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DESIGN OF ACTIVELY TUNED, THERMALLY CONTROLLED
TUNED-MASS DAMPERS

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The scenarios for the strategic defense systems will require that, for their success, the effects of structural dynamics be reduced. In many SDI structures, one vibrational mode plays a dominant role in the dynamic response, such as the first mode of a metering structure during a slew maneuver. Passive damping using tuned-mass dampers (TMD's) is a well-known, weight-efficient approach to suppressing these vibrations; only a small amount of added weight is needed to achieve relatively high levels of damping. However, to be effective, TMD's must be kept tuned to the frequency of the offending mode. A prototype TMD that would tune itself to an offending mode has been designed and built, and it will keep itself tuned to that mode, even if that mode changes frequency.

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19. The properties of viscoelastic materials (VEM's) used in TMD's are temperature dependent, which is usually a disadvantage. In the present effort, the temperature dependence has been exploited as a mechanism for tuning the TMD. This was accomplished by developing and implementing an active control system where the dynamics of the TMD and base structure are sensed and the temperature (and thus the stiffness) of the viscoelastic material in the TMD is controlled for optimum tuning. The thermal control will also allow the TMD to work in the harsh space environment. The ultimate goal of this research is the production of a self-contained, "smart" TMD: one that can adjust to gradual changes such as fuel expenditure or sudden changes such as docking of an orbiter or release of a missile.

The "smart" TMD can produce the same damping level as a Proof Mass Actuator but has many advantages, such as lower power requirements (and thus weight) and no detrimental effects on the structure if power or control is lost.

FOREWORD

This report documents the work performed under a SBIR Phase I contract issued in response to SDIO SBIR Topic 88-12, "Space Structures." The specific topic addressed was the feasibility and design of a self-tuning, or "smart," tuned-mass damper. This work was sponsored by SDIO/IST and contracted by the Air Force Systems Command under Contract No. F33615-89-C-3214. The technical monitor was Mr. Arnel Pacia of WRDC/FIBG, and the contract negotiator was Ms. Kathy Walston of ASD/PMRNA.

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1. Significance of the Problem and Summary of Phase I

The scenarios for the strategic defense systems will require that, for their success, the effects of structural dynamics be reduced. Techniques that would reduce the dynamic motion of SDI's sensitive structures fall into two categories: vibration isolation and vibration suppression. Each of these can be further divided into categories of active and passive. Vibration isolation means reducing the transmission of energy from the sources to the sensitive component, usually by reducing the transmissibility of the connecting structures. Vibration suppression is directed towards reducing the dynamic motion that a component experiences by mitigating that motion. Passive damping and active control systems are two very well known approaches used to suppress the effects of vibrations on sensitive components. Passive damping can improve system performance by both suppressing vibrations and easing the task of designing active control systems. The ability to design, predict, and apply passive damping accurately will be a critical, perhaps enabling, technology requirement for SDI systems.

Passive damping can be added to a structure through a variety of mechanisms, including constrained layer treatments, discrete dampers, and tuned-mass dampers. Each damping treatment performs best for certain classes of vibration problems. Constrained layer treatments can be designed to suppress many types of modes, and they are usually effective over a wide frequency range. Discrete dampers include nonsurface treatments such as damped links, damped truss members, and damped truss joints. The applications for the damped truss elements are evident, and the damped links are most applicable where high relative displacements exist between portions of a structure.

Quite often, one vibrational mode plays a dominant role in the dynamic response, such as the first mode of a metering structure during a slew maneuver. Passive damping using tuned-mass dampers (TMD's) is a well-known approach to suppress vibration of a single mode (or a group of modes). Its main advantages over the other types of passive damping treatments is that only a small amount of added weight is needed to achieve relatively high levels of damping, and it has minimal side effects on primary structure design.

A TMD is itself a simple damped structure that when applied properly acts to suppress a single vibrational mode of a base structure. The addition of a TMD to a structure adds another natural frequency. When properly tuned, the original offending mode of the base structure and the TMD's mode couple together to result in two highly damped modes, one on either side of the original frequency. Figure 1 shows the effects of adding a TMD to a structure having widely spaced modes.

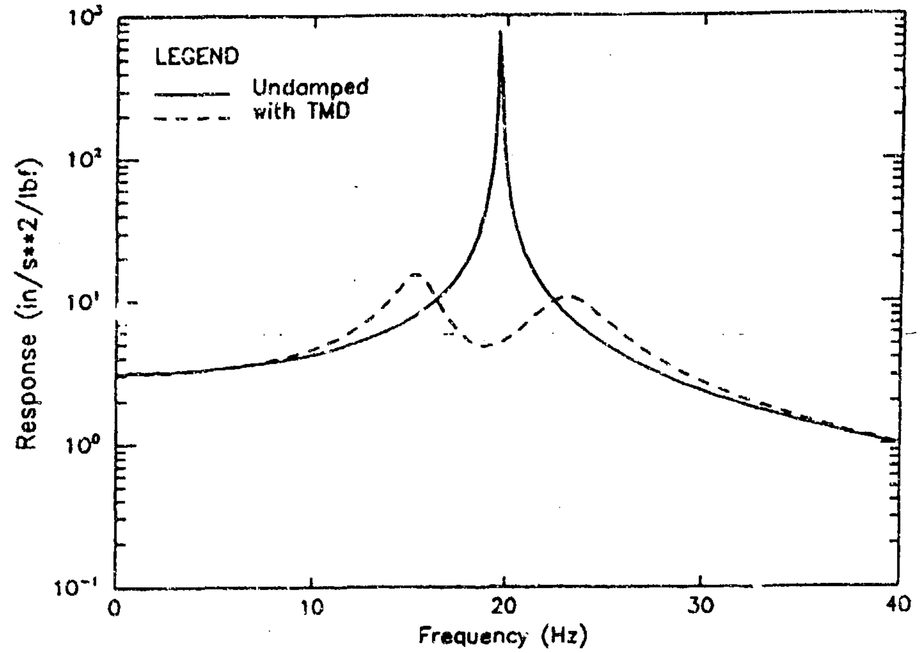


Figure 1. Effects of a tuned-mass damper

To be effective, TMD's must be kept tuned to the frequency of the offending mode. The simplest TMD may be thought of as a mass on a complex spring, i.e., spring with both stiffness and damping properties. Changing the stiffness of the spring changes the frequency of the TMD.

Viscoelastic material properties are temperature dependent, which is usually a disadvantage, but in the present effort, the temperature dependence has been exploited as a mechanism for tuning the TMD; it will work to maintain optimal tuning no matter the cause of the mistuning. Therefore, if the complex spring is a viscoelastic spring, changing its temperature will change its stiffness, thereby changing the frequency of the TMD.

The goals of Phase I were to investigate the feasibility of designing, and then build, a prototype TMD that would tune itself to the offending mode and keep itself tuned to that mode, even if that mode changes frequency. It was proposed to design and test an active control system where accelerometers would be used to sense the dynamics and a thermal system would be used to control the temperature (and thus the stiffness) of a viscoelastic material in the TMD. The control system would keep the TMD tuned to the mode as well as maintain the thermal environment for space applications. Since only a small amount of viscoelastic material is used, very little power will be required to achieve high levels of damping.

For realistic applications, CSA required the control system met two requirements:

1. The natural frequency and damping of the mode of the base structure to be damped (hereinafter referred to as "the base mode") is initially unknown and

can change with time in a continuous or noncontinuous manner. For instance, a slowly decreasing fuel load or articulated members would produce a slowly changing natural frequency, while the docking of an orbiter or release of a missile would produce a near instantaneous change in natural frequency.

2. The excitation is unknown in type and level, and both may change in time. The excitation (and thus the response) may also drop below the sensitivity of any sensor used by the TMD. This loss of excitation and response must not result in a wrong action by the controller.

These requirements provide the broadest, most realistic view of how a TMD might be required to interact with a large space structure. However, these requirements severely limit the types of information available to the controller. In effect, the controller has virtually no a priori knowledge of the base structure or how it will be excited. This means that the controller will have to work soleiy on the basis of response measurements and assumptions about the underlying structure, TMD, and the control objectives.

Phase I was successful! A prototype control system was derived which could change the temperature of a viscoelastic material so that the TMD tuned itself to the base mode and kept itself tuned, even when the mode shifted in frequency. A prototype TMD was then designed and tested. The prototype confirmed that a "smart" TMD could be built.

A schematic of what such a device may look like is given in Figure 2. The solar cells will supply the power required for the TMD and its control system i.e., it is self-contained. The "mass" will actually be the batteries to store energy and the electronics for the control system. Thin-foil heaters will be embedded in the viscoelastic material for thermal control. The entire unit will be insulated to minimize the total amount of power required. Once the TMD is tuned, very little power will be required to maintain the tune or shift the stiffness to follow a mode.

The smart TMD should be compared to a Proof Mass Actuator (PMA), an active control device used to suppress vibration modes. Both the TMD and the PMA should be placed at a point on the structure with large modal deflections. Both the TMD and the PMA use a moving mass to suppress vibrations, but the similarities stop there. A PMA uses a linear motor to move the mass out of phase with the vibrations of the structure. This requires large amounts of power and a corresponding large amount of mass that must be put in orbit to supply that power. On the other hand, the TMD's mass is moved by the vibrations of the structure and damping is obtained by the loss properties of the viscoelastic material, i.e., the displacement of the TMD is out-of phase with the base structure due to loss properties (phase lag) of the viscoelastic material. A very small amount of power (and resulting weight) is required to keep the TMD tuned. If power is lost to the PMA, major problems can occur since the mass will not be supported (or will be supported by a light spring against which the motor must operate). On the other hand, loss of power to the TMD will cause the TMD to become untuned, but the mass and its mode will be damped, causing no harm other than the mode to be suppressed will not be damped.

