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PASSIVE VIBRATION CONTROL VIA UNUSUAL GEOMETRIES: EXPERIMENTS ON MODEL AEROSPACE STRUCTURES

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In a previous paper [1], one of the present authors described a method whereby passive vibration isolation could be designed into a lightweight aerospace structure by the adoption of unusual geometric configurations. This was achieved by the use of Genetic Algorithm methods to optimize the isolation characteristics of the structure. The current brief paper follows on from that work and describes an experimental investigation into the performance of the structures designed in the earlier study. It is shown that, although the detailed behaviour of the experimental models is somewhat different from that predicted theoretically, the same broadband vibration isolation characteristics shown by the theory exist in practice. It is seen to be possible to achieve an average of 50 dB isolation in the vibrational energy transfer between the ends of a network of beams over a 100 Hz bandwidth in an essentially undamped structure. These experimental results are sufficiently encouraging that work is now underway to apply Genetic Algorithm based design methods to more complex structural vibration problems.

1. INTRODUCTION

A number of modern engineering structures are fabricated from beam elements, often arranged in some kind of regular lattice type structure; some of these are subjected to vibrations and it is then useful to be able to prevent vibrational energy from travelling along them. This problem reaches its most severe form in the booms of satellite structures which are normally constructed from very lightweight alloy materials and which are operated in environments where there is almost no damping (because of the lack of atmospheric damping and the desire to minimize payload weights). Moreover, such structures may well be required to hold sensitive instruments in very precise alignments, which make them very susceptible to unwanted structural vibrations. Recently, much interest has been focused on using active vibration control methods to overcome such difficulties. It would, of course, be better to build passive vibration isolation characteristics directly into such structures and this is the task investigated by using theoretical models by Keane [1]. In that work a regular two-dimensional lattice structure was modelled by using energy flow methods [2, 3] which were used to form the vibration control problem into an optimization task. Subsequent manipulation of this model by Genetic Algorithms [4] allowed a revised design to be produced that had significantly improved noise isolation in a specified frequency range. In Figure 1 is shown the original geometric configuration studied and in Figure 2 the final, optimized, design. In Figure 3 are shown the "before" and "after" vibration performances of the designs in the form of the energy level in the end-most vertical bar caused by unit amplitude forcing of an element near the base of the

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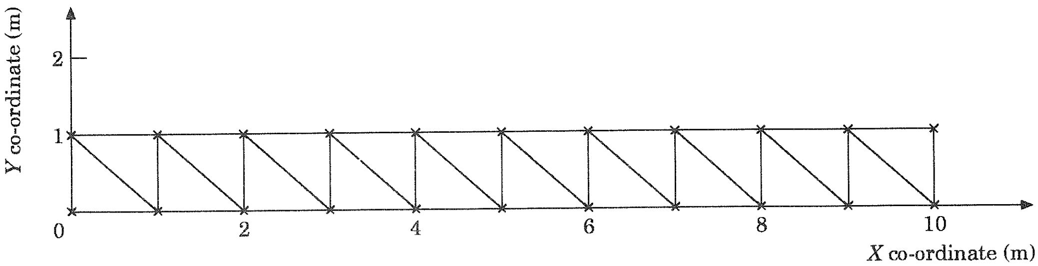

Figure 1. The initial structural design.

structure. Here, both structures are two-dimensional and the vibrations analyzed were those in the plane of the designs. It is clear from Figure 3 that a significant degree of noise isolation is predicted over the 100 Hz band, starting from 150 Hz.

This paper describes an experimental investigation into the performance of these designs, which was carried out to see whether or not the theoretical predictions could be realized in practice. To that end, two aluminium alloy structures were built and tested by using standard experimental techniques. The investigation showed that, although the detailed behaviour of the practical structures showed some variations from the theoretical designs, the predicted noise isolation was achievable in practice.

