MODULAR WIDEBAND ACTIVE VIBRATION ABSORBER

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ABSTRACT:

A comparison of space experiments with previous missions shows a common theme. Some of the recent experiments are based on the scientific fundamentals of instruments of prior years. However, the main distinguishing characteristic is the embodiment of advances in engineering and manufacturing in order to extract clearer and sharper images and extend the limits of measurement. One area of importance to future missions is providing vibration free observation platforms at acceptable costs.

It has been shown by researchers that vibration problems cannot be eliminated by passive isolation techniques alone. Therefore, various organizations have conducted research in the area of combining active and passive vibration control techniques. The essence of this paper is to present progress in what is believed to be a new concept in this arena. It is based on the notion that if one active element in a vibration transmission path can provide a reasonable vibration attenuation, two active elements in series may provide more control options and better results.

The paper presents the functions of a modular split shaft linear actuator developed by NASA's Goddard Space Flight Center and University of Massachusetts Lowell. It discusses some of the control possibilities facilitated by the device. Some preliminary findings and problems are also discussed.

INTRODUCTION:

This work was prompted by discussion of the vibration control terminology such as isolation, suppression, and cancellation. The authors were evaluating the effectiveness of applying feed-forward acoustic noise cancellation techniques to mechanical vibration problems when this concept evolved. Berkman and Bender [1] defined active vibration isolation as reduction of the transmissibility of force or vibratory response across discrete structural connections. Active suppression was defined by them as "reduction of the global vibratory response of a structure using control actuators at locations that do not coincide with discrete power insertion points."

Vibration isolation in this paper also refers to reduction of transmissibility of force in a mechanical connection. However, vibration suppression will be thought of as the force cancellation across the connection. Therefore, isolation across a two stage connector is achieved by connecting one stage of the coupler to the structure so that it will move in unison with its vibration, while the other stage is attached to a detector or a device which is to be free of vibration. As such, the best theoretical isolation is achieved when the two stages of the coupler are physically disconnected. Vibration suppression in this paper will refer to a situation where a force equal and opposite to the source of vibration is applied in order to stabilize the vibration source itself. In this case, the coupler would see maximum force but

minimum movement. The coupler discussed in this paper is believed to facilitate position control as well as several combinations of isolation, suppression, and tuned vibration absorption in one package.

DEVICE DESCRIPTION:

The final design of this device evolved from an initial search for a mechanism that would connect two components together while acting as a "non-contact" or "virtual" coupler. Since this non-contact coupling is impossible with passive elements, such a device was first termed a "zero force" actuator. Later, when the concept of an actuator was a more developed, it was dubbed the "split shaft" actuator since it looks like a linear motor whose shaft is broken in the middle of the housing as shown in Figure 1. After further exploration of potential uses of this actuator, it was concluded that this multifaceted device could also be used as a tuned passive vibration absorber, and provide vibration attenuation over a wide range of frequencies. From then on, this modular device was termed the "Modular Wideband Active Vibration Absorber" or MWAVA.

At the two magneto-mechanically coupled ends of this device are two voice coil actuators. One of the two light weight bobbins is to be attached to the vibrating structure and will move with the disturbance in order to isolate its stator from high forces. Therefore, the bobbin should move with the same vibration amplitude as that of the source. The second actuator will then be connected to a target device which is to be held stationary. Obvious, this device can also be used as a servo system for positioning a detector, with a possible loss of some of its vibration control effectiveness. The three basic components of this device are a left bobbin, a right bobbin, and the actuator housing that contains the magnetic circuit and referred to as the proof mass.
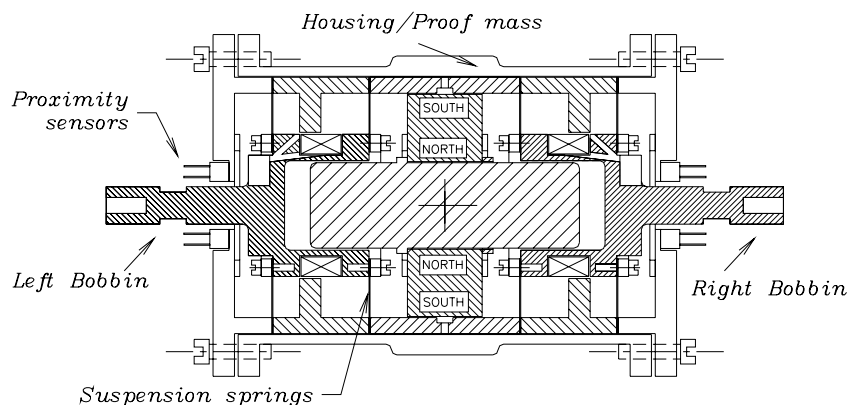