In summary, the overall objective of this SBIR effort (Phases I -- III) is to develop, demonstrate, produce, and market a tuned-mass damper that will achieve and maintain optimal performance with no "outside" assistance. The TMD will be self-contained in both its control system and power. Its performance will be controlled by tuning the stiffness of the viscoelastic material by controlling its temperature. The thermal control will also allow the smart TMD to operate in the harsh space environment. Once optimal tuning is achieved, the TMD will have the ability to "follow" the target mode if this mode changes due to fuel depletion, crew movement, orbiter docking, etc.

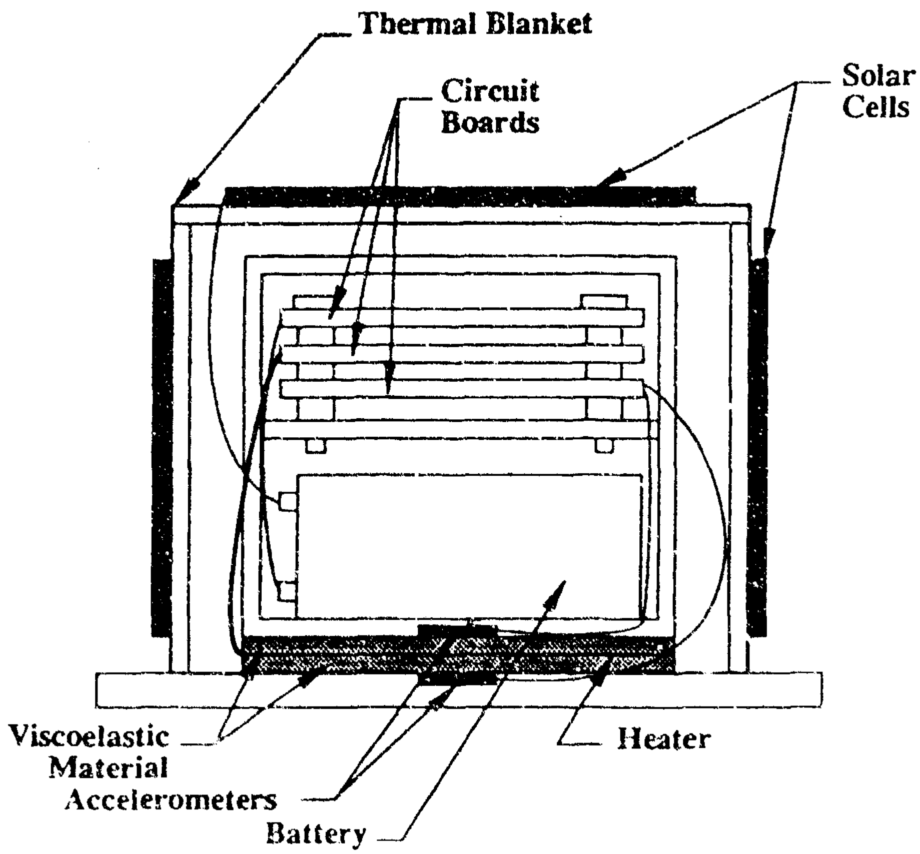


Figure 2. Schematic of a smart TMD

2. Technology Development

To achieve the overall objectives discussed earlier and demonstrate the feasibility of the concept, there were several issues to be addressed before the demonstration article could be designed and built. Tasks to be done were to

- develop a practical method and control strategy for automatic tuning,
- understand tuning of TMD's for complex structures,
- investigate heating elements and heating-control instruments.

The first task is the key to this effort and much of the effort was devoted to its development.

2.1 Background on Tuned-Mass Dampers

A tuned-mass damper, also known as an auxiliary mass damper, is a vibration-damping device consisting of a mass and a damped "spring" attached to a structure at or near an antinode of a troublesome mode of vibration. The damped spring is normally composed of a viscoelastic material. This device is capable of damping either one mode or several very closely spaced modes and is usually more weight-effective than other types of damping (passive or active). It must be tuned precisely, however, in order to function properly, i.e., specific values of spring constant, damping loss factor, and mass are critical to the successful design of the TMD. A TMD will split the base mode into two modes, one lower and one higher in frequency. An optimally tuned TMD will damp these two modes equally.

There is no standard way of constructing a TMD. It may be as simple as a block of metal attached to the base structure via a complex, i.e., damped, spring. A simple, often effective type of spring is a pad of high-loss viscoelastic material (VEM) acting either in compression or shear. Two simple TMD's are shown in Figure 3.

Experience has shown that simple TMD's are fairly easy to build for frequencies above about 50 Hz, but are difficult to construct and test in Earth's gravity field for lower frequencies. To achieve a low resonant frequency, the VEM must be very soft relative to the tuning mass. When in the Earth's gravity field, at some combination of mass and stiffness, the VEM spring can no longer support the tuning mass without beginning to creep, a common problem with highly stressed VEM's. This will not be a problem with TMD's for space applications, but must be considered here for testing and demonstration purposes.

An alternative TMD for low-frequency applications is a damped sandwich beam. There are several advantages of the damped beam: i) since the VEM "spring" is

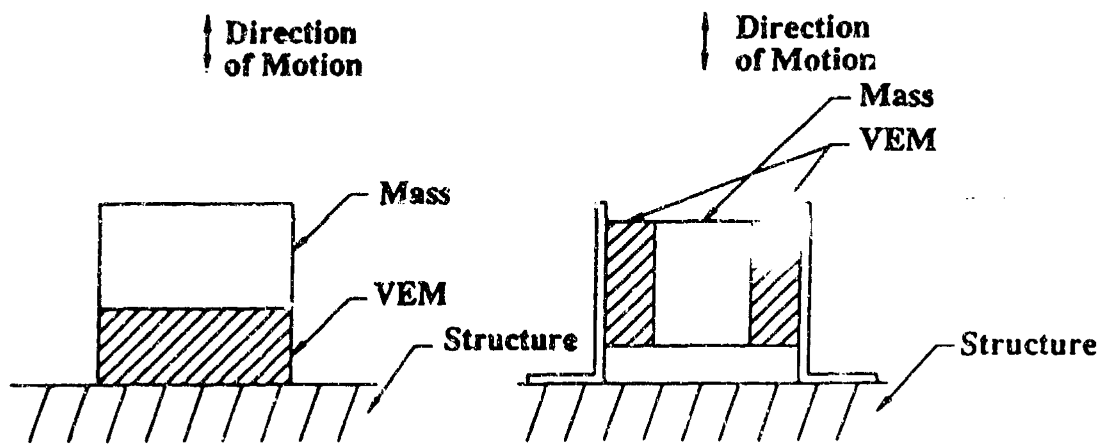


Figure 3. Typical simple TMD's

in parallel, with the facesheets, creep is much less a problem; and 2) it is easier to design a damped beam to have a specific loss factor since many combinations of facesheets and VEM's can be used; and for test purposes, the tuning frequency is easily adjusted by changing the length of the beam. Another advantage of a cantilevered beam is that the TMD can be designed so that its first mode is the mode used to provide the damping of the structure.

2.2 Deriving the Control

The parameters that may be varied for a tuned-mass damper are its tuning mass and the stiffness and loss factor of its complex spring. In a simple laboratory spring-and-mass TMD's, mass is added or removed to achieve to tune the TMD to the desired frequency. When seeking a tuning algorithm suitable for automatic control, changes in a mass are impractical. However, the shear modulus of a VEM can be affected by controlling the temperature of the VEM.

The main concept of this project is that a TMD may be actively tuned to provide optimal damping for a changing structure by controlling the temperature of the VEM. For purposes of deriving a specific control, precise limits and assumptions must be stated.

2.2.1 Assumptions Used in Deriving the Control

As stated previously, the most intriguing assumptions that must be worked with are as follows:

1. The natural frequency and damping of the mode of the base structure to be damped (hereinafter referred to as "the base mode") is initially unknown and can change with time in a continuous or noncontinuous manner.
2. The excitation is unknown in type and level, and both may change in time. The excitation (and thus the response) may also drop below the sensitivity of any sensor used by the TMD. This loss of excitation and response must not result in a wrong action by the controller.

These assumptions provide the broadest, most realistic view of how a TMD might be required to interact with a large space structure. However, these assumptions severely limit the types of information available to the controller. In effect, the controller has virtually no a priori knowledge of the base structure or how it will be excited.

For the purposes of deriving a control the designer was allowed to assume the following:

1. The base mode is restricted to a known frequency range. That is, changes to the structure will not move the base mode outside of this pre-known range, but other than this, the initial base mode natural frequency is not known.
2. There will be only one mode in the restricted range.
3. The TMD may temporarily degrade the overall system. That is, there is no requirement that the TMD is never allowed to make the system response worse than if the TMD were not present.
4. The TMD is not required to tune in the absence of base excitation.
5. The connection point of the TMD to the base structure is capable of observing the base mode of interest. In other words, the TMD must be applied to the base structure at a location that exhibits relatively high displacements in the mode of interest.

The problem of deriving the controller can be stated as a sequence of questions.

1. What is the control algorithm?
2. What measurements are to be made that will allow a control to be synthesized?
3. How are the measurements to be processed to arrive at the control command?

4. What are the sensing, processing, and command requirements?
5. What is the performance of the control?
6. How might the control fail?
7. How robust is the control?
8. How is the control to be verified?

2.2.2 What is There to Work With?

The assumptions enumerated above limit us to working with response measurements and knowledge of how the underlying structure behaves. That is, we are allowed to assume that the base structure is a linear, elastic system.

Broad-band response is what distinguishes an undamped structure from one with an optimally tuned TMD attached to it. That is, the energy of structural response due to excitation is dissipated by the lossy TMD. For an unchanging excitation the base structure response is a minimum when the TMD is optimally tuned. But using the base response as a measure of tuning accuracy is unacceptable given the assumption that the excitation may also go to zero. Base response is, obviously, an absolute minimum when the excitation is zero (assuming the excitation is zero for a very long time). What is needed is a response measurement (or measurements), independent of excitation level and/or type, that yields a quantity related to that of an optimally tuned system.