2. THE STRUCTURES

As has already been noted, the geometry of the original structure before optimization is illustrated in Figure 1. This consists of 40 Euler–Bernoulli beams all having the same properties per unit length. In the earlier study, EA was taken as 69·87 MN, EI as 12·86 kNm^2 and mass per unit length as 2·74 kg/m. The beams were all initially either 1·0 m or 1·414 m long and the joints at (0, 0) and (0, 1) were taken to be pinned to ground; all other joints were assumed free to move. The structure was chosen to be two-dimensional for simplicity of analysis and only motions in the plane of the structure were studied. The theoretical model was excited by a point transverse force halfway between (0, 0) and (1, 0) and, during optimization, the aim was set as the minimization of the vibrational energy level in the right-hand end vertical beam (which in practice might carry an instrumentation package or similar) between 150 and 250 Hz. The damping of the structure for the theoretical analysis was fixed so that the normal modes of the uncoupled beam elements all had a constant bandwidth of 20 s^{-1}.

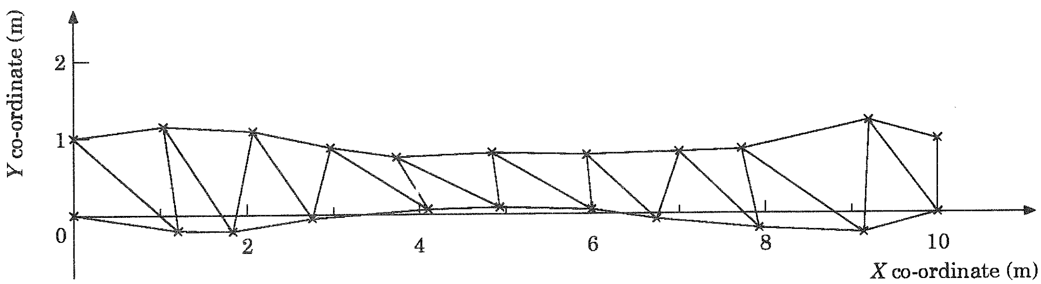

Figure 2. The GA optimized design (with limits of ±25% on 18 joint positions and using 4500 evaluations over 15 generations).

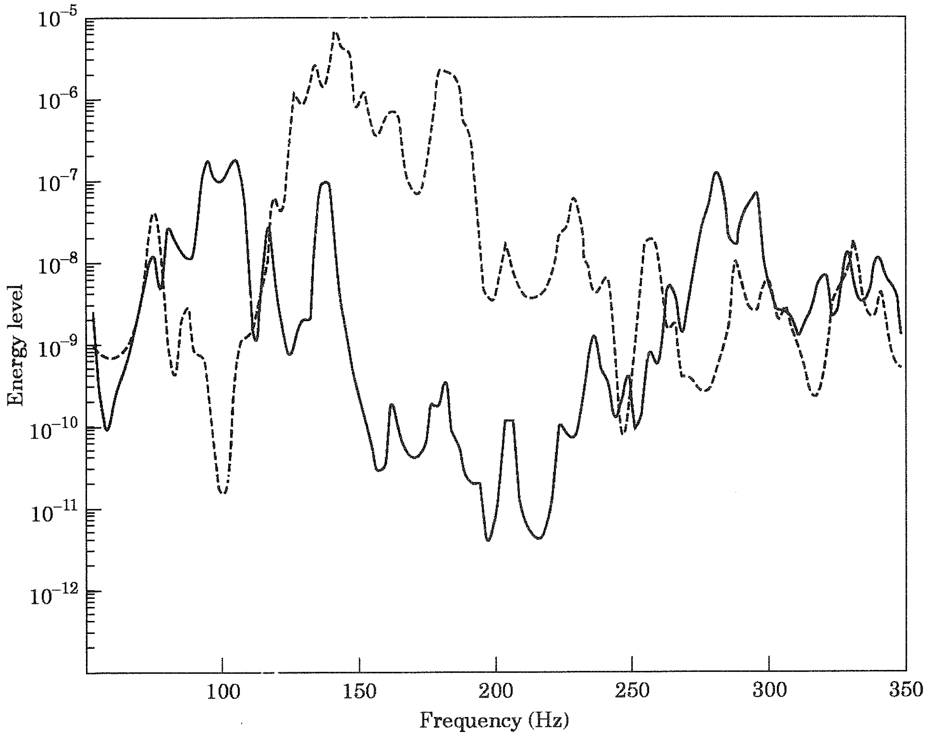

Figure 3. The theoretical frequency responses of the original (dotted) and the optimized (solid) designs.

