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"ACTIVE CONTROL OF FOUNDATION TRANSMISSIBILITY USING MAGNETOSTRICTIVE ACTUATORS"

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"ACTIVE CONTROL OF FOUNDATION TRANSMISSIBILITY USING MAGNETOSTRICTIVE ACTUATORS"

A Trident Scholar Project Report

by

Midshipman Kenneth E. Poorman, Class of 1992

U. S. Naval Academy

Annapolis, Maryland

Robert S. Reed

Adviser:

Associate Professor Robert S. Reed

Department of Weapons and Systems Engineering

Accepted for Trident Scholar Committee

Francis D Correll

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The purpose of this project was to determine if vibrations could be actively canceled using a magnetostrictive actuator, a device which applies force when exposed to a magnetic field. The periodic vibrations associated with rotational equipment such as motor-generators were the primary focus of this research. Using acceleration feed-back of the foundation only, the magnetostrictive actuators were driven by a filtered signal while being biased with a DC current. The magnitude of vibrations decreased by almost twenty decibels at the resonant frequency. Frequency feedback from the tachometer was added to control an "activation"threshold. As a result of this researc[h] it is predicted that a digital controller would be better suited to active control of rotationally induce[d] vibrations.

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ABSTRACT

The purpose of this research project was to determine if vibrations could be actively canceled using a magnetostrictive actuator, a device which applies a force when exposed to a magnetic field. The periodic vibrations associated with rotational equipment such as motor-generators were the primary focus of this research.

Using acceleration feedback of the foundation only, the magnetostrictive actuators were driven by a filtered signal while being biased with a DC current (to produce a constant magnetic field). The result of this control effort was a decrease in the magnitude of vibrations by almost twenty decibels at the resonant frequency; the effect was less dramatic at other frequencies. For this reason, frequency feedback from the tachometer was added to control an "activation" threshold.

As a result of this research, it is predicted that a digital controller would be better suited to active control of rotationally induced vibrations. A digital controller would be able to incorporate past performance of the system with present feedback data and use that information to better predict what modified signal should be sent to the actuators. Initial experiments with open loop control, where the actuators were being driven by an independent oscilloscope, fully supported this prediction.

PREFACE

First and foremost, I must thank my adviser, Associate Professor Robert S. Reed, for all of his help and inspiration. There were several times during the past year that I had gone astray, and it was he who set me back on track. He saved me many hours (maybe days?) of useless adventure.

I would also like to thank the staff of the Department of Weapons and Systems Engineering at the Academy for their help in all of the finer points of my project: actually building the model from scratch or help with the technical aspects of hardwiring the controller circuits. Their dedication and knowledge should be at all researchers' disposal!

No less important, I would like to thank my mother and step-father for their understanding of my infrequent visits to see them in Washington, D.C. while the deadline for my project rapidly approached. Many other friends were visited less frequently too, such as my initial midshipman sponsor, CDR William Davidson and his family, and my newest sponsors, the Nitsches. I did not mean to neglect any of you, and I will see you all plenty over the summer while I am TAD at the Naval Academy. (I will even take you sailing to help make up for the lost time.)

Finally, I would like to dedicate this work to the students of Jones Elementary School in Severna Park, MD. While a first class midshipman, I helped in the Mids-N-Kids program there and was often inspired by their overwhelming drive to learn. It kept me on track and kept me motivated even during the slower times. I am sure that many of the students I worked with will become engineers or scientists someday and I hope that, in some way, I helped make that decision easier for them.

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PROBLEM STATEMENT

Rotational equipment, because of its inherent eccentricities, is often a source of periodic mechanical vibrations. These vibrations are commonly transmitted through the foundation of the equipment to surrounding equipment (such as a submarines hull) and can lead to troublesome noise or elevated material stress levels. Industry has traditionally reduced the level of noise transmitted through foundations by employing passive vibration controllers such as shock mounts, but it has also surrendered the rigidity of the equipment. Active control would have the possibility of being much more effective than passive control and can maintain the integrity of the system if proper controllers were developed.

