a possible solution to the problem of establishing a suitable error signal when it is not possible to locate the error sensing microphone in the desired quieting region.

The simplest choice is to minimise the total sound level at points in the near field, this may cause a reduction in the acoustic pressure beyond these points (we desire a reduction all the way to the far field). This choice puts no constraint on the far field acoustic pressure and probably will not be very successful. In this paper we use near field to describe the region approximately 1 to 10 meters from the sound source and far field is used to describe distances beyond 400 meters. In all cases we use simple geometric spreading to describe the spatial variations of the sound field and thus do not include any near field source effects in our calculations.

The next possible strategy is to try to force the pressure at a set of microphones in the near field to a condition that will minimise the far field acoustic pressure. This requires a priori knowledge of the solution, which can be found by performing experiments that minimise the far field pressure under various conditions, while taking measurements of the near field pressure. This may require many experiments to determine the solutions for a wide variety of conditions. It may be possible to develop sound field modeling programs to examine several cases and reduce the experimental work.

A more computationally extensive solution is to use the signals from an array of microphones in the near field to extrapolate into the far field using the Helmholtz integral. This estimate of the far field is then used for the error signal. A priori knowledge of the field shape would be used to minimise the number of microphones required in the array.

3. INITIAL EXPERIMENTS

For the sake of simplicity the initial experiments used a small, single cylinder, 4 cycle, 800 watt, Honda generator as the noise source. A digital audio tape recorder with four channels captured and stored the data. Initial measurements used two microphone channels, an accelerometer mounted on the engine block, and a pressure transducer in the exhaust pipe. In later measurements three microphone channels and the accelerometer were used. The experiment was carried out in an asphalt parking lot on a quiet morning with little wind. The asphalt was selected to provide a known ground impedance and reduce the number of unknowns in the measurements.

Sound recordings were made at various distances from the generator at a height of about one meter above the ground. In a second experiment measurements were taken at 30 separate points on a vertical grid 2.7 meters from the generator.

An important result of these experiments was that the accelerometer signal was found to contain all the spectral components of the exhaust signal, and, of course, more. Also, the spectral components in the accelerometer signal were found to contain all the spectral components found in the microphones. This implies that only the accelerometer is necessary as a noise source reference signal.

4. MODELLING

To investigate the effect of various canceling strategies a model was constructed that calculates the complex pressure field in three dimensional space, including ground, due to a set of point sources. The operator can locate several error sensing microphones and noise canceling sources in the sound field. The positions of the secondary sources and the error sensing microphones are specified and the secondary source strengths are then calculated to minimise the sum of squared pressure at the microphone points [6]. The complex pressure is then calculated on a plane through the field for any desired orientation.

5. RESULTS

5.1 simulating cancellation

A pattern of point sources placed in a region 3 m by 2 m was used to simulate a distributed noise source and the secondary sources were also placed within this region. Several different arrangements of microphone points were investigated, at three different distances from the sources. While different microphone arrangements at a fixed distance produced different results, these differences were not very large, and the resultant field patterns were very similar. However, when the microphones were moved to a new location the resulting fields were quite different. Table 1 summarises the results.

FREE FIELD ANC:

Table 1

average source-microphone distance	typical reductions in far field	secondary source power required	comments
450 m	30 → 40 dB	20 times noise power	strong reduction in a cone in front of and beyond microphones
50 m	20 → 30 dB	14 times noise power	strong reduction on outside of cone, weaker reduction in center of cone
2 m	-10 → 20 dB	0.4 times noise power	very unpredictable and widely varying results

Figures 2 to 5 show contour plots of the resultant field intensity on a grid 1000 meters square. The separation between contours is 20 dB. The frequency is 85 Hz. Figure 2 shows the field when no canceling is attempted. The vehicle is located at the far left of the figure, the right side of the figure we describe as being in front of the vehicle.

Figure 3, the first in a sequence of figures where the microphones are sequentially moved from 450 meters closer to the source, shows a quiet zone directly in front of the vehicle. The hatchering shown in all these figures are keyed in figure 3. The most intense sound levels, 60 dB, are near the source and subsequent regions are separated by 20 dB. Figure 4, microphones at 50 meters, produces a similar quieting cone to that of figure 3, but the level of quieting achieved in the center is less.

The effect of minimising the very near field eliminates the single cone of quieting achieved when the microphones were further away. In figure 5. two cones of quieting are seen, while directly in front of the vehicle (in the far field region) sound levels increase. This indicates that it may be necessary to control the secondary source sound pattern in the direction of the desired cancellation region. It will be necessary to perform additional experiments with different secondary source spacing and various positions of the microphones to determine whether the 'dual cones of quieting' can be merged into one.