To model this structure physically, some changes were needed since the theoretical models' overall lengths of 10 m could not be readily accommodated in the available laboratory facilities. It was therefore decided to make physical models where the fundamental natural frequencies of the individual bar elements were similar. A free–free bar with properties as per the theoretical model has a first (flexural) natural frequency given by $(9{\cdot}067\pi/8L^2)\sqrt{EI/\rho A} = 243{\cdot}9$ Hz. To achieve this first frequency the physical models were constructed out of 1/8 in by 1/2 in section 2024 aluminium alloy at a 1:0·2603 scale with the beams flexing about their thinner dimension, making the overall models some 2·6 m long. A 4:1 cross-section was chosen so as to drive the transverse vibrations of the individual elements well above the frequency range of interest here. The two models used, here shown mounted via a stiff steel structure to a 1 tonne base mass sitting on cushioned feet, are illustrated in Figures 4 and 5. As can just be made out from the figures, the models were restrained from transverse motion by three pairs of light nylon horizontal guys place at strategic places along the structures and then carried over retort stands to heavy masses. Since the horizontal lengths of the guys were many times the vertical motions being excited these did not act to restrain the vertical vibrations but considerably stabilized the rather low frequency overall horizontal bending modes of the models.

3. INSTRUMENTATION AND TESTING

The model structures were excited by using a Brüel and Kjaer (B&K) 4809 electromagnetic exciter driving through a B&K 8203 small force cell and powered by a B&K 2706 power amplifier. The responses of the end bar of the structures were measured by using a B&K 4374 lightweight accelerometer attached by a small dab of wax. The signals from

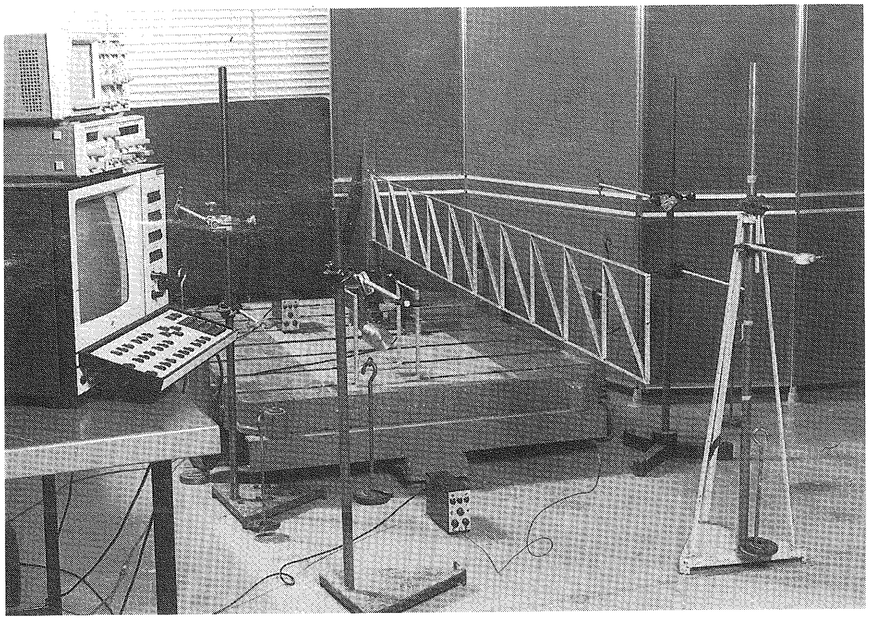

Figure 4. The experimental set-up for testing the initial structural design.

the accelerometer and force cell were taken through B&K 2635 charge amplifiers to a B&K 2304 dual channel spectrum analyzer which was also used to provide pseudo-random noise signals to the power amplifier. The drive through the force cell to the models was via a small, stiff stainless steel stinger attached with "super-glue", which prevented lateral excitations reaching the models. The whole force/response measuring chain was calibrated by being applied to 4·401 kg ballistic pendulum, which may be seen in the foreground of