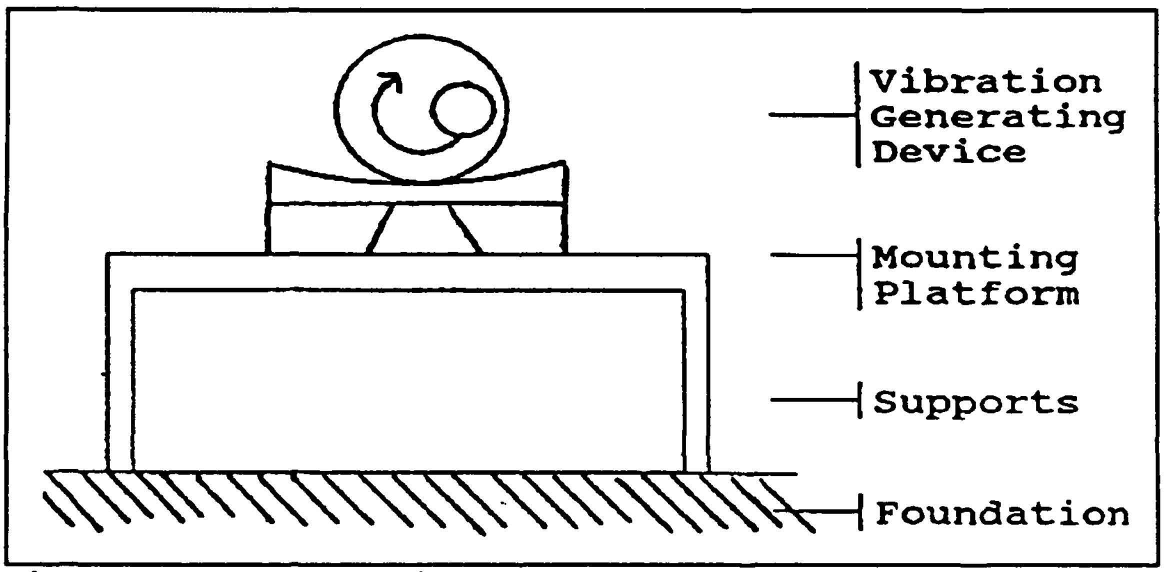

Figure 1 - The physical model of the components of the original vibrating system

The object of this research project was to develop a controller with magnetostrictive force actuators that actively cancels or modifies the vibrations transmitted by the foundations of rotating equipment.

BACKGROUND

Interest in active vibration control has been growing quickly in the engineering community, especially because of its applications in system isolation and reduced material stress levels. Rotating equipment, such as motors and generators, always have some degree of eccentricity and therefore a mechanical vibration associated with them. The United States Navy has a vested interest in this field of research and has been developing techniques to isolate mechanical vibrations from their surroundings for many years, especially in the interest of submarine quieting. Most developments which have been successfully implemented, however, have been of a passive nature. Other applications for active vibration control would include sensor isolation, such as that found on helicopters, where careful, noise free measurements must be made, or even luxury cars, where passenger comfort depends on elimination of vibrations and road noise before it reaches the passenger compartment.

A very significant development in the area of active vibration control was Terfenol-D, a highly magnetostrictive rare-earth alloy developed at the Naval Ordnance Laboratory by Arthur E. Clark [1]. As the dynamic element in a force actuator, it is capable of a large force output and a fast frequency response. The force developed is an intrinsic quality of Terfenol-D's magnetostrictive properties, i.e., its tendency to expand when exposed to a magnetic field. Although the physical dimensions change only slightly if free to do so, tremendous forces can be produced if it is restrained.

A magnetostrictive force actuator is a device which has as its main element a magnetostrictive rod such as Terfenol-D. The rod passes through the center of a coil of wire which is connected to a power amplifier providing the current necessary to generate and control the ambient magnetic field of the magnetostrictive rod and therefore its effective force output.

With the recent introduction of Terfenol-D, interest in active vibration control has been rejuvenated. This particular research project was inspired by independent research done at the United States Naval Academy by Associate Professor Robert S. Reed of the Weapons and Systems Engineering Department [2, 3]. Professor Reed investigated the possibility of using magnetostrictive actuators as a shock isolation mechanism and determined that further research should be done on the use of magnetostrictive actuators in vibration control.