Because of the simplicity of the structures and the mathematics that describe them, a high-performance control could be derived easily if a reliable estimator of the base mode could be found. The weakness of this approach lies in the estimation of the base mode. The assumptions of a time-varying base mode coupled with the assumption of time-varying excitation make the problem of deriving a base mode estimator very difficult. Many schemes that attempted to estimate the value of the natural frequency of the base structure were proposed. All of these proposals failed. The scheme that finally resulted arises from estimating only the current value of the base structure's natural frequency with respect to the current natural frequency of the TMD. That is, we do not estimate the optimal tuning parameters, but estimate the change in the current value of the tuning that will bring the system more closely to being tuned.

The physical system used to derive the controller is shown in Figure 4. An accelerometer is attached to the base structure at the point where the TMD is attached to the base structure. A second accelerometer is attached to the moving mass of the TMD. A thermocouple measures the temperature of the VEM in the TMD. A resistive, ribbon heater is embedded in the VEM. The box labeled "oven" refers collectively to the VEM, thermocouple, and the surrounding material and is

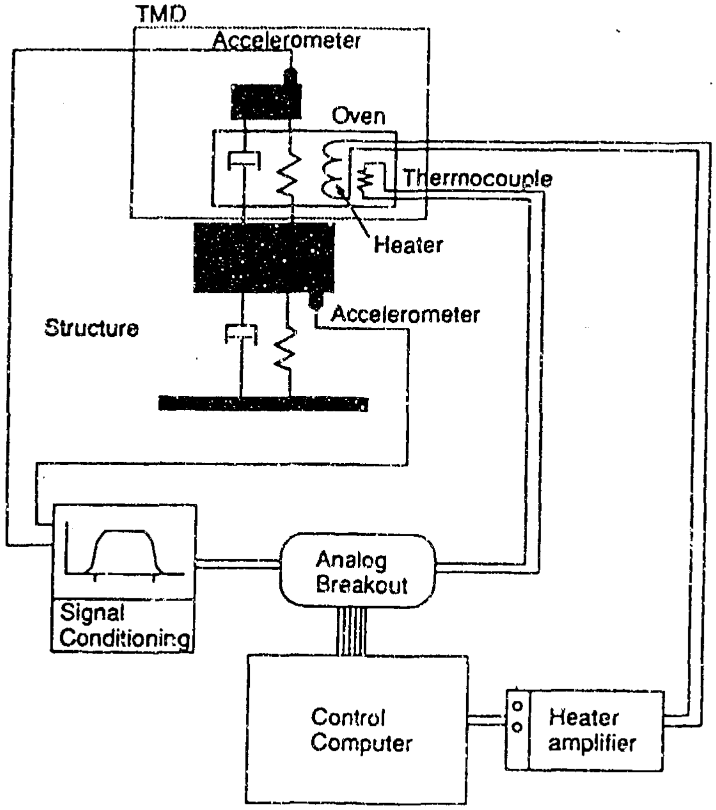


Figure 4. Schematic of the base structure, TMD, sensors, and controller

intended to convey the idea that the elastic and lossy components of the TMD are affected by the heater. This model contains all of the physical parameters that we are allowed (by the constraints enumerated above) to measure or control.

The tuning scheme arises from the curves shown in Figure 5. The curves labeled "Base" and "TMD" represent the RMS response to broad-band excitation of the base structure at the first accelerometer and the moving mass of the TMD when the VEM complex stiffness varies over the limits shown. (The loss factor of the VEM is assumed to be constant over this range.) This figure comes from solving the equations of motion for the system of Figure 4. The curve labeled "ratio" is simply the numerical ratio of the TMD RMS response to the Base RMS response. What is significant about this figure is that it shows that the maximum value of the RMS ratio occurs at the same TMD stiffness that produces the minimum base response. The minimum TMD response occurs at a significantly different stiffness.

This is believed to be a new result and represents the major step in constructing self-tuning TMDs. We now have a fundamental estimator for determining if a TMD is tuned. By exploiting the properties of this function we should be able to derive a controller that automatically and robustly tracks the underlying structure's dynamics and produces minimum response.

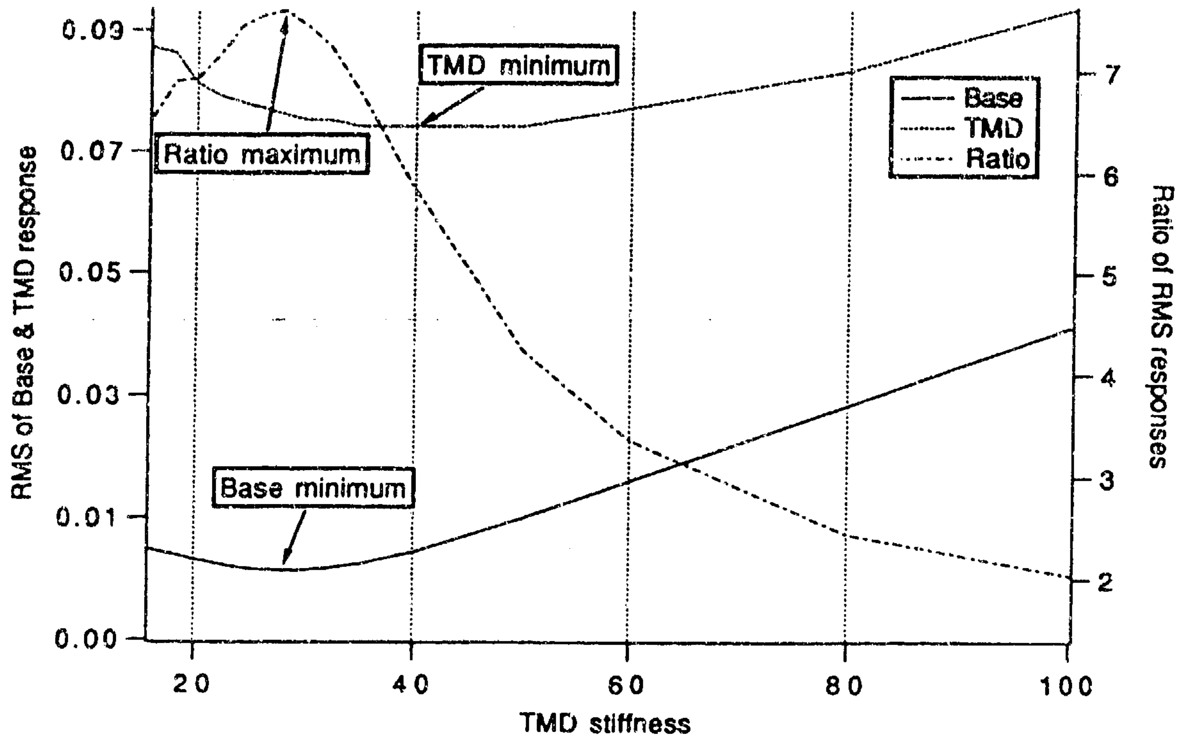


Figure 5. RMS of base structure, TMD, and their ratio.

2.2.3 The Control Algorithm

A PID (proportional-plus-integral-plus-derivative) type of control would be very desirable for many reasons. However, it is not appropriate because the set point (the maximum value of the RMS ratio) is unknown and varies with time. PID controllers actually minimize the difference between a set point value and the estimated current value of the process. In our case we are not trying to minimize a quantity, but maximize one. In addition, the estimation of the plant state may be interrupted due to the possibility that structural excitation may temporarily be zero. Another way to state this is that the process sensors may temporarily be disabled.

It needs to be kept in mind that the RMS ratio curve in Figure 5 can change with time. The change can be both in terms of where the maximum occurs (in terms of TMD stiffness and base structure) and the value of the ratio.

Because the underlying system can change continuously and abruptly, it is difficult to derive a controller that can tune the TMD by estimating the current state of the plant. Therefore, the approach we take is to estimate the "direction" that our control variable (temperature of the VEM in the TMD) must be adjusted. (Temperature is our sole control variable and we know the functional relation between VEM temperature and TMD state.) We explicitly assume that the TMD will never tune. That is, any estimate of the current plant state will lead to an update in the control command.

The RMS ratio curve in Figure 5 shows that a controller can be derived from the estimation of two RMS response quantities. These are the basis of our tuning algorithm. What is required is an RMS estimator for each sensor channel and some simple mathematics to compute their ratio and determine what the update to the VEM temperature is to be.

2.2.4 Implementing the Control

The control algorithm was implemented using a setup that essentially mimics the schematic of Figure 4. Charge-mode accelerometers were coupled to laboratory-grade charge amplifiers. These signals were fed through anti-aliasing filters to an analog breakout box. The breakout box was connected to a data acquisition card mounted in a Macintosh chassis. This data acquisition card has an eight-channel analog-to-digital (A/D) converter and two digital-to-analog (D/A) converters (Model MacADIOS II from GW Instruments). A digital-to-analog converter was used to convert the temperature control into a voltage. This voltage was input to a DC power amplifier which, in turn, was connected to the resistive heater embedded in the VEM of the TMD. The actual control loop was implemented using a software package called LabVIEW, from National Instrument Corporation. This software permits the building of block diagrams on the display using icons to represent operators and real hardware devices. Lines connecting the icons indicate passing of data. The software provides a wide variety of simulation, logical, signal-processing, and display/recording features. In addition, the user can construct their own virtual-instruments (as the icons are called) to implement a special operation.

LabVIEW breaks the simulation construction into two parts: the Front Panel and the Diagram. The Front Panel is for displaying simulation data (in digital readouts, meters, X-Y plots, strip charts, and lights) and creating controls such as On-Off switches and slide controls. The Diagram represents the instrument behind the Front Panel and consists of the various instrument building blocks and the "wires" (data paths) which connect them.

