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Active Vibration Isolation on an Experimental Flexible Structure

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ABSTRACT

The paper describes experimental research in the area of active vibration isolation. The objective of' the research is to quantitatively assess the performance gained by augmenting a passive isolator with an active stage. Vibration isolation experiments were carried out on a flexible structure utilizing a proof-mass shaker as the disturbance source and. a newly developed active member as the isolator. Broad ban d force feedback control demonstrated more than 20 dB reduction in force transmissibility over passive isolation alone. The broad band controller was augmented with notch filters which resulted in reducing force transmissibility by 40 dB over the passive stage in select narrow bands.

1. INTRODUCTION

Future precision space structures will be required to operate at nanometer level vibration environments. Noisy space machinery such as reaction wheels, tape recorders, centrifuges and rolling trolleys emmit disturban ces which after being amplified by the highly resonant dynamics of the strut.ture may result in vibration amplitudes well into the micrometer range.

Mech anical isolation is an effective design solution for the reduction of transmissibility of disturbances from one structural subsystem to another. in the last 15 years, there has been a great deal of progress made in the application of active control to enhance passive isolation, duc to advances in new actuator materials, digi tal signal processing, and to a lesser degree, control theory.[1,2,3] The literature contains dozens of references in implementations of active isolation, most of which arc for single degree of freedom systems; the m aturit y of multi- axis active isolation design is far from complete. in nearly all of the references, both the substructure and the payload are assumed to have known, non-resonant output (or input) mobilities. Local error sensors, such as base or machine acceleration, force, or gap measurements, are selected based on convenience, cost or signal to noise ratio.

Isolation is an attractive layer in the overall control design approach for controlled strut.tures[4~5] because machinery and payload mounts act as disturbance bottlenecks which are targets for active control. Local actuators and sensors lead to collocated control designs for the reduction in transmissibility of force or displacement. The flexibility of the base structure, however, presents difficulties in the design of active isolation, because resonant base dynamics can couple into the plant transfer function. It was shown for the case of disturbs.nc.c! isolation from a vibrating machine to a flexible base[6] that use of base acceleration as an error sensor leads to an unacceptably high degree of interaction with base strut.ture resonances in the feedback loop, prohibiting broadband control with robust rolloff. Yet in another study[7] base acceleration was used effectively for active enhancement of passive isolation, but in this case the base resonance frequencies were well above the passive stage corner frequency.

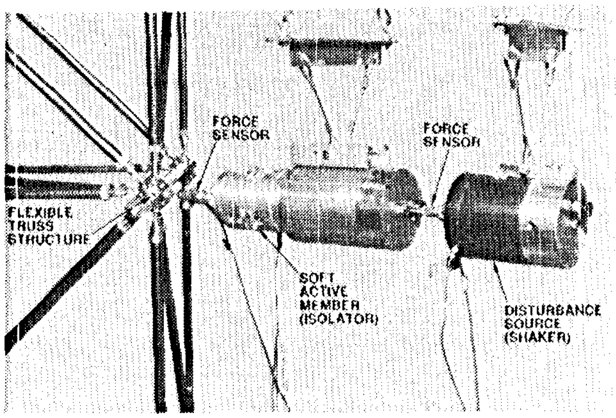

Figure 1. Vibration isolation experiment.

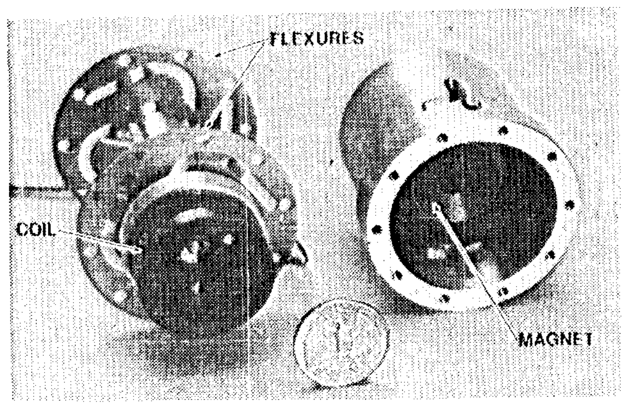

Figure 2. Soft Active Member (SAM),

The selection of actuator/sensor pairs for control has been re-examined for cases in which resonant structural dynamics are present.[8] To summarize the results of that work:

1. A passive softmount yields not only nominal isolation performance, but also permits (with appropriate sensor/actuator pairing) decoupling of the control system from resonant strut.iural dynamics.
2. Two actuators were identified for active enhancement of isolation; a.force actuator parallel to the passive softmount, and a displacement actuator in series with the passive mount. For the force actuator, the sensor which minimized coupling with resonant strut.tuml dynamics is transmitted fore.c. For the displacement actuator, the appropriate sensor is less constrained but still important.

This paper investigates the use of a force actuator (voice coil) in parallel with a passive softmount (elastic flexures). The error sensor used for feedback is the force transmitted to the base. The main objective of the research is to quantitatively assess the performance gained by augmenting a passive isolator with an active controil system. The work is motivated by the need to attenuate machine generated disturbances on board space vehicles but the results obtained are relevant to a large class of vibration isolation problems.

2. EXPERIMENT DESCRIPTION

A single-mis disturbance isolation fixture was designed, built, and installed on a precision truss structure known as the J]'], Phase B Interferometer Testbed.[5] The truss structure simulates an elastic foundation on which a vibrating machine is mounted. The Testbed is a highly resonant structure and early system identification tests[9] revealed the presence of approximately 20 lightly clamped modes below 100 Hz.

The experiment hardware consist of three components: the disturbance source, the isolator, ant] the flexible base structure. Figure 1 shows the system configuration and how the three components are interconnected along an axis perpendicular to the direction of gravity.

The disturbance source is a proof-mass shaker suspended from the c.tiling and attached to the isolation fixture via a stillger-type connector. The proof mass used in the experiments was approximately 2 Kg. A force sensor was installed between the shaker and the isolator in order to measure the force acting on the isolator. This was one of two sensors used to measure force transmissibility.

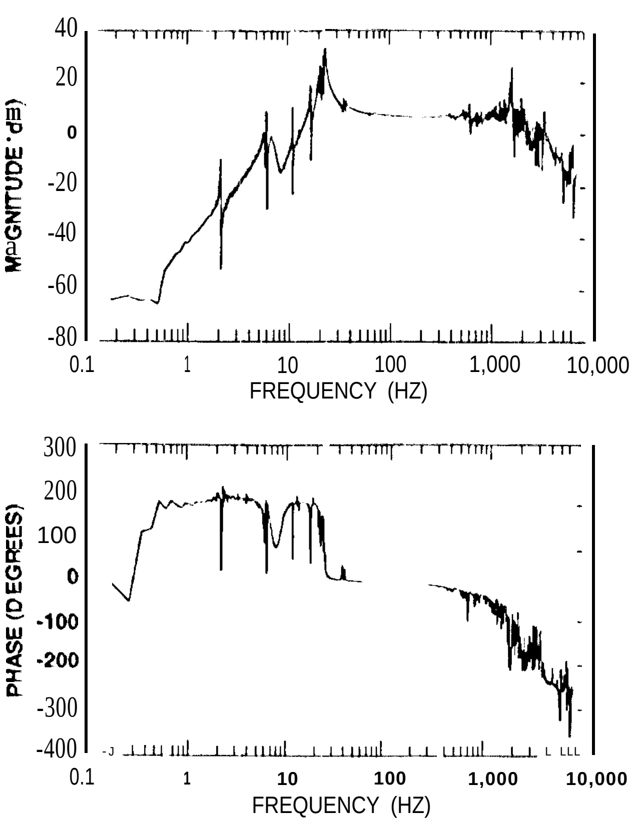

Figure 3. Plant transfer function.