METHOD AND MATERIALS

The method of active vibration control used in this research project depends on feedback from sensors to drive a second, "counter-vibration" source. The physical explanation of this control method is given by the principal of superposition, which, as it applies to this problem, states that two vibrations (manifested as waves) propagating through a common medium will be added together where they meet [4]. In theory, if the two vibrations are "opposites," they cancel themselves and if they are the same, they double (where an infinite number of possibilities also exists between these two extremes). If such "opposite" vibrations could be collided intentionally, all vibrations in the foundation would be canceled, thus preventing all vibrations from reaching the surrounding environment. The three major components of this type of active controller are:

1) the actuators,
2) the master controller, and
3) the feedback sensors.

The first of these components, the actuators, have the only moving parts in the controller and therefore limit the controller's response time. The actuator assembly consists of four sub-parts, including a Terfenol-D rod, a wire-wound spool, two o-ring spacers,and a preloading mechanism (See the Construction Details in Appendix 1).

The magnetostrictive element in each actuator is

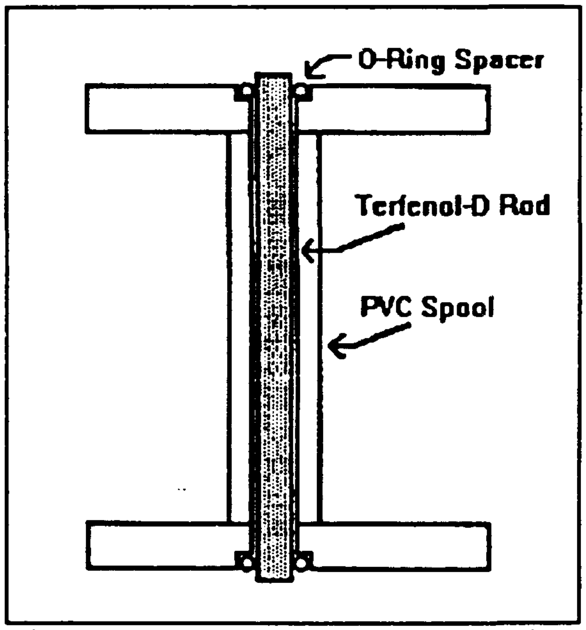

Figure 2 - The spool housing for the wire coil has a space for the Terfenol-D rod to pass through, using o-rings to maintain a small gap.

Terfenol-D, a rare-earth alloy which, in short, is highly sensitive to magnetic fields and develops a force proportional to it. (Details of the material properties of Terfenol-D will be covered later in this section.) For this project, each element was shaped into a rod 3 inches long with a diameter of 0.4 inches. The rod passed through the axis of spool of wire (the inner diameter slightly larger than 0.4-inches) and was held away from the insides of the spool by the two o-ring spacers, as demonstrated in Figure 2. A current passing through the wire wound around the spool created the necessary magnetic field, on the order of 1000 Oersted, for optimal performance of the Terfenol-D alloy [1,5]. Mechanical preloading of the Terfenol-D (in order to maintain a state of compression) was accomplished by tightening the support screws which held the motor to the foundation. A small amount of mechanical loading also occurs when the magnetic field is biased with at DC current. For optimal operation, a total stress biasing of 1000 psi was required [1,5,6,7].

The properties of Terfenol-D that make it so well-suited as a force actuator for mechanical vibration applications is its ability to respond quickly and significantly to a changing magnetic field [1,6,7,8,9]. Specifically, if the rod's natural tendency to expand in a magnetic field is restricted, a large force is developed in the endeavor to expand.