In our Diagram, the A/D converter is represented as an icon with selector data going into the icon (such as which channels to sample, number of samples, sample rate, etc.) and the sampled data comes out as "wires" containing the sampled data. The sampled data is then processed and results in a command to the power amplifier. This command is created by passing the numerical value of the command to the D/A icon, which formats and passes the command to the data acquisition card.

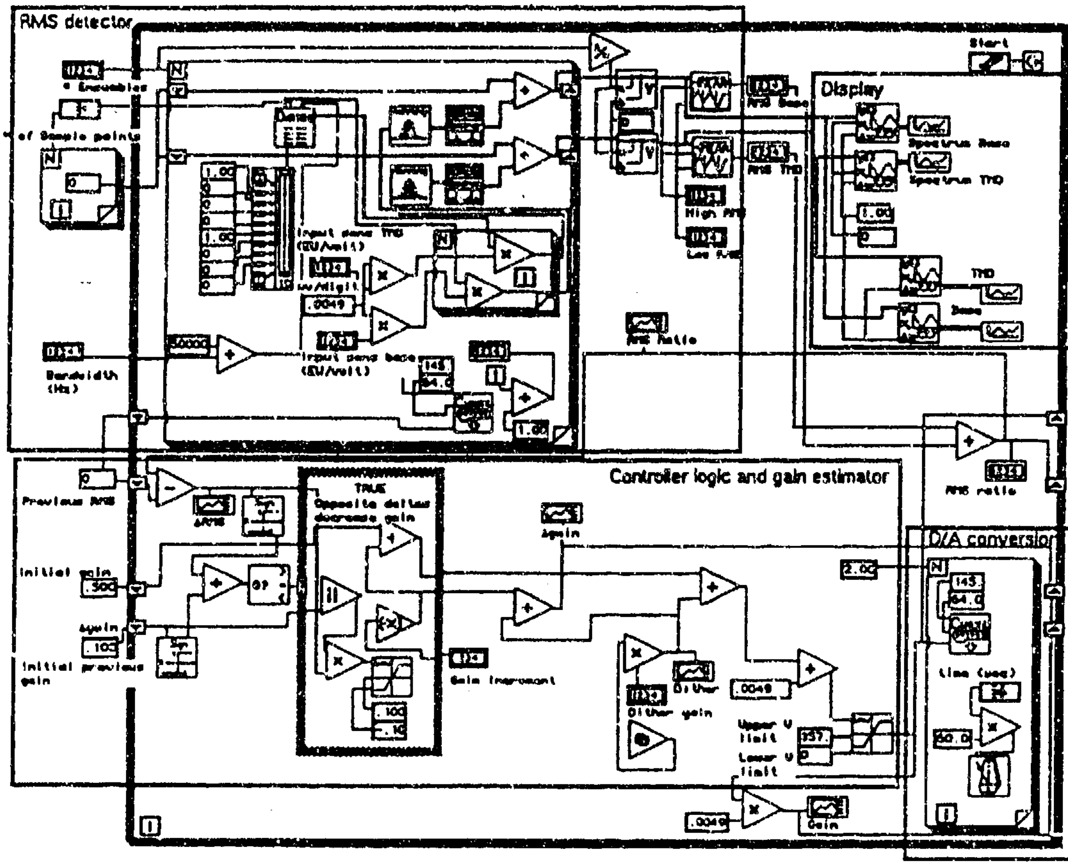


Figure 6. Control Simulation Diagram

With this system, a moderately complex simulation using real hardware can be implemented quickly, interactively, and inexpe.,sively.

2.2.5 The Control Diagram

The control simulation Diagram is shown in Figure 6. This implementation is block iterative, i.e., an estimate of the current RMS ratio is computed and passed on to the update algorithm, which in turn updates the VEM heater and displays various quantities to the experimenter. The entire sample-update loop depends on the sample rate, number of samples in the ensemble estimate of the RMS ratio, and the control update.

A key component in this implementation is the RMS detector (or estimator). The RMS values of each channel are obtained by integrating the RMS of the ensemble of FFTs of each sample. A number of ensembles are used to reduce the variance of the estimate (the degrees-of-freedom of the estimate are proportional to the number of spectral lines times the number of ensembles). The RMS function is integrated only over the spectral bandwidth where the base structure is confined to be.

The Diagram consists of four main components:

1. **RMS Detector.** The RMS detector samples the number of samples selected on the Front Panel for the two accelerometer channels. The DIGITIZE icon represents the data acquisition board. The sampled data are put into arrays (denoted by the thick wires) and passed through a block structure that converts them to scaled engineering units (a Front Panel box allows the calibration factor of the channel to be supplied), then into a HANNING weighting function, and then through a power SPECTRUM transform. The entire sampling block is inside an iterative structure which allows multiple ensembles to be used in estimating the power spectrum. Finally, the averaged power spectrum is converted to RMS units and integrated over the bandwidth where the base structure's natural frequency is assumed to be. The RMS quantity is then passed onto the controller logic block and the RMS spectrum is passed to the Display block.
2. **Controller Logic.** The controller estimates the updated temperature command using the current estimate of the RMS ratio, the previous RMS ratio, the previous temperature command (labeled Gain in the Diagram), and the change in Gain at the previous iteration. The controller uses these values along with the knowledge of the shape of the objective function (of Figure 5) to determine how much to increase or decrease the temperature gain value.
3. **D/A conversion.** The updated temperature gain is passed through a limiter (to avoid overheating the VEM) and put out through the Cwrite D/A converter. The loop then pauses for a specified time to allow the TMD temperature to equilibrate.
4. **Display.** The Display block constructs the graphical displays of the response spectra of the TMD and Base structure. These displays are made on the Front Panel.

2.2.6 Strengths of the Controller

The approach chosen does not require a calibrated TMD, i.e., the exact relation between TMD natural frequency and VEM temperature or stiffness and VEM temperature need not be known. What is known is that the natural frequency of the TMD at room temperature is higher than the upper limit of the allowed frequency range and that at maximum usable temperature the TMD's natural frequency is lower than the lower limit of the allowed frequency range. In other words, it is known that the TMD can change its natural frequency over the range of interest.

Furthermore, the TMD stiffness versus temperature and loss factor versus temperature relations are not required to be linear, but they are assumed to be continuous.

With the addition of the dither circuit (discussed next), the controller is persistent and will eventually produce optimal tuning assuming that the iterative loop time of the controller is faster than the rate at which the base structure changes.

2.2.7 Weaknesses of the Controller

A drawback to this type of proportional control is that it can get "stuck." In other words, the adaptive controller may conclude incorrectly that the system is optimally tuned when it is not. The reason for this is that the primary quantity used to estimate the direction to tune is based on the ratio of two quantities that might have a low statistical accuracy. Therefore, dividing two low-accuracy quantities can lead to a wrong control update. In some cases the ratio will, at random, have the same value as the ratio computed at the previous iteration. If this happens the controller will assume that the system is tuned because the RMS ratio is no longer changing with gain. This happens only at the peak of the tuning function where the derivative of RMS ratio with respect to TMD stiffness is zero.

The solution to this problem is to add what is known as a gain-dither circuit, usually referred to as just a dither. This circuit adds a random disturbance to the actual control (thermal control) so that the likelihood of estimating an unchanging RMS ratio is very small. Even if there is a single occurrence of an inaccurate estimate, the chances of two successive unchanging estimates is unlikely, and the chance of three successive unchanging estimates is highly unlikely.

The RMS detector is a relatively slow estimator due to the few DOF's available. Estimating the RMS value of a random quantity is, in statistical terms, a low-confidence process. We are looking for an accurate estimate of the two RMS quantities but to do so requires that we observe the response signal for a long period of time. The approach of estimating the RMS by integrating the magnitude of the FFT of the signal requires that we have a large number of discrete frequencies in the bandwidth and/or a large number of ensembles in the magnitude function. A large number of discrete frequencies requires a long observation window, and a large number of ensembles, obviously, requires a long time.

There is no temperature feedback in the current configuration. That is, is no explicit knowledge of the instantaneous temperature of the VEM. Thus, it is possible to overheat the VEM if the heater amplifier gain is too high. Temperature feedback would be a simple addition and would make using the setup in a development phase more robust. Ultimately, temperature feedback could be integrated into the controller to provide more information for tuning.

The minor shortcomings of the controller developed in Phase I will be corrected and improved upon in Phase II.

2.3 Design and Prediction Methods for Complex TMD's

The prediction by finite element techniques of damping levels in complex structures for constrained layer and discrete damping treatments has been very successful and is well documented.[1] [2] [3] These damping levels are found by using the Modal Strain Energy (MSE) method and the results from real-eigenvalue analyses. However, the MSE method is not applicable for TMD's, complex-eigenvalue solutions are needed to predict levels of damping.

The parameters that govern the tuning frequency and level of damping of a TMD are its "tuning" mass and the stiffness and loss factor of its complex spring. These parameters are normally expressed in the following manner:

1. Effective mass ratio (ψ), the ratio of the mass of the TMD to the modal mass of the target mode normalized to unity at the TMD's attachment point. (The modal mass will always be less than the physical mass, usually being on the order of 10 to 50 percent of the physical mass.)
2. Loss factor (η), the ratio of the imaginary part to the real part of the complex stiffness of the TMD spring.
3. Tuning ratio (α), the ratio of the natural frequency of the grounded TMD to the frequency of the target mode. This ratio and the mass of the TMD determine the required spring stiffness for the TMD.

Even though TMD design cannot be accomplished with the modal strain energy method, semi-analytical guidelines exist for designing a simple TMD for a structure that resembles a single-degree-of-freedom system in the neighborhood of a mode of interest.[4] [5] These guidelines are a set of formulas that dictate design parameters for a chosen application point and a desired level of damping. For a given effective mass ratio (ψ) selected by the designer, the TMD loss factor (η) and tuning ratio (α) are given by

[1] C. D. Johnson, and D. A. Kienholz, "Finite Element Prediction of Damping in Structures with Constrained Viscoelastic Layers," *AIAA Journal*, Vol. 20, No. 9, September 1982.