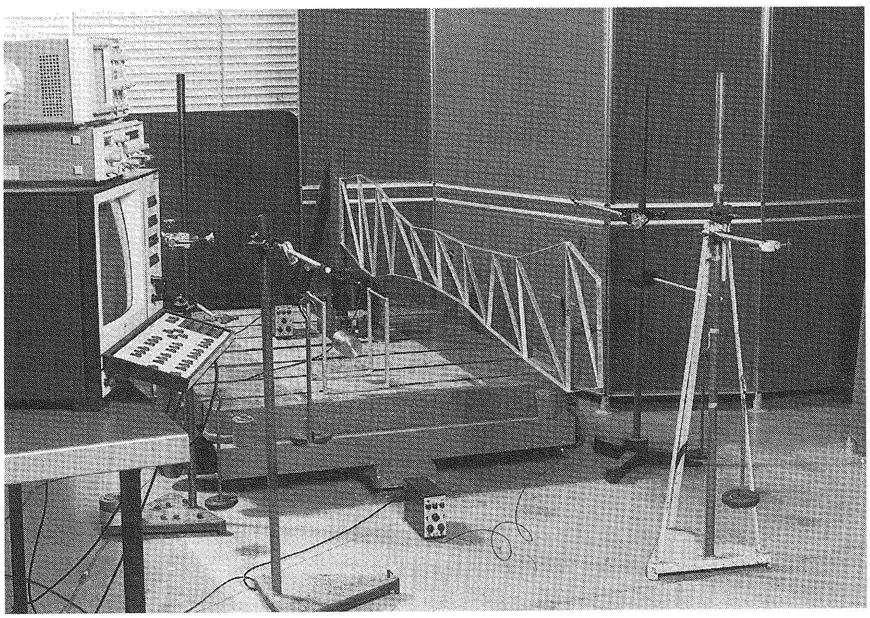

Figure 5. The experimental set-up for testing the optimized structural design.

Figures 4 and 5. The calibration constant produced in this way was within 0·1% of that predicted by using the manufacturer's calibration data for the equipment used.

Investigations of the coherence plots seen when studying these structures showed that good results could be achieved by using four 50 Hz bandwidths with averages over 100 16 second samples, each weighted with a Hanning window and 50% overlap averaged: i.e., by using nearly one hour of data per test. To build up the response curves for the structures over the range of interest here the results were downloaded from the 2304 analyzer and processed off-line. Logarithmic decrement and half power bandwidth measurements on typical peaks in the response spectra of the structures revealed that they had typical damping factors of 0·00006: i.e., they were essentially undamped. No steps were taken to increase these damping values to levels similar to those assumed in the theoretical studies.

4. RESULTS

In Figure 6 are shown the frequency response functions between force and velocity (i.e., the mobility) for the original and optimized structures for a suitably chosen response point on the end beams (because the individual beam element natural frequencies are of the same order as those being investigated here, it matters little where the response point is taken on the end beam, provided that it does not lie at a simple fraction of the element's length and thus at a node of its responses). In Figure 7 are shown the coherence functions for these two tests, which can be seen to be quite high throughout the frequency range being studied (they have average values of 0·97 and 0·98, respectively). The frequency response functions show that significant noise isolation is present, despite the very lightly