Unfortunately, Terfenol-D also exhibits highly non-linear properties, which can in some cases complicate the modeling process greatly [2,3,7,10]. The non-linear properties of Terfenol-D are a function of loading, magnetic field strength and ambient temperature. Detailed characteristics for Terfenol-D can be found in Clark's "Terfenol Magnetostrictive Materials," complete with graphs of Actual vs. Theoretical Behavior [1], and in the brochure "Typical Material Properties" made available by Etrema Products, Inc. See Appendix 2 for more details, but the general properties of interest for this research project are:

Density	0.33 lb/in^3
Bulk Modulus	$13.1x10^6$ lb/in^2
Tensile Strength	$4.1x10^3$ lb/in^2
Compressive Strength	$101.5x10^3$ lb/in^2

A quick look at these mechanical properties shows that one of Terfenol-D's greatest vulnerabilities is its tensile strength. While Terfenol-D has a very large compressive strength, on the order of 10^5 lb/in^2, its tensile strength on the order of 10^3 lb/in^2 makes it very brittle, breaking easily if mishandled. With a market price of $450 for each 3-inch rod [11], care must be taken at all times to keep it in compression. (See Appendix 3 for a price list.)

Due to the nature of this project, the properties of Terfenol-D which were most important were those relative to preloading and magnetic field strength, specifically elongation as it relates to force output; temperature effects on the rod were reduced by a set of two fans blowing over the coils and short periods of activation. Additionally, the Terfenol-D rods were not in direct contact with the coils, the PVC providing some degree of insulation. For more general applications with more massive rods and coils, some substantial method of cooling, such as water cooling, would be required to prevent the coils and Terfenol-D rods from overheating.

Figure 3, showing the graph of Field Strength vs. Strain [1, 6, 8, 10], reveals the symmetry and hysteresis associated with the expansion of Terfenol-D in magnetic fields; Terfenol-D expands regardless of whether it is subjected to a positive or a negative magnetic field. It can also be seen from the Figure 3 that there are two regions (not accounting for symmetry) where Terfenol-D behaves fairly linearly (if mechanically preloaded with 1000 or 2000 psi): at low magnetic field strengths (250 Oersteds) and at high magnetic field strengths (1500 Oersteds). In order for the actuator rods to operate in a linear region, the magnetic field had to be biased with a DC current. In addition to magnetic field biasing, the rods had to remain in compression, and for this