[2] C. D. Johnson, and D. A. Kienholz, "Prediction of Damping in Structures with Viscoelastic Materials," *MSC/NASTRAN User's Conference Proceedings*, March 1983.

[3] C. D. Johnson, D. A. Kienholz, E. M. Austin, and M. E. Schneider, "Design and Analysis of Damped Structures Using Finite Element Techniques," presented at the 1985 ASME Conference on Mechanical Vibration and Noise, September 1985, Cincinnati, OH (Paper 85-DET-131).

[4] A. D. Nashif, D. I. G. Jones, and J. P. Henderson, "Vibration Damping," Wiley and Sons, 1985

[5] "Shock and Vibration Handbook," 2nd Edition, McGraw-Hill, 1976

$$\eta = 2\sqrt{\frac{3\psi}{8(1+\psi)^3}} \tag{1}$$

$$\alpha = [1+\psi]^{-1/2}[1+\eta^2]^{-1/4} \tag{2}$$

Investigations into the reliability of these design guidelines were performed using finite element analysis and MSC/NASTRAN. Iterative analyses were executed on models of a variety of structures. First, a simple TMD located at the center of a pinned-end beam was analyzed. This is a very simple structure with widely spaced modes. Normal modes analysis was used to determine the undamped modes, and complex eigenanalysis was performed to determine the amount of damping produced by the TMD. It was found that the guidelines gave useful starting values for TMD loss factor and tuning ratio, but these values needed to be changed to obtain optimum damping. For a given effective mass ratio, a higher TMD loss factor could be used and more damping could be obtained than predicted by Equation (1). (As long as a TMD can be tuned, i.e., equal damping in the split modes, a higher TMD loss factor will produce a higher level of damping.) A similar analysis was then performed on a curved panel with stiffeners and pinned boundary conditions, simulating a section of an aircraft skin. Again, a higher TMD loss factor than was predicted by the design formula was used to obtain a higher level of damping. This analysis cycle was repeated on complex finite element models, including an aircraft vertical stabilizer and the focal-plane plate of a satellite telescope, and similar results were obtained.

The results of these analyses indicated that traditional design guidelines for the loss factor and tuning ratio of TMD's are conservative and give values for design parameters that will produce buildable TMD's, but ones that produce less than optimal damping. Further study would be required to develop formulas that give these parameters exactly, but it can be stated that the existing formula for TMD loss factor will give only a starting value, and higher loss factors are possible that will produce higher levels of damping. If too high a loss factor is used, however, the TMD will not be "tuneable."

An important observation obtained from these analyses is a relationship between amount of damping (modal loss) and effective mass ratio. This relationship enables a TMD designer to select the weight of a TMD based on the amount of damping desired. Using the results from the above analyses, modal loss factor was plotted versus effective mass ratio. The plot shown in Figure 7 illustrates the amount of damping at the system level, i.e., the modal loss factor, that is predicted by the analyses as a function of effective mass ratio.

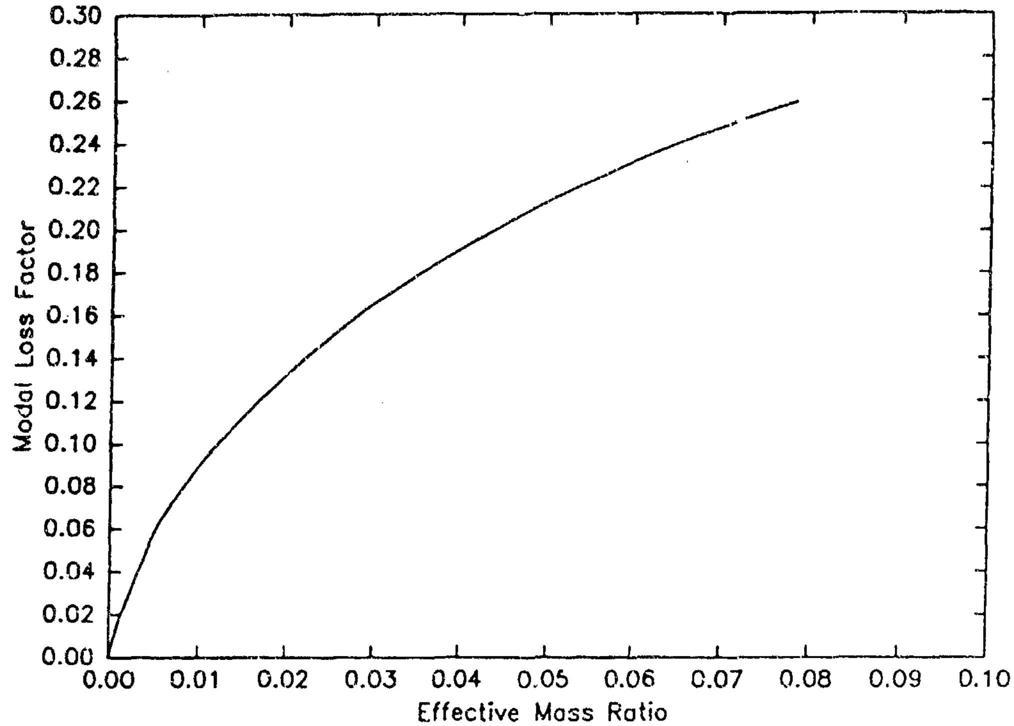


Figure 7. System modal loss factor versus effective mass ratio

2.4 Thermal Regulation

It was proposed to regulate the temperature of the viscoelastic materials by embedding a heater in the VEM. However, a number of issues needed to be addressed before the demonstration could be conducted.

The heating device has to be relatively efficient, have enough heating capacity to raise the temperature of the VEM quickly, and be compatible with ordinary laboratory instrumentation. Various types of heaters were considered. The most suitable for this application were thin-film strip heaters. They convert electrical resistance into heat via flat foil elements and are fabricated using techniques adapted from printed circuit boards. These come in a wide variety of sizes and power ratings and are thin enough to be embedded in the VEM without significantly altering the VEM's effectiveness. This type of heater transfers heat more efficiently and over a wider surface area than a wire-wound heater, and it can safely operate at wattages twice as high as the wire-wound equivalent.

With the type of heater used and its efficiency, it was determined that multi-layers of different viscoelastic materials would not improve the performance of the prototype smart TMD. Multi-layers of different viscoelastic materials may be useful to obtain optimal performance in certain design situations. However, it will be problem related and will be studied during Phase II when a number of different types of smart TMD's are built.

The temperature of the VEM was not critical to the operation of the TMD controller, since the heater was turned on and off based on the processing of accelerometer output by the controller. Nevertheless, knowledge of the VEM temperature at any given time was useful for the debugging of the control system and for observing the effects of temperature variations on the VEM. For this purpose, a resistance temperature detector (RTD) was installed in the TMD adjacent to the VEM. An RTD is a temperature sensing device based on resistance changes in a metallic element such as platinum. The main advantages of an RTD over a thermocouple or thermistor are its repeatability and stability (industrial models typically drift less than 0.1°C per year). Also, platinum-element RTD's follow a more linear curve than thermocouples or most thermistors, and they are effective over a wider temperature range.

3. Concept Demonstration

3.1 Design of the Test Article

To demonstrate the smart TMD, an undamped structural test article was required on which the smart TMD could be tested. The test article was chosen to be a long beam with clamped end conditions so that the modes would be fairly low in frequency and widely spaced. An 8-foot-long steel square tube with a constant cross-section measuring 2 by 2 inches and a wall thickness of 0.060 inches was used. To provide the boundary conditions and clearance for a shaker, the beam was bolted at each end to 10-inch-high steel right-angle fixtures that were clamped to a heavy workplate. The TMD was bolted to the beam at the point of maximum deflection for the first bending mode, i.e., at the midpoint. The test setup is shown in Figure 8.

The target mode of the steel square beam is around 30 Hz. Due to the factors cited in Section 2.1, the TMD chosen for this application is a cantilevered sandwich beam. A schematic of the beam TMD used is shown in Figure 9. The beam was designed so that at room temperature before heat was applied to the VEM its natural frequency was around 40 Hz. The design was implemented so that heating of the beam to over about 38°C (100°F) would be required to lower the frequency to that of the base structure. This design was arrived at through finite element analysis using MSC/NASTRAN. The final TMD design that met these specifications was 10 inches long by 1 inch wide. The facesheets were made of 0.090-inch-thick graphite-epoxy (E $\simeq$ 8.E6 psi) and the VEM was 0.040-inch-thick Soundcoat DYAD 606. The actual layup of the VEM was two 0.020-inch-thick layers between which the heating element was sandwiched. The "tuning" of the TMD is accomplished because of the variation of the shear modulus of the DYAD 606 with temperature. This relationship is plotted in Figure 10.

The heater used was a MINCO Thermofoil (TM) 9.4-ohm heater with a thickness of 0.0075 inches, bonded between two layers of VEM. The device was relatively efficient, and had enough heating capacity to raise the temperature of the VEM from room temperature to approximately 55°C (130°F). It was also compatible with ordinary laboratory instrumentation.

The RTD used was a MINCO Thermal-Ribbon (TM) Resistance Thermometer, 0.020 inches thick and measuring 0.3 inches by 0.3 inches. This RTD has a base resistance of 100 ohms and a temperature coefficient of resistance of 0.00385 ohms/ohms/°C. It has a temperature range of -40°C (-40°F) to 200°C (392°F) and a time constant of 0.15 seconds.