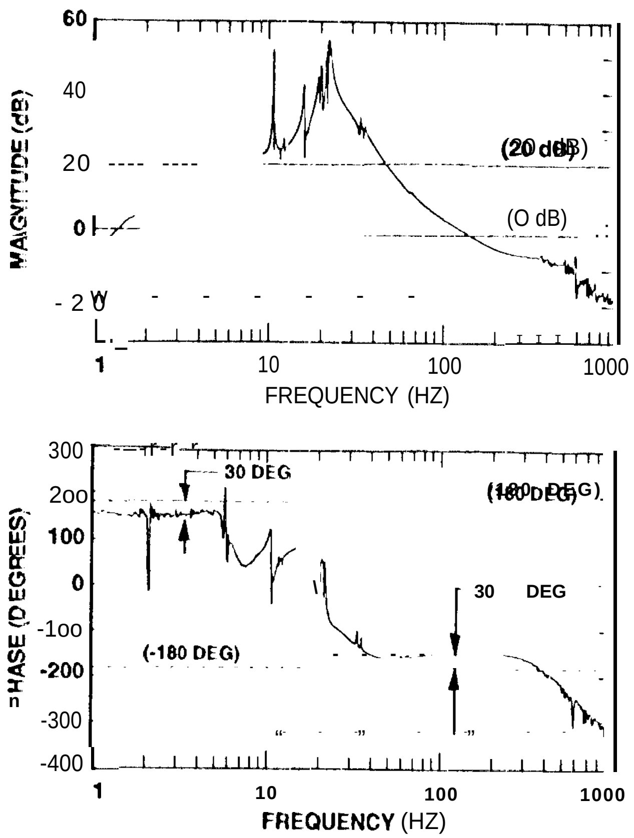

Figure 4. Loop gain and phase.

The isolator consists of a newly developed active strut along with a 4 Kg mass suspended from the ceiling. The key elements of the active strut are a set of aluminum flexures and a voice coil actuator. The hardware is shown in Figure 2. The active strut will be referred to as the Soft Active Member (SAM) since the flexures can be made sufficiently soft to satisfy different passive isolation requirements of corner frequency, load and stroke. The stiffness of the flexures used in the experiments was 70 N/mm. A gap sensor was installed inside the SAM to monitor relative displacement and detect motion saturation. A second force sensor was installed between the isolator and the truss structure in order to measure the force transmitted into the structure. This sensor was used for feedback control as well as for performance assessment.

The hybrid nature of the isolator is evident from its passive and active constituents. The 70 N/mm flexures and 4 Kg blocking mass represent the passive stage of the isolator while the force sensor along with the voice coil actuator are elements of the active stage. The mass and stiffness of the isolator were selected to achieve a 20 Hz passive mount corner frequency. This was an attempt to capture the corner frequency of the passive mount in the Hubble space telescope[10] We note that the Hubble passive mount is one of few isolation systems currently in space.

3. ACTIVE ISOLATION

The active isolation stage is introduced in an effort to increase performance beyond that obtained from

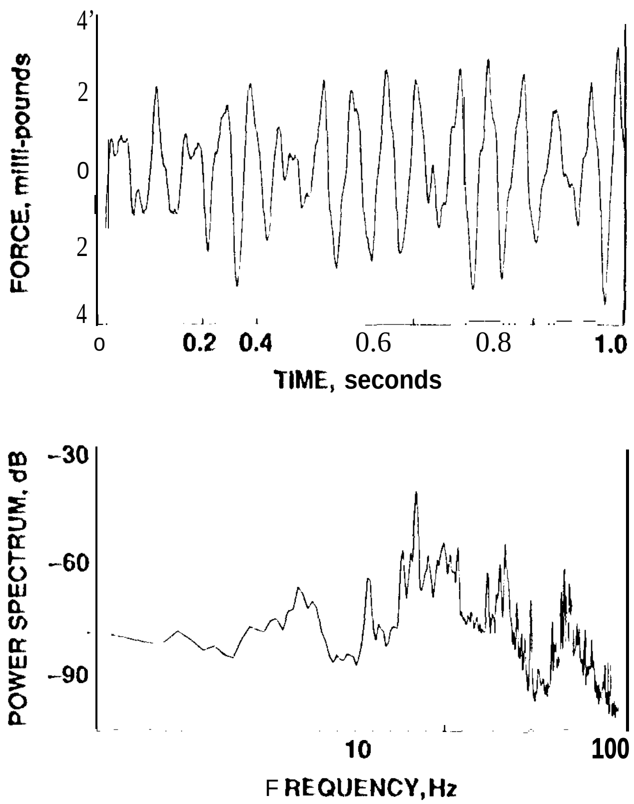

Figure 5. Open loop response to ambient noise.