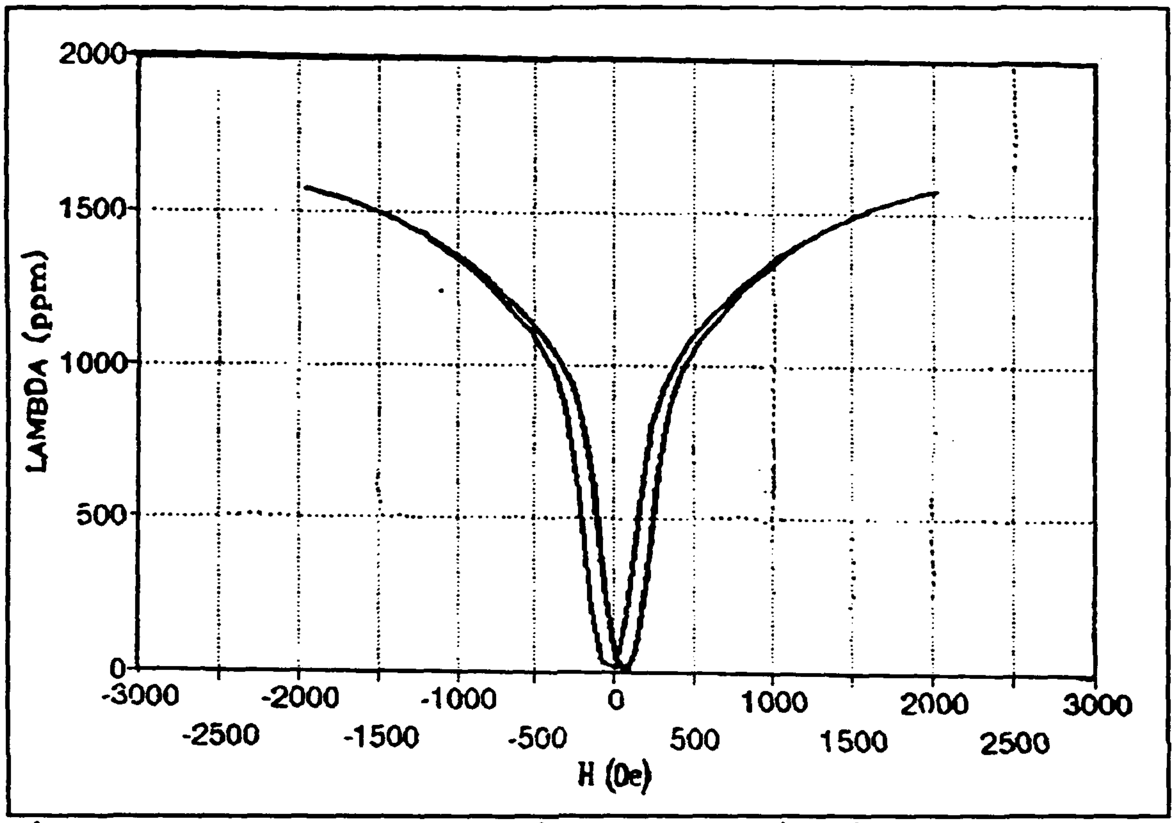

Figure 3 - The graph of Strain vs. Magnetic Field Strength for Terfenol-D reveals its non-linear relationship but also shows where there are nearly linear operating regions.

a preloading set-screw and three Belleville washers were added to the platform and support. (It should also be noted that by biasing the magnetic field, loading was biased as well due to the corresponding displacement of the Terfenol-D rod.) By controlling the magnetic and mechanical biasing, it was possible to take advantage of both positive and negative strains and forces. In order to control the mechanical vibrations efficiently, large, prompt forces were necessary. The short optimum response time suggested that the lower magnetic field region would be most useful as it offered the greatest ratio of strain vs. field strength.

The wire coil responsible for generating the magnetic field for biasing and control was wrapped around a PVC spool, which served at least three purposes: 1) it acted as a housing for the coil; 2) it provided a free zone for the rod to pass through the wire coils; and 3) it insulated the Terfenol-D rod, as mentioned earlier.

The spool of wire did not come into contact with anything other than the o-ring spacers which kept an air gap between it and the Terfenol-D rod. This was accomplished by making the spool slightly shorter than the Terfenol-D rod, allowing the rod to protrude from both ends, as can be seen in Figure 4. This configuration was favorable because it also allowed the actuator rod to interact directly with the surfaces of the system, simplifying modeling procedures. Although careful modeling of the magnetic field strength degradation through the spool might prove to be necessary for larger or more advanced applications, it was considered negligible for this project.

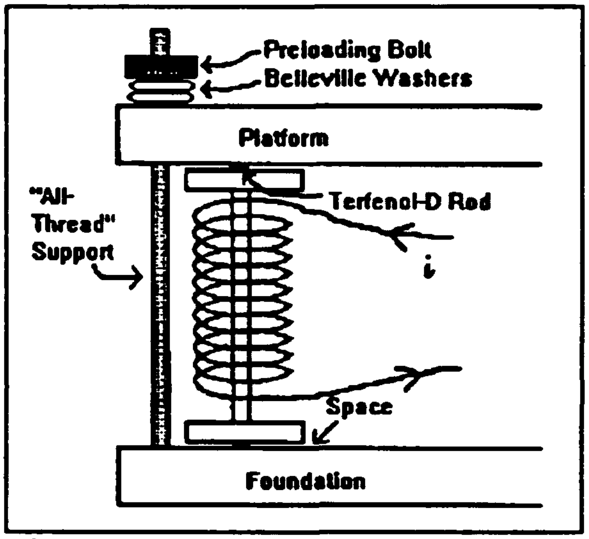

Figure 4 - The actuator assembly, including the platform, foundation, set screw, coil and Terfenol-D positional relationship.