Preliminary tests were performed on the prototype without any control system, but with heaters in the VEM. The temperature of the viscoelastic material in the dampers was varied manually to check the performance of the thin-film heater. This

Figure 8. Photo of test configuration

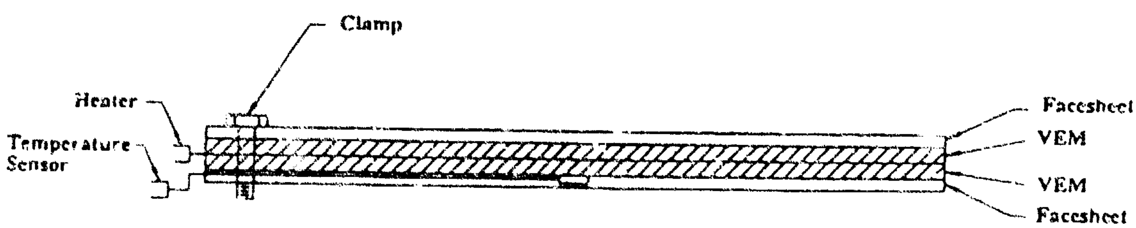


Figure 9. Cantilever beam TMD

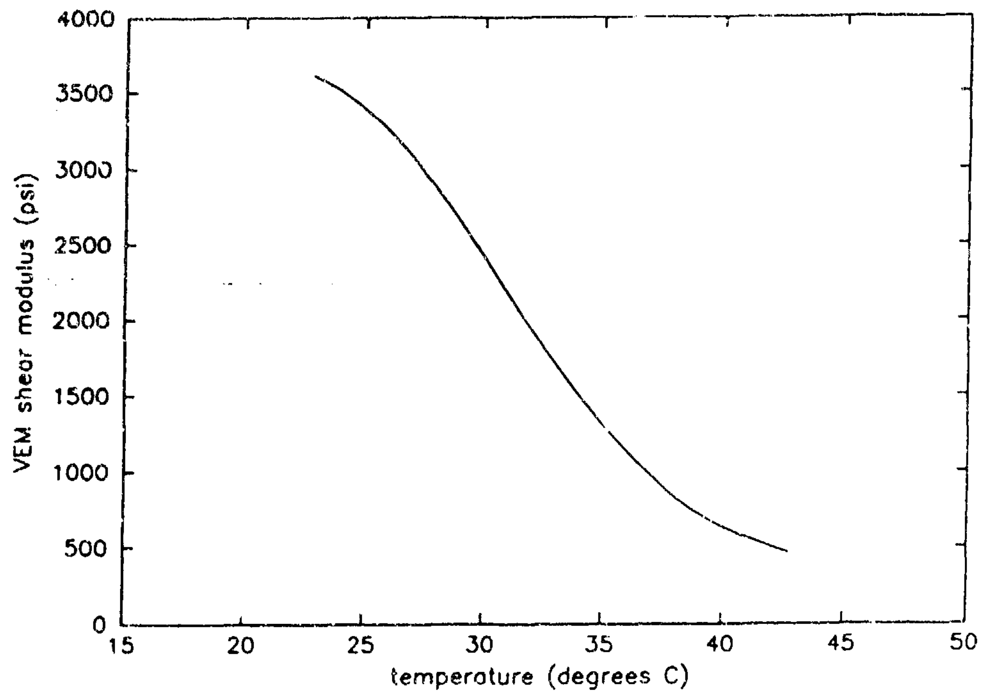


Figure 10. Variation of shear modulus with temperature for DYAD 606 around 35 Hz

open-loop check also verified that the frequency of the tuned beam was indeed in the range of the controller.

3.2 Experimental Verification of the Control

Experimental verification of any concept is ideally done with as few simulated items as possible. For this application it means that a real, elastic structure has a real tunable VEM attached to it. The control should be implemented with the proper sensors and conditioning and the controller should run at the proper rate (as decided by theory or experiment). For our experimental validation we have been able to provide all of these conditions.

Various configurations were used in setting up and debugging the experiment. Some configurations simply checked out the calibration of the A/D instrumentation channels or the TMD heater. Two experiments that show the controller under various assumptions and how it performed are documented here. These two experiments capture all of the main features of the self-tuning TMD. Obviously, not all of the possible excitation and structural configurations were used, but these two do demonstrate successfully the features we are seeking.

3.2.1 Experiment 1 — Self-Tune

This experiment is intended to show solely that the controller can, under a certain configuration, optimally tune the TMD to produce a minimum structural response.

3.2.1.1 Configuration

The experiment consists of the beam structure, the thermally controlled TMD, the two accelerometer channels, the control computer, and the excitation provided by an electrodynamic shaker. The shaker is attached to the beam (as shown in Figure 8) and a white-noise signal is applied to the shaker. The shaker produces enough response in the accelerometers to give a good signal-to-noise ratio. The level of the force is unchanging throughout the experiment. In short, this demonstrates that the controller can tune the TMD, from start-up, when the structure and its excitation are constant.

3.2.1.2 Explanation of Results

Figure 11 shows a "snapshot" of the controller Front Panel at the first iteration of the control. It shows two distinct modes of the structure and TMD as expected. There is, as yet, no current going to the heater; the ambient temperature is about 73°F (not shown on the Front Panel).

The Front Panel shows graphical displays of the RMS ratio quantities in two formats: the two graphs on the right are in logarithmic mantissa units and the graphs on the left are in linear units. The channel calibration units are shown left of the logarithmic displays. The controls at the bottom affect sample rate, number of samples, integration range, and equilibration time. The displays at the bottom are strip chart records of various measurement quantities over time.

Figure 12 is a snapshot taken at the end of the experiment. A comparison of Figures 11 and 12 show that the undamped response of the beam is reduced by a factor of four (note change in scale).

There are a number of interesting features shown in the individual displays of the panel. The data in the display were recorded, reformatted, and displayed in Figure 13 for more ready explanation.

Each view of Figure 13 is plotted versus time. Here, time is actually iteration steps for the controller.

1. Figure 13a is the estimate of the objective function: the RMS ratio.
2. Figure 13b is the change in the RMS ratio at each iteration. The large jump at Time 01 is due to a start up transient in the estimator.

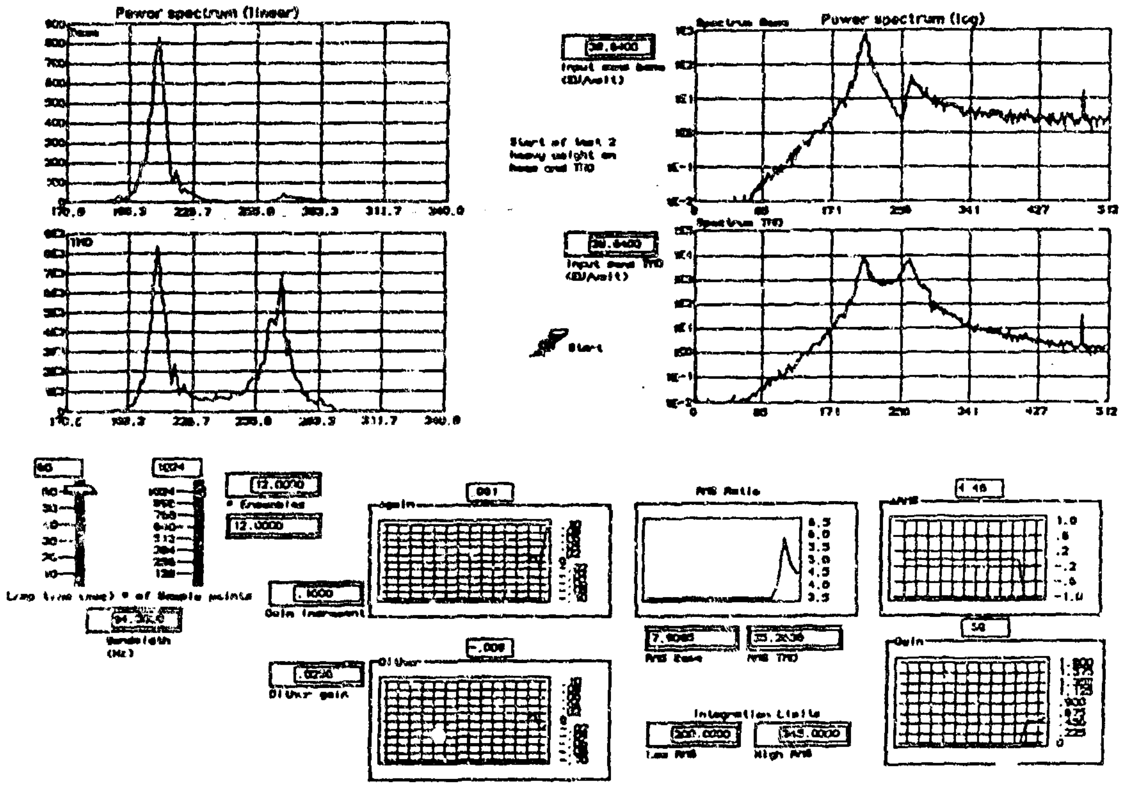


Figure 11. Front Panel at first iteration of controller

3. Figure 13c is the D/A gain (in Volts) to the TMD heater amplifier.
4. Figure 13d is the change in heater gain from iteration to iteration.
5. Figure 13e is the temperature of the VEM.
6. Figure 13f is the power dissipated in the TMD by the heater. (Note that this configuration had no thermal insulation. This was done in the interest of speeding up the response time of the TMD to cut down on experiment run times.)

Figure 13a shows most of the features of interest from this experiment. After the start up transient (Time 02) the controller immediately gets "stuck." The reason is that the TMD was set up such that it was too stiff for the range over which it was to work. That is, the overlap of the TMD and base structure modes was too small for the controller to affect the RMS ratio by shifting the TMD's natural frequency. The TMD should have been slightly better tuned to start with. Nonetheless, the figure shows that at Time 10, the dither circuit was able to "unstick" the control and start the control in the right direction. At Time 17 the RMS ratio becomes a maximum and the control correctly locks onto the gain level and maintains tune.