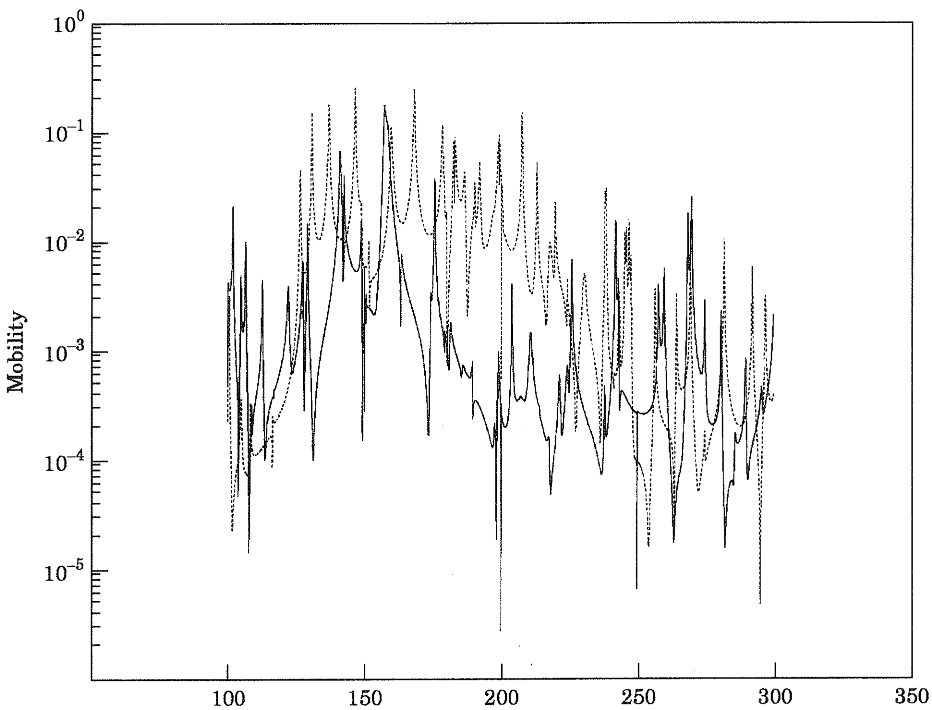

Figure 6. The measured point mobilities of the original (dotted) and the optimized (solid) designs.

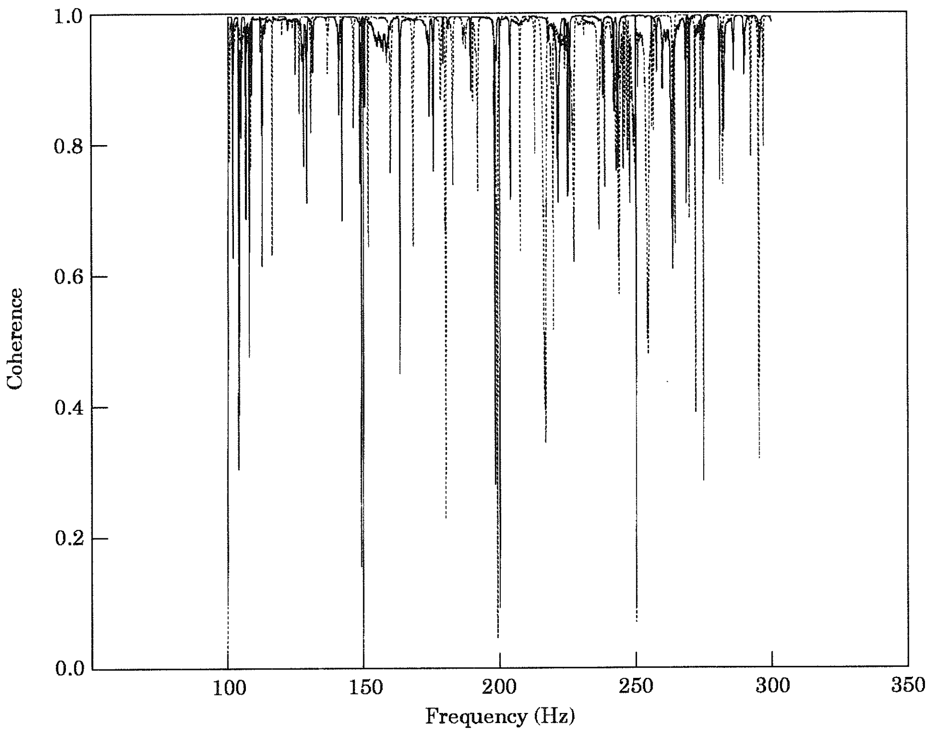

Figure 7. Coherence plots of the original (dotted) and the optimized (solid) designs.