The second major component of the active controller is

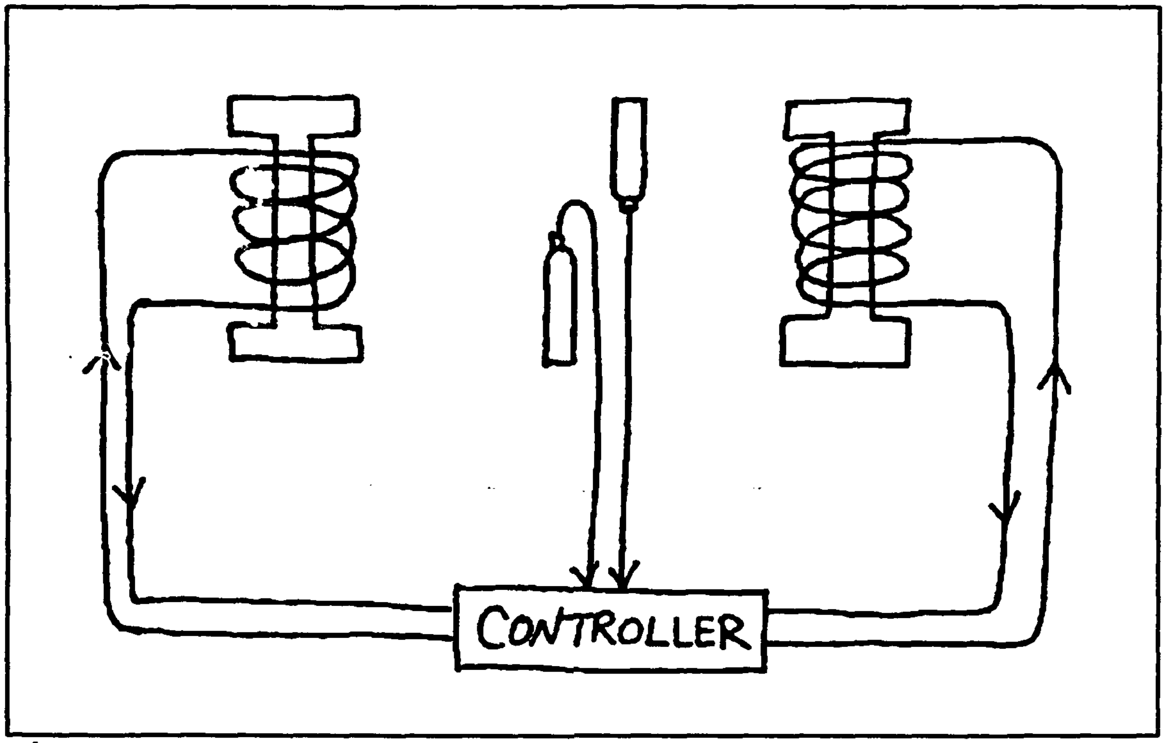

Figure 5 - The layout of the actuators, feedback accelerometers and master controller.

the master controller, which is no less important to the vibration elimination process than the revolutionary Terfenol-D alloy itself. Using an analog controller has inherent disadvantages (such as phase lag), but because of its simplicity and size for eventual "real" applications, it was the type of controller chosen for this project.

Finally, the third major component of the controller is the sensor array. Sensors are a vital part of this dynamic system because they provide the necessary feedback information to the controller. Although only accelerometers were used for this project, others sensors, such as force rings, strain gauges, and vibrometers would be effective as well. Mounting strain gauges to the Terfenol-D rods, though not incorporated into this project, is common practice (for current feedback) and should be considered for future work [2,3,7,10]. Figure 5 is a diagram of the actuator and sensor layout.

ANALYSIS OF AN IDEAL SYSTEM

The primary objective of this project was to actively control vibrations, for which the controller must be kept constantly "informed" of the system's current status. Additionally, for the controller to predict what signal should be produced to eliminate any given vibration, it must be known how the system will react to any given input by the actuators. This requires an accurate model of the entire system, including the presence of the actuators - the more finely tuned the model, the better the results. In order to better understand the system as a whole, it had to first be broken down into its constituent parts. Some systems are easily modeled by low-order equations, but in this particular case, many low-order systems had to be combined.