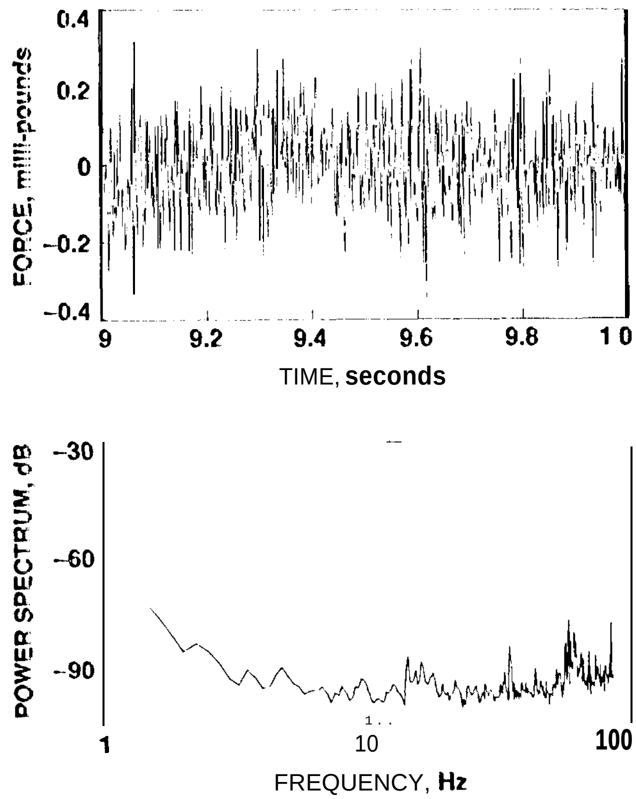

Figure 6. Closed loop response to ambient noise.

the passive stage. The work described in this section is separated into three phases: (i) measurement and identification of the plant transfer function, (ii) design and implementation of the compensator, and (iii) measurement and comparison of open and closed loop performance. Each phase will now be discussed in sequence.

The transfer function from the voice coil actuator to the force sensor was measured using an HP 3566A spectrum analyzer. Both sine-sweep and band-limited white noise excitations were used to identify the plant transfer function over the frequency range 0.2-6,000 Hz. The result is shown in Figure 3. We observe that at frequencies below the passive! mount corner frequency of 20 Hz the magnitude of the response increases with increasing frequency at the rate of +40 dB per decade. Clearly visible is the presence of eight lightly damped modes of the truss structure in the range 2-20 Hz. Above the corner frequency of the passive mount the magnitude of the response becomes flat until it eventually rolls-off above 2,000 Hz. The identified transfer function closely resembles that measured by Watters et al[6] who experimented with the isolation of disturbances generated by a Diesel engine on an elastic foundation. The result is also in agreement with analytical studies by Blackwood and von Flotow[8] who predict that use of a force sensor will largely decouple the plant transfer function from resonant base dynamics at frequencies well above the passive mount corner frequency.

With the plant transfer function identified, the control system was synthesized using loop shaping techniques

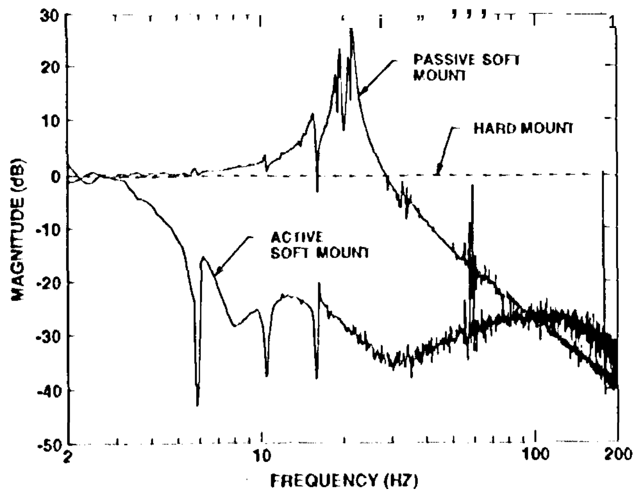

Figure 7. Isolator force transmissibility.