Figure 1. Assembly drawing of the Modular Wideband Active Vibration Absorber

The intended use if this device is to attach one bobbin to a structure and connect the other bobbin to a detector or any device that is to be isolated from a vibrating structure. The actuator housing is left free to "float" as the mass of a vibration absorber or proof mass damper. The two bobbins are connected to the proof mass by a pair of disc type suspension springs. Details of the actuator are described by Smith et al [2]. There are proximity sensors at each end of the device in order to detect the relative position of the proof mass with respect to the left $X_{(P/L)}$ and the right $X_{(P/R)}$ bobbins. Another sensor is needed in order to determine the location of each component with respect to an inertial reference system.

CONTROL OVERVIEW:

Typical control systems are composed of plants, controllers, feedback elements and transducers. The original concept was to use one end of the M-WAVA as a vibration isolation stage and the other end to maintain a stable platform. An overview of the system is shown in Figure 2. The three components of this device are shown together with various control options in this figure.
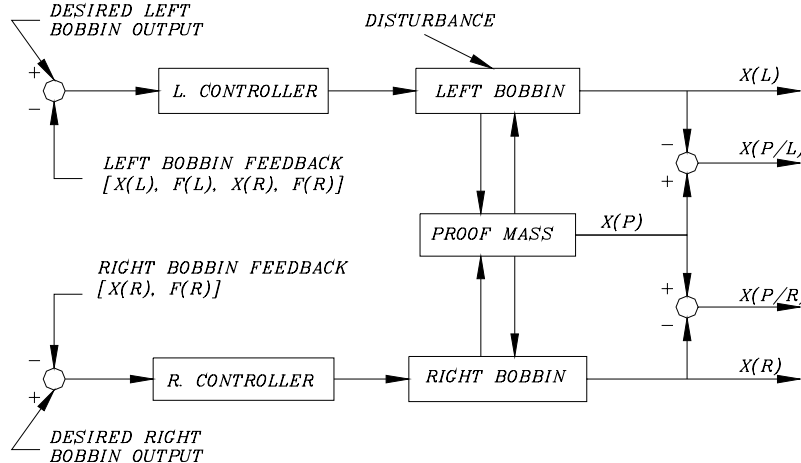

Figure 2.
Overview of the Modular Wideband Active Vibration Absorber

The letters and subscripts L, R, and P in this paper refer to the left bobbin, the right bobbin, and the proof mass, respectively. One example of the versatility of this device is that the disturbance which acts on the left bobbin can be either a force source or a displacement input. Since one can use two controllers, we have several options as to what feedback signal to use for each one. As shown in this figure, there are four different feedback options for the left bobbin controller. These signals are left bobbin vibration, left bobbin net force, right bobbin vibration, or right bobbin force. Any one of these options can be use with any of the two feedback signal options for the second stage. The feedback signals for the right bobbin can be the right bobbin force or vibration. The intent of this paper is not to

find an optimum control arrangement or philosophy, but to show preliminary findings from selected combinations.

VIBRATION INPUT:

This section assumes that the left bobbin is connected to a vibrating structure. Therefore, the mass of the left bobbin becomes part of the structure and drops from the governing equations. The two normal mode frequencies, neglecting insignificant damping for this two degrees of freedom system, is defined by equation 1. In this equation,

$$\lambda_{1,2} = \frac{K(2M_R + M_P) \pm \sqrt{[K(2M_R + M_P)]^2 - 4K^2 M_P M_R}}{2M_P M_R} \qquad \ldots \qquad \ldots \qquad (1)$$