As a key element to the control effort, the only moving part was the Terfenol-D rod in the center of a magnetic coil. Due to the properties of the magnetic fields and the Terfenol-D, the actuator itself had to be well understood if a usable system model was to be made. Associate Professor Reed has already done extensive work on modeling the actuator and his work was the foundation of this project [2,3,7,10]. The block diagram he developed of the actuator assembly is shown in Figure 6, which includes the effects of amplifiers, magnetic coils, Terfenol-D rods and strain feedback. Following Figure 6 is the corresponding transfer function given by Equation

(1). Professor Reed demonstrated that the actuator could be modeled with this relatively simple equation for moderately simple systems.

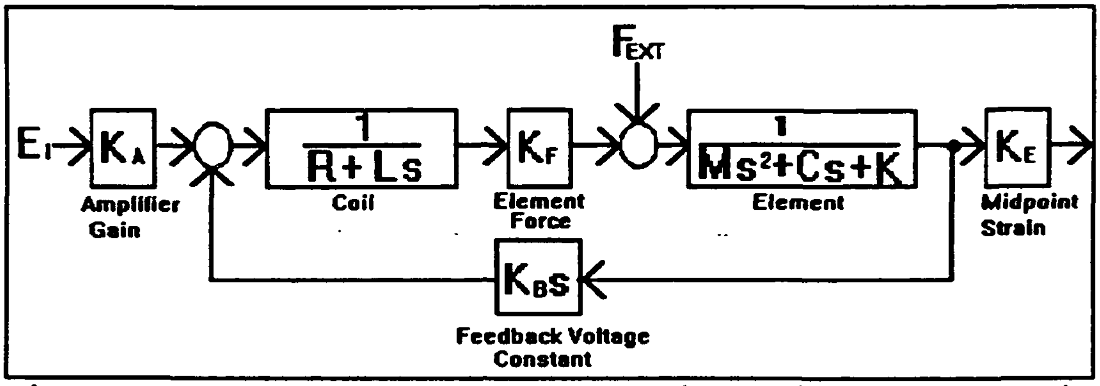

Figure 6 - A model of the actuator, including the magnetic coil, the Terfenol-D rod and midpoint strain feedback.

$$\frac{Inch}{Volt} = \frac{1.16}{s^3+4.9s^2+6570s+5570} \tag{1}$$

The next part of the overall system to be modeled was the supporting mechanism. In Figure 7, the model of one support shows how it can be comprised of several incompressible mass components, each connected by a spring and a damper. The larger mass at the top accounts for a portion of the motor and motor platform masses which were resting on the supports. In the model shown here, only four masses are used, primarily to make the presentation clearer; the actual calculations used for the project used ten masses for enhanced resolution and quality of the simulation. The values of the individual mass units were calculated by taking the total mass of the support and dividing it evenly over the total number of mass units modeling it. In some cases, it was necessary to add mass to some internal mass units to account for cases of support-mounted actuators, so uniform mass density was not assumed in the modeling process. The spring constant between each mass was determined as a function of the modulus of elasticity and the dimensions of the material used, in this case aluminum, but unlike the mass units, they were considered to be uniform throughout the support. The damping constant for the system was a little more difficult to obtain and had to be determined experimentally. Once calculated, the damping constant was also considered to be uniform. For any graphs included in this report, the values used for damping are not