FREE FIELD ANC:

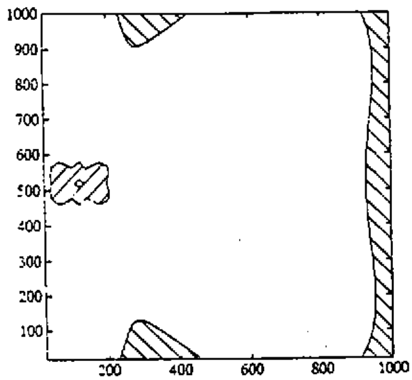

figure 2. no canceling

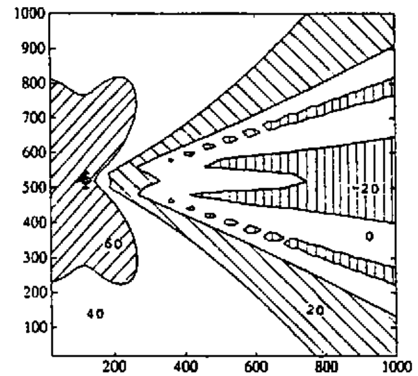

figure 3. canceling at 450m

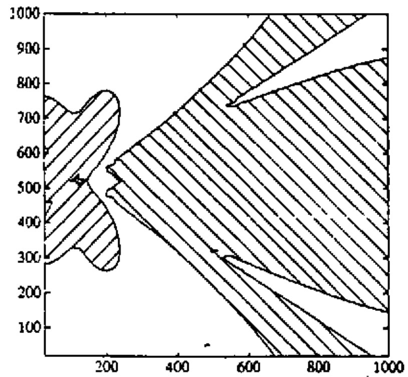

figure 4. canceling at 50m

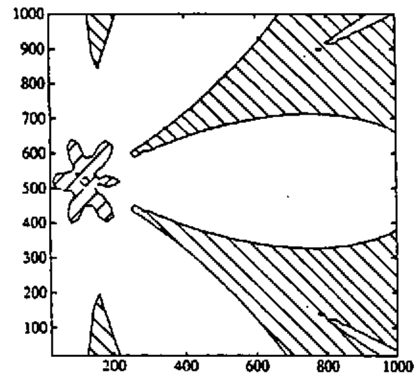

figure 5. canceling at 2m

5.2 comparison between model and experimental data

In this part of the work the generator was represented as a single point source at the height of the exhaust. The field was evaluated for several prominent spectral frequencies at the grid positions mentioned earlier. The ground was assumed to be a perfect reflector, as could be expected for asphalt at these frequencies. The measured generator sound field from the central column of the grid was compared with the sound field of the model point source.

Figure 6 shows the points measured, *, on a vertical line from about 0.3 m above the asphalt to a height of 2.8 m. This line was located a distance of 2.7 m from the generator exhaust. The solid line represents model data at the measurement frequency, 85 Hz; as can be seen from the figure the results are very close. Figure 7 is similar to figure 6 except the measured frequency was changed to 215 Hz. Again the results compare favorably, indicating the model is valid for its purpose. The noticeable reduction in sound at 1.2 m is a 'ground effect dip' caused by the asphalt surface.

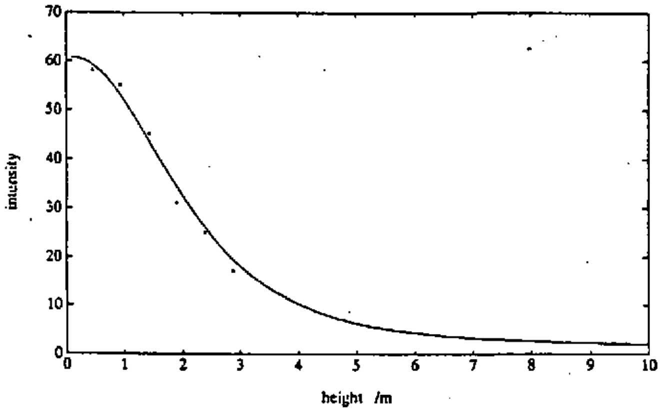

figure 6. central position of grid, frequency = 85 Hz

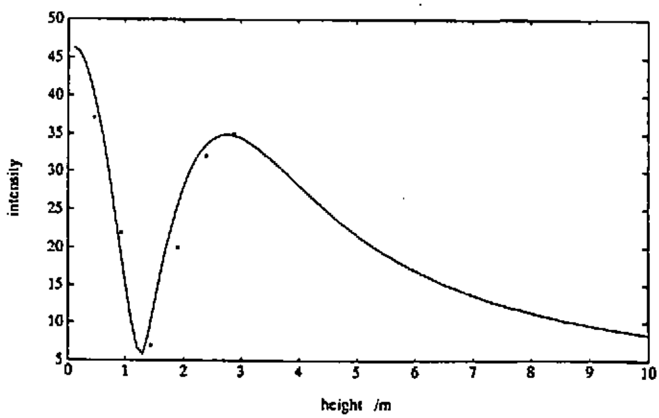

figure 7. central position of grid, frequency = 215 Hz

6. SUMMARY

An approach to the problem of ANC in the free field for a mobile, distributed noise source has been put forward. The goals of the work have been defined, and some major consequences of these discussed. In order to develop canceling strategies, with particular emphasis on where the acoustic pressure field should be minimised, a model was constructed using point sources. The model results have been compared to experimental measurements from a simple system, and were found to be close.

7. REFERENCES

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