$\lambda = \omega^2$, ω is the natural frequency in radians/second, M_R and M_P are the masses of the right bobbin and the proof mass, respectively. In this equation, K is the spring constant which is the same for both ends. Substitution of numeric values into this equation gives the two normal mode frequencies as 3.4 and 17.2 Hz. If we add a couple of controllers and feedback loops, the resulting system block diagram looks like the one shown in Figure 3. As shown in this figure, a sinusoid of 0.1 inch (2.54 mm) amplitude and 4 Hz is applied as the vibration disturbance. This rounded frequency is chosen because it is close to the first natural frequency of the system. The steady state vibration of the right bobbin without any active controller is a sinusoid of amplitude 0.16 in. This means that, at this particular frequency, the output bobbin has a higher level of vibration than the left bobbin by a factor of 1.6 because the forcing frequency is close to one of the natural frequencies of the system. When the two controllers are active, the vibration level of the right bobbin drops to 0.0027 inch, giving a 37 fold (31 dB) reduction of vibration. A review of the frequency response plots ($X_{(R)}/V_{(L)}$) show a minimum attenuation of 35 dB over the entire range of frequencies. The reduction at the first mode frequency of 3.4 Hz is 66 dB and at the second mode frequency of 17.2 Hz is 58 dB.

The controller in this figure has only proportional and integral elements and no effort has been made to design an optimum compensator at this time. In this arrangement, any motion of the right bobbin is perceived as error by both controllers. Other ideas worth explored for left bobbin feedback are vibration of the proof mass or net force on the left or right bobbins. Some preliminary experimental results of this concept are discussed in Smith et al [3].
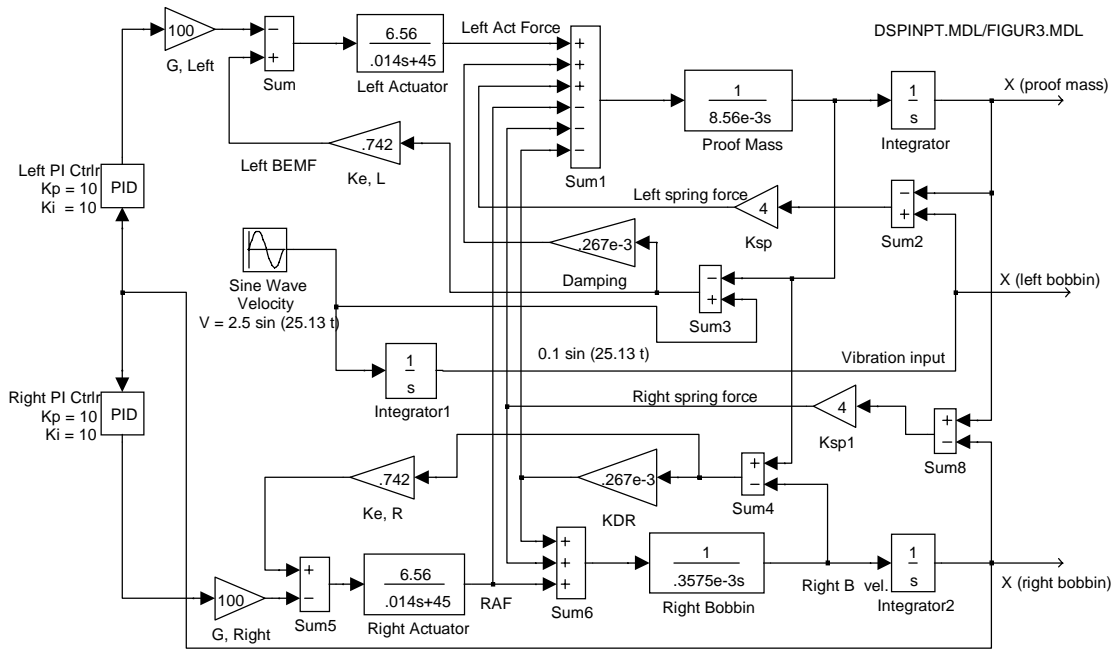
Figure 3. System block diagram with vibration (displacement) input and two PI controllers
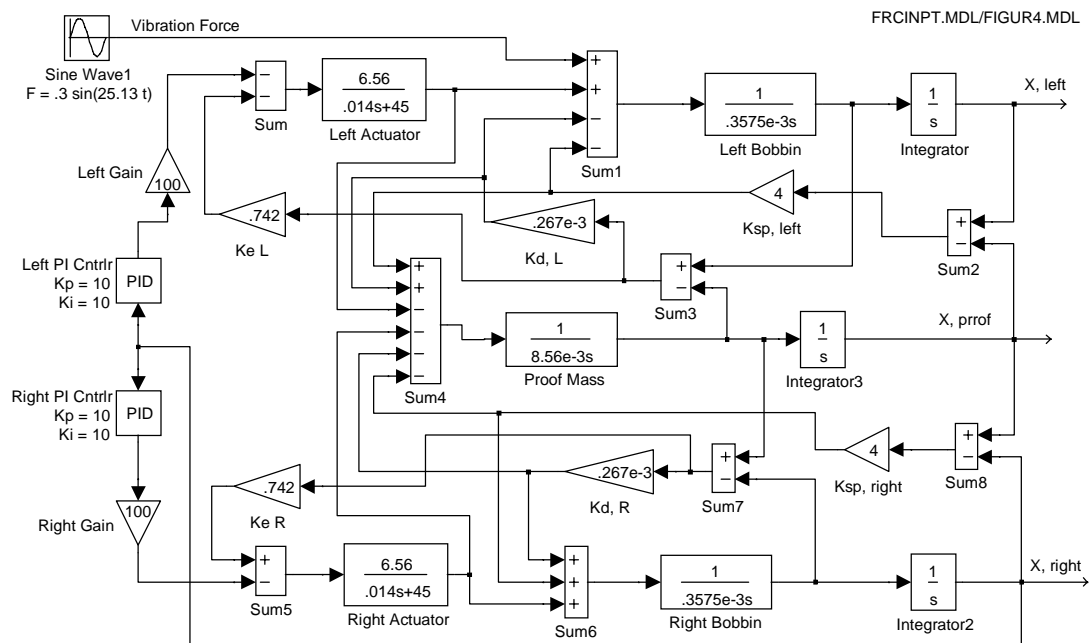
FORCE INPUT:

There are situations when not only does a vibrating device need to be isolated from a "quiet" structure to which it is attached, but its own vibration also needs to be reduced. An example of this application is reciprocating devices such as sterling cycle coolers. A sterling cycle cooler may need to be isolated from the structure at the same time that it's cold plate is to be free of vibration because it is attached to a precision detector. In this case, the vibration force is finite and might be suppressible. Here, the left bobbin can actively fight the vibration force without being overwhelmed and the system becomes a three mass system. Unlike vibration disturbance in the previous section, the mass of the left bobbin cannot become part of the structure and, therefore, appears in the governing equations and in the system diagram. An analysis of the free vibration of this system yields a characteristic equation whose solution indicates a degenerate three degree freedom system with one zero root and only two elastic vibration frequencies [4]. The elastic frequencies can be determined from the following equation:

$$\lambda_{1,2} = 0.5\,[\,K(\frac{1}{M_R} + \frac{1}{M_L} + \frac{2}{M_P}) \pm \sqrt{[K(\frac{1}{M_R} + \frac{1}{M_L} + \frac{2}{M_P})]^2 - 4K^2(\frac{M_L + M_P + M_R}{M_P M_L M_R})}\,]$$

....(2)

By substituting the numeric values of the parameters, the two normal mode frequencies extracted from equation 2 are 16.84 Hz and 17.52 Hz. Frequency response plots using a computer aided design tool yield peaks at about 17.5 Hz for X_R/F_L and X_P/F_L. The X_L/F_L, plot shows a double peak in the

neighborhood of 17 Hz and an anti-resonant frequency of about 3.1 Hz. The frequencies resulting from the simulations match the two frequencies above, as well as an anti-resonant at the first fundamental frequency of the previous section. A system model of this concept with two controllers is shown in Figure 4. The disturbance force in this case is only one of the forces acting on the left bobbin.