in the frequency domain.[11] The design objective was to reduce force transmissibility both below and above the passive mount corner frequency of 20 Hz. This is achieved by increasing the loop gain over the desired bandwidth. The compensator was designed to provide both broadband and narrowband performance. Loop shaping resulted in an 18th order design consisting of a second order lowpass filter, 4 second order note.lL filters, a fourth order lead and a fourth order lag filter. The lowpass filter provides the necessary high frequency roll-oft', the lead arid lag filters provide 30 degree phase margins at the high and low frequency cross-over points, and the notch filters boost the loop gain at the frequencies of the most dominant base resonances. The loop shape shown in Figure 4 shows that the compensator has introduced more than 20 dB of gain over approximately a decade of frequencies and more than 40 dB at select narrow bands. The design was implemented as a parallel bank of 9 second order filters on a Heurikon V3E/50MHz 68030 processor running at 4,000 Hz. The delays associated with the computer implementation (i.e., zero order hold, transport and computational delays) were taken into account and their effects arc included in the loop phase of Fig. 4.

The first experiment involved measuring the background noise in the feedback sensor. Figure 5 shows the out put of the force sensor when both the disturbance source and control system are off. The laboratory background noise is on the order of 3 mini-pounds peak. The power spectrum of the waveform further shows that it is dominated by a single frequency component at 16 Hz. Since the truss structure has a lightly damped mode at 16 Hz (see Fig. 3) it is believed that laboratory ambient noise excites the 16 Hz resonance and this motion is picked up by the force sensor between the trusswork and the isolator.

Two closed loop experiments were carried out first with the ambient disturbance and then with disturbances generated by the shaker. Figure 6 shows the output of the force sensor when the feedback loop is closed. The 0.3 milli-pounds peak amplitude implies a factor of 10 improvement (20 dB) over open loop and this represents the noise floor of the experiment under closed loop control.

With the disturbance source on, the force transmissibility of the isolator was measured with and without active control. The result is shown in Figure 7. The ‘(hard mount” represents the case of having the isolator removed and the disturbance source mounted directly to the truss structure. The "passive soft mount" shows the classic second order behavior of a second order lightly damped spring-mass system which provides 40 dB/decade isolation above its characteristic corner frequency. The "active soft mount" shows the performance improvement gained by augmenting the passive stage with an active stage.. Clearly, the active stage provides 20 dB of broadband improvement over the passive stage and 40 dB of narrowband improvement at specific frequencies, This improvement is achieved at the expense of increasing transmissibility by approximately 5 dB at frequencies outside the control system bandwidth.

4. CONCLUSIONS & FUTURE WORK

A single-axis disturbance isolation fixture consisting of a shaker in series with a newly developed active strut was installed and tested on a precision truss structure, Vibration isolation experiments were carried out utilizing the shaker as the disturbance source and the active member as the isolator. Broad band force feed-back control designed using classical loop shaping techniques demonstrated more than 20 dB of performance improvement over passive isolation. Furthermore, narrow band control resulted in reducing transmissibility by an additional 20 dB over the broad band. The main result of this research is that an active stage can significantly reduce the transmissibility of a passive isolator both below and above its characteristic corner frequency.

Future work will focus on multiple-axis isolation with an improved passive/active mount. Near term efforts will concentrate on enhancing the performance of both the passive and active stage. The passive stage can be improved by lowering its characteristic corner frequency. However, in lowering the corner frequency the challenge is to design softer flexures that can accomodate longer strokes while still maintaining a linear force-displacement relationship. The active stage can be improved by increasing the bandwidth of the controller. This can be achieved by minimizing time delays and improving the noise and bandwidth characteristics of the. control system hardware. Emphasis will also be placed on compensators with self tuning elements that can restore nominal performance despite changes in the plant and disturbance dynamics.

5. ACKNOWLEDGMENTS

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