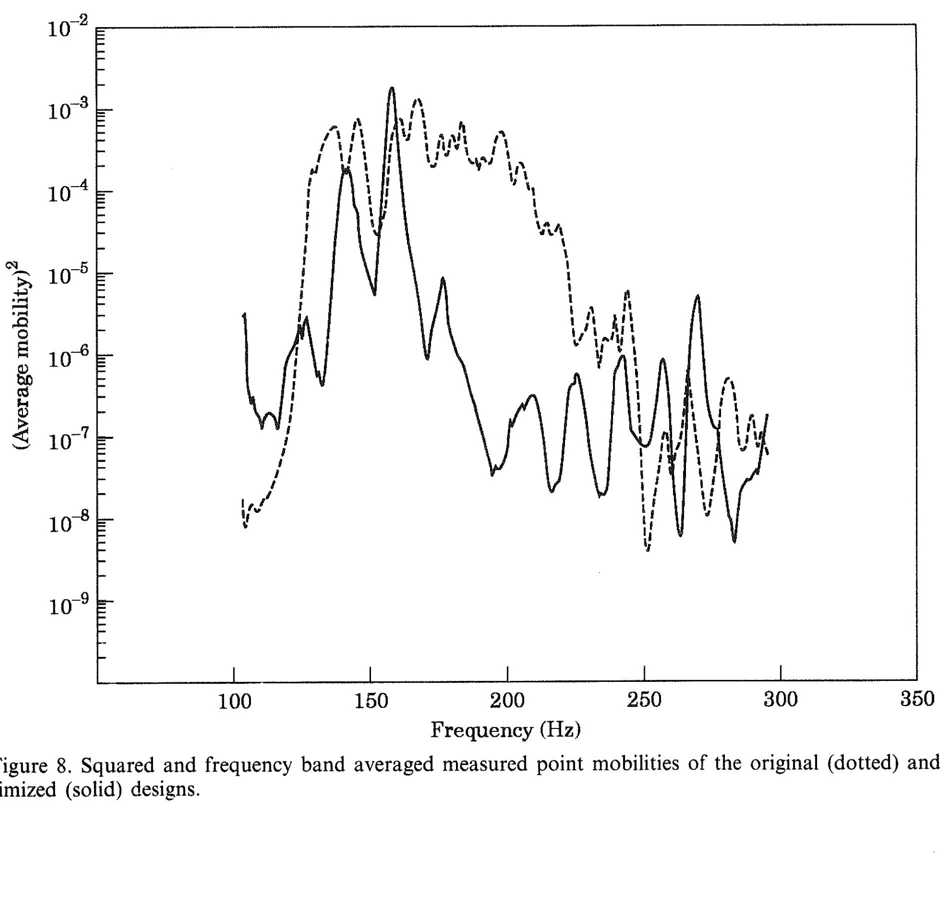

Figure 8. Squared and frequency band averaged measured point mobilities of the original (dotted) and the optimized (solid) designs.

damped nature of the structures. Moreover, it is being achieved over the desired range of frequencies.

To make this comparison more explicit, the velocity responses can be squared and then smoothed with a simple moving average (by using 3·125 Hz bands applied to the logarithmic data); see Figure 8. The data in the figure is squared to be more comparable to that in Figure 3 since it is twice the kinetic energy transfer that is plotted there, i.e., it is related to $\frac{1}{2}mv^2$; it is smoothed with a moving average to simulate the increased level of damping in the theoretical study. Comparison of this figure with Figure 3 shows that the behaviour of the actual and theoretical structures is quite similar, with similar magnitude isolations being achieved in practice to those predicted (to within a simple scaling factor needed to relate average point mobility squared to average kinetic energy). Note also that the theory correctly predicts the rather larger responses in the lower half of the 150–250 Hz band being studied although the behaviours of individual peaks are not identical.

5. CONCLUSIONS

It has been shown that significant noise isolation characteristics can be introduced into a regular structure by modifying it in a controlled way. The designs postulated in an earlier piece of work have been used to construct two model structures, each about 2·6 m long. These have been tested by using simple laboratory equipment and good results achieved. These tests have shown that it is possible to build lightweight, lightly damped aluminium truss structures that have up to three decades, or 50 dB, of energy transmission isolation between their ends, without using additional damping materials or active control methods.

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