If we remove the controllers from Figure 4 and apply a sinusoidal force of 4 Hz to the left bobbin, the motion of the three masses appears as sinusoids riding on boundless ramps. This would be because the system is not anchored to an inertial system, and thus the three masses float due to computation tolerances. Despite this, it was decided to ignore the anomaly and see what would happen if we installed a couple of controllers to the system as shown in this figure.   This time, we got a surprising result that remains un-explained at this time. The left bobbin reached a steady state oscillating between -.025 and +0.1 and the right bobbin reached a steady state oscillation of 0.002 inch after riding on an over-damped trajectory. The unexplained phenomenon is that the proof mass continues to oscillate on an unbounded ramp which is a physical impossibility! In order to gain a sense of the usefulness of this control architecture, limit bumpers were inserted in the simulation model to limit the motion of the housing. By doing so, the overall system remains within bound. The left bobbin oscillates between ±0.07, the housing oscillates between ±0.09, and the right bobbin vibration is limited to ±0.005 inch. This gives us a 14 fold reduction or 23 dB attenuation at this frequency. A review of the frequency response plots $(X_{(R)}/F_{(L)})$ show a minimum attenuation of 30 dB over the entire range of frequencies. The reduction around the resonant frequencies of 17 Hz is 110 dB.

Figure 4. PI controller with force input and displacement feedback

CONCLUSIONS:

The use of vibration as the system input for straight two-stage vibration isolation yielded the expected results. In this case, the source of vibration is assumed to be infinite and the three mass system is reduced to a two mass system. However, there was an unexpected, and unexplained, problem when the disturbance was a finite force source. It is hoped that this paper will spark the curiosity of other researchers and inspire them to examine the concepts presented in this paper.

It should be stressed that, in conventional vibration absorber applications, there are two frequencies at which the motion of both masses grows without bound while, theoretically, total vibration absorption of the primary mass takes place only at one frequency [4,5]. If this device is to be used as a vibration absorber, it will become a two mass system and this point should be addressed in the design or selection of its suspension springs and active control system.

ACKNOWLEDGMENT:

REFERENCES:

1. Berkman, E.F., Bender, E.K., "Perspectives on Active Noise and Vibration Control", Sound and Vibration, pg 80-94, 30th Annivarsary Issue, January 1997

2. Smith, D.R., D'Angelo, J.P., Zewari, S.W., Blumenstock, K., Lee, K., and Sturmfels, R., "Design and Prototyping of an Actuator for Vibration Control", 6th International Symposium on Smart Structures and Materials, Newport Beach, CA, March 1999.

3. Smith, D.R., D'Angelo, J.P., Lee, K., and Sturmfels, R., "Actuator-based Vibration Control using Position Feedback," to appear in the 1999 SPIE Symposium on Smart Structures and Materials, Newport Beach, CA, March 1999.

4. Thomson, W.T., "Theory of Vibration with Applications," Prentice Hall, Inc., 1981

5. Cochin, Ira, "Analysis and Design of Dynamic Systems," Harper & Row, Publishers, 1980.

APPENDIX

LIST OF ABBREVIATIONS AND ACRONYMS:

λ	Natural frequency squared (ω^2)
ω	Natural frequency (rad/sec)
dB	Decibels
GSFC	Goddard Space Flight Center, USA
Hz	Cycles per second
K	Spring stiffness
L	Left bobbin
M	Mass
MWAVA	Modular Wideband Active Vibration Absorber
NASA	National Aeronautics and Space Administration, USA
P	Proof mass
PI	Proportional and Integral Control
R	Right bobbin
V	Velocity
X	Displacement

NUMERIC VALUES OF SYSTEM PARAMETERS:

K	Spring constant per bobbin	7.0 N/cm	4 Ib/in
Ke	Actuator Back EMF	0.292 V/cm/sec	0.742 V/in/sec
Kf	Actuator force constant	29.18 N/Amp	6.56 Lb./Amp
L	Actuator Inductance	0.014 H	0.014 H
M_L	Left bobbin mass	0.0626 kg	0.0003575 Lb. sec^2/in
M_P	Proof mass (Actuator housing)	1.5 kg	0.00856 Lb. sec^2/in
M_R	Right bobbin mass	0.0626 kg	0.0003575 Lb. sec^2/in
R	Actuator resistance	45 Ohms	45 Ohms