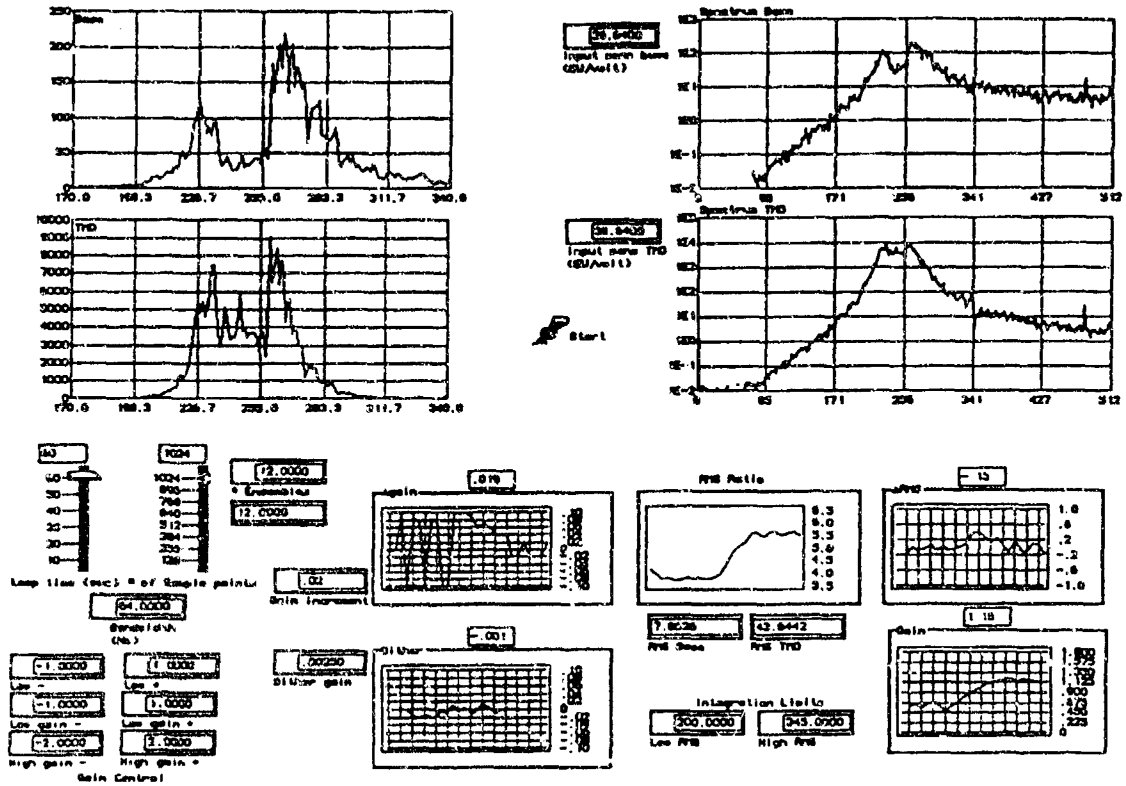


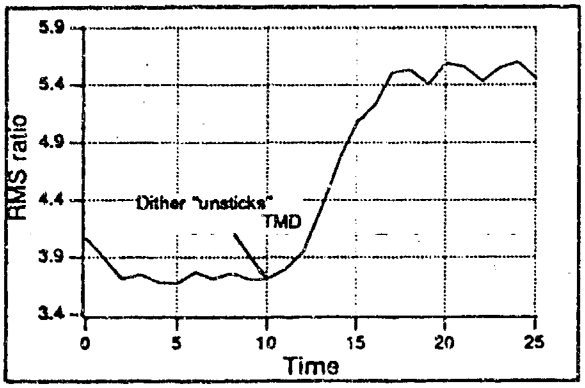
Figure 12. Front Panel at the end of a tuning exercise

The RMS ratio that the control estimated to be the maximum is, in fact, the maximum. This was verified independently by running the heating open-loop and manually finding the maximum. Figure 12 qualitatively shows that the system is properly tuned.

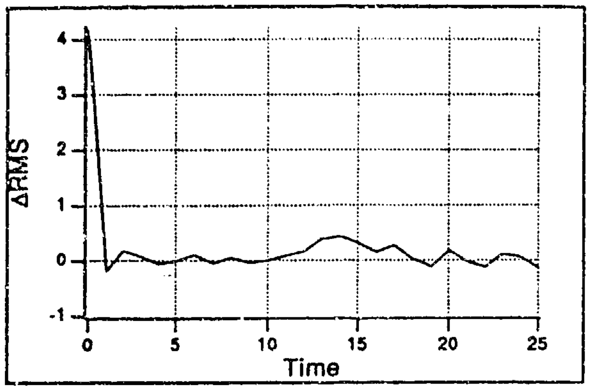
3.2.2 Experiment 2 — Changing Structure

The second experiment is similar to Experiment 1, but now we test the ability of the controller to re-tune when the base structure changes. The goal of the controller (identical to the Experiment 1 configuration) is to first tune the TMD from start-up and then re-tune after a change has been made to the structure. The change will be abrupt as opposed to gradual. (Abrupt means the change occurs in less than one control iteration and gradual means the change occurs over several iterations.)

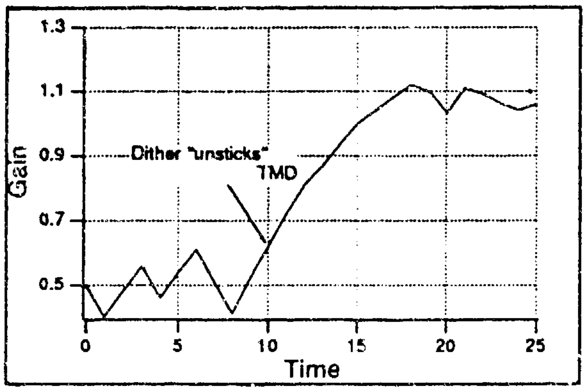
This experiment is not a great deal different than Experiment 1. The primary difference is whether the controller can tune when removing the assumption that the natural frequency of the base structure is less than the TMD's (the start-up assumption of Experiment 1).



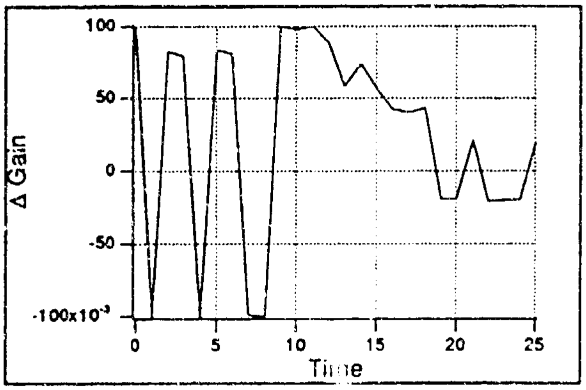
a.



b.



c.



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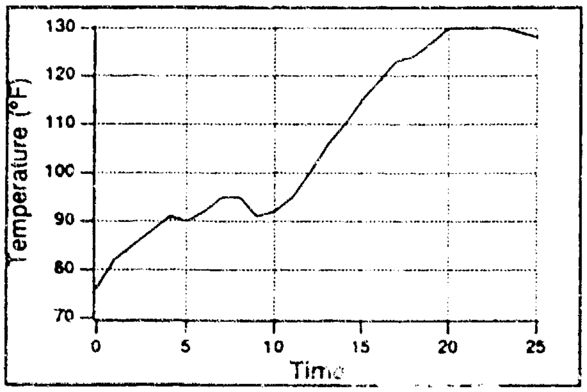
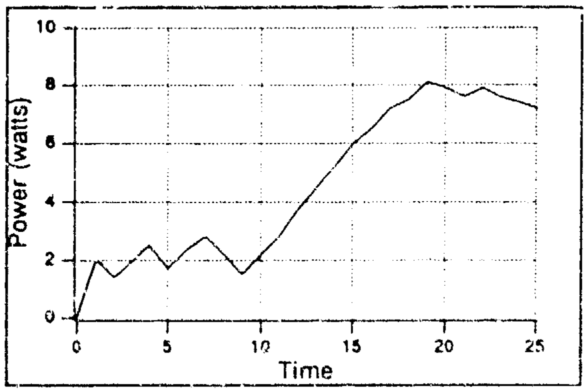


e.



f.

Figure 13. Detail of Front Panel at the end of a tuning exercise

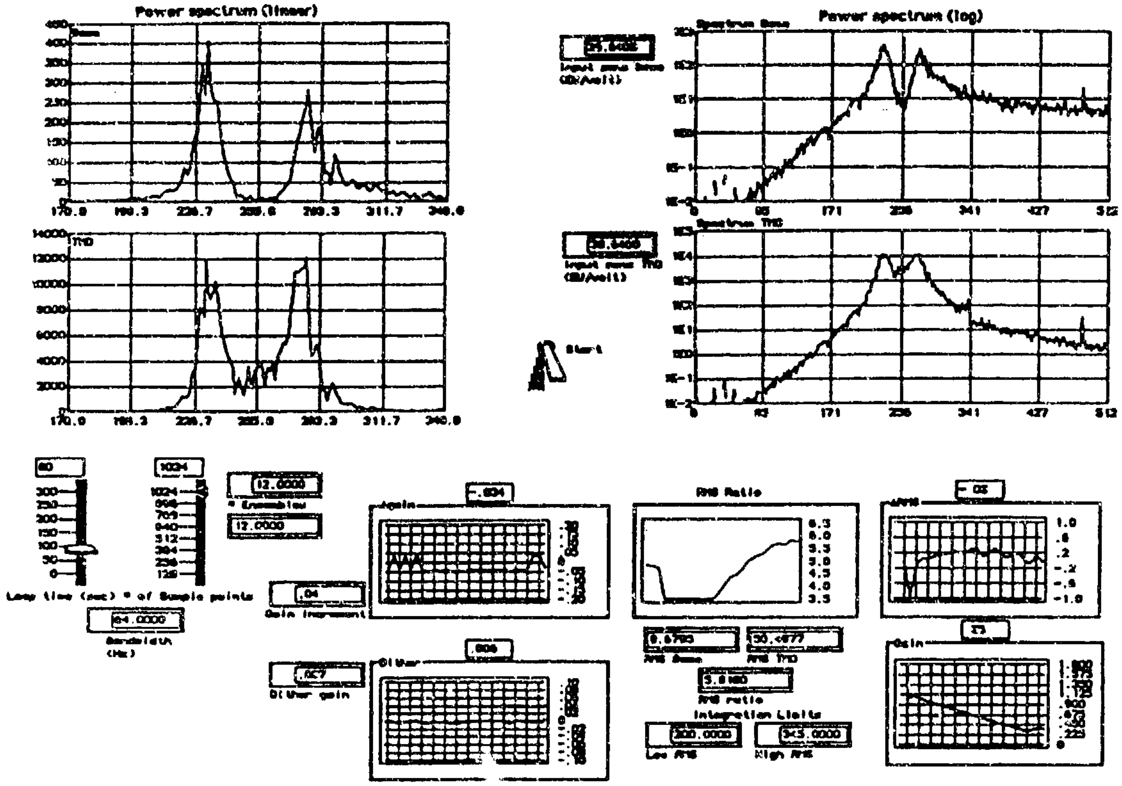


Figure 14. Front Panel at the end of re-tuning for the changing structure

3.2.2.1 Configuration

The experiment configuration starts out exactly the same as in Experiment 1, except that a weight was attached at about one-third span to lower the beam's first natural frequency. After the control had successfully tuned the TMD, the structure was changed by removing the weight. Thus, the natural frequency changes abruptly but remains in the pre-ascribed range.

3.2.2.2 Explanation of Results

Figure 14 shows a "snapshot" of the controller Front Panel at the final iteration of the control. It shows that the TMD is currently tuned. The data in the display of Figure 14 were recorded, reformatted, and displayed in Figure 15 for more ready explanation. Each graph in Figure 15 corresponds to the same graph as in Figure 13.

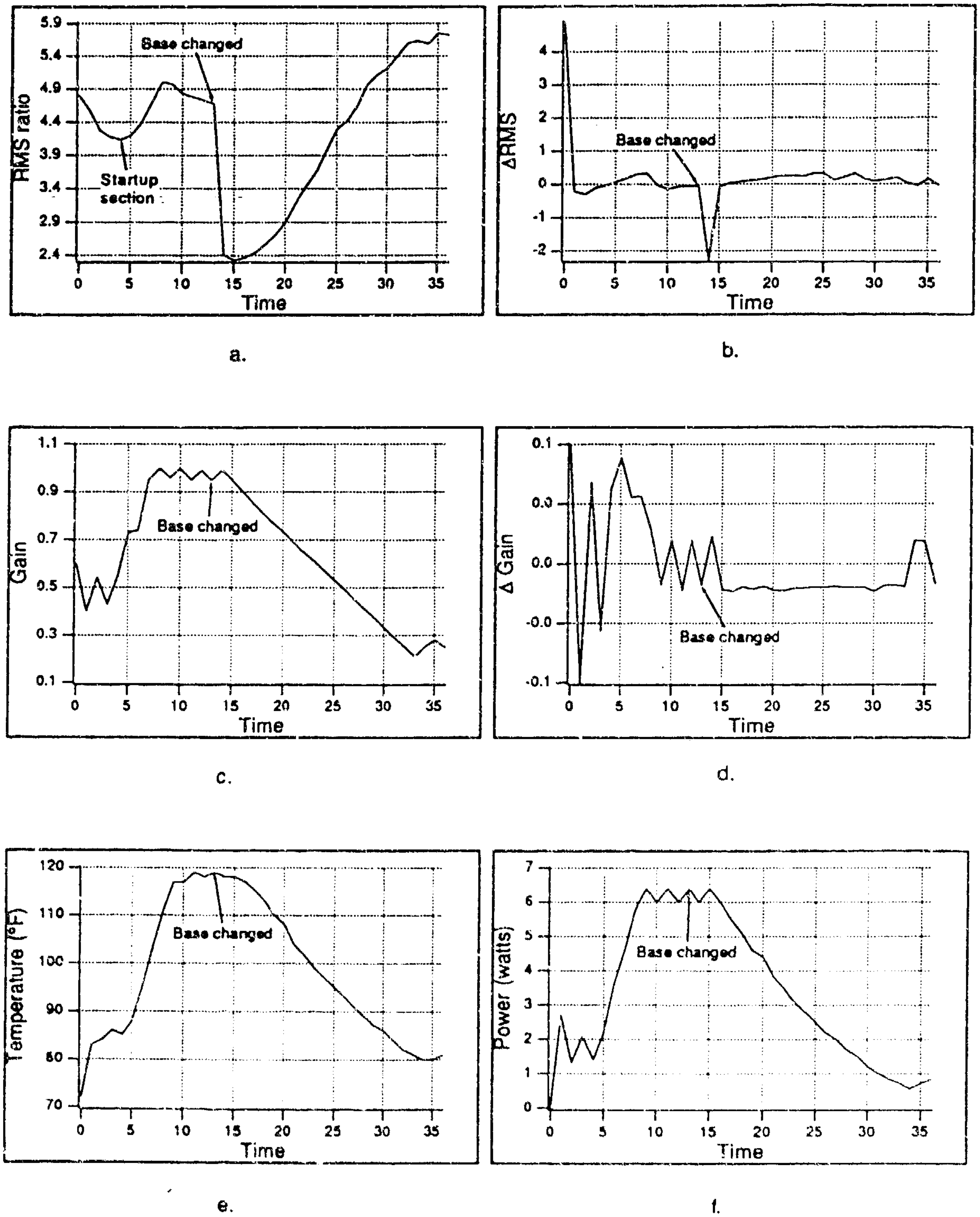


Figure 15. Detail of front panel at the end of re-tuning for the changing structure

Figure 15a shows that after the startup transient (after Time 04) the controller correctly tuned the TMD (Time 08). The base structure was then abruptly changed by removing the weight. The current gain setting was now incorrect and the controller correctly found the tuning direction and proceeded to retune the TMD (Time 32).

The long tuning time is due to the gain circuit having a limiter which kept the gain from changing too quickly. In this experiment the limiter was probably kept too tight. A broader limit range would have allowed the controller to retune faster after the structural change.

4. Conclusions and Phase II Recommendations

Conclusions

The study was successful in demonstrating all of the essential aspects of the self-tuning TMD. The controller can find and track a single structural mode and produce the maximum system damping for that particular TMD. The use of the RMS ratio as the basis for the tuning control has been shown to be essentially correct and accurate for optimum performance. The controller, through its dither circuit, is capable of tuning even when the base structure or TMD fall slightly outside the assumed tuning range or when the quality of the RMS detection is degraded.

Thin-film heaters were shown to be adequate for regulating the temperature of the VEM as dictated by the control system. This type of heater was ideal for the tuned beam used in this experiment since the VEM could be embedded in the VEM, providing uniform temperature regulation. In addition, thin-film heaters are relatively efficient in converting voltage into heat, and they can be custom-made to match any shape VEM surface.

Phase II Recommendations

There are two major goals for SBIR's. The first and primary goal is to develop a technique, product, device, etc., which benefits the customer (SDIO and the Air Force in this case). The other goal of SBIR's is for the small business to develop a product or service which will be pursued in an unfunded Phase III and can be used to develop a new business base. CSA believes that this SBIR satisfies both goals.

Phase I has laid the groundwork for a successful Phase II. All of the original goals and tasks of Phase I have been completed successfully, and Phase I has demonstrated the mechanics and control logic necessary for a self-tuning TMD. The concept demonstration was done in a laboratory setting, and much of the instrumentation and signal processing is tied into powerful modal analysis hardware. The transition between laboratory and industry is the goal for Phase II.

The recommended course for Phase II work is to develop fully the potential of self-contained "smart" TMD's. The envisioned final product would be a self-contained unit that could be easily attached to a variety of components used in space structures. The primary goals are as follows:

1. condense the electronics down to a few custom circuit boards
2. improve the speed of the controller
3. investigate power collection and storage (i.e., self-contained power system)

5. Potential Post Applications

CSA's experience has shown that there is a great need for passive damping and the accurate prediction of such damping for a large variety of structures. It has been shown by many investigators (both DoD and NASA) that passive damping will be required for the success of many space missions, especially those performing precision pointing. Though the demonstration "smart" TMD was a tuned beam, it is likely that the production version will appear more like the sketch in Figure 2. This configuration will allow the device to be used for a variety of applications.

At present, passive damping treatments using viscoelastic materials cannot perform over a wide temperature range. With the inherent thermal control built into this TMD, the possible applications will be numerous. CSA feels that "smart" TMD will find uses in many space-based and Earth-bound structures where discrete, troublesome modes hinder performance. The strongest selling point will be that the TMD will not need the constant attention of a trained test engineer to maintain its proper tuning. We feel that this will have wide market appeal, whether the structure is space bound or in the Earth environment.

Although it is well known that TMD's are very weight-effective for producing high damping for single modes, they are not used to the extent that they could be used because they become un-tuned as the environmental temperature changes or as the viscoelastic material properties change slightly. The smart TMD will eliminate these drawbacks. Therefore, the appeal of such a device may lead to quantity production, which will in turn lead to the cost of the smart TMD to be not a great deal more than a conventional TMD. The power source for Earth-bound TMD's may also come from ordinary sources, which will save costs.

Potential applications not only include space structures, but also aircraft, TV and power transmission towers, rotating machinery, ground transportation vehicles, helicopters, truss structures, buildings for wind excitation and earth-quake control, optical telescopes, and on and on.

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