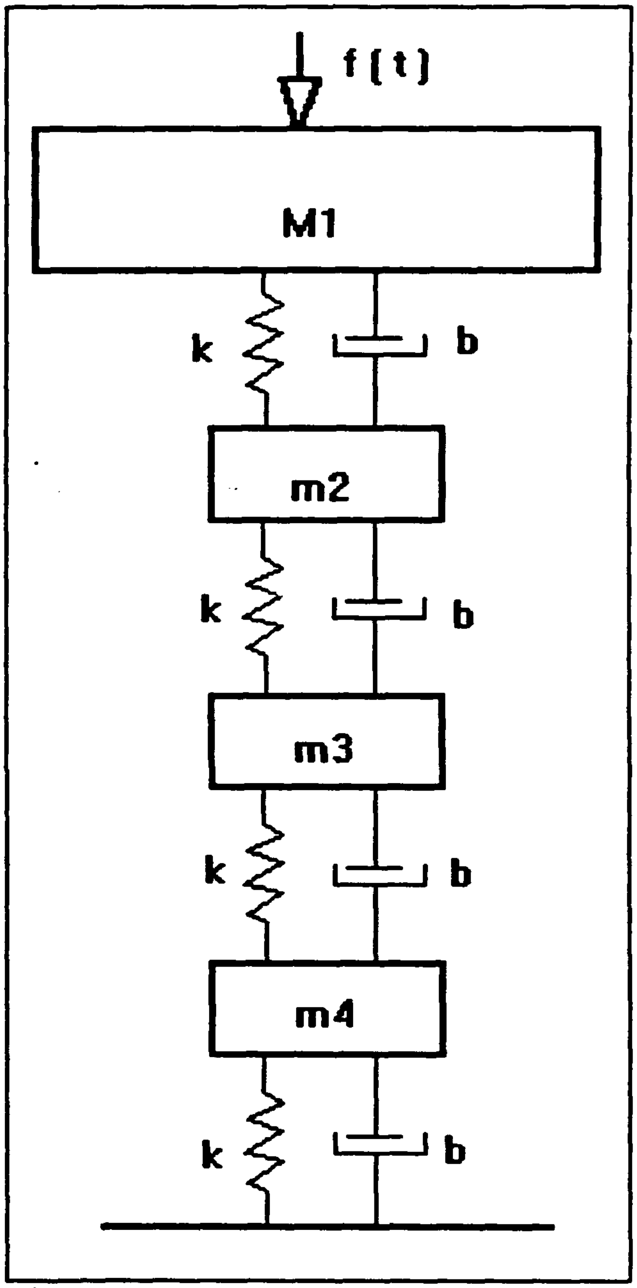

Figure 7 - A model of the supports shows how it can be represented by several incompressible mass units connected by springs and dampers.

actual values calculated for this particular system, but rather values which were chosen to better demonstrate how the system operates.

For all good mathematical models, an appropriate transfer function has to be determined [12, 13]. For the system shown in Figure 7, comprised of non-uniform mass units but assuming a uniform spring constant and damping coefficient, one can determine that

$$m_1X_1'' = f - b(X_1'-X_2') - k(X_1-X_2),$$

$$m_2X_2'' = b(X_1'-X_2') + k(X_1-X_2) - b(X_2'-X_3') - k(X_2-X_3),$$

$$m_3X_3'' = b(X_2'-X_3') + k(X_2-X_3) - b(X_3'-X_4') - k(X_3-X_4),$$

et cetera.

The state variables for the system, indicated by X_1, X_2, X_3 and so on, represent the mass positions and their velocities. Here, X_1 is the position of mass m_1 and X_2 is the velocity of mass m_1, while X_3 and X_4 are the position and velocity for the mass m_2. For a four-mass system, $\mathbf{X}' = \mathbf{I}(\mathbf{AX} + \mathbf{BU})$, where

$$\mathbf{A}=\begin{bmatrix} 0 & 1 & 0 & 0 & 0 & 0 & 0 & 0 \\ -k/m1 & -b/m1 & k/m1 & b/m1 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 1 & 0 & 0 & 0 & 0 \\ k/m2 & b/m2 & -2k/m2 & -2b/m2 & k/m2 & b/m2 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 1 & 0 & 0 \\ 0 & 0 & k/m3 & b/m3 & -2k/m3 & -2b/m3 & k/m3 & b/m3 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 1 \\ 0 & 0 & 0 & 0 & k/m4 & b/m4 & -2k/m4 & -2b/m4 \end{bmatrix}$$

For larger systems, **A** increases by two rows and columns for each new mass, while **B** adds 2 rows - this phenomena is caused by the fact that each mass represented in the model requires two state variables, namely its position and its velocity. Although the matrix above did not assume equal masses for each state ($m_1 = m_2 = m_3 = m_4$) such will be the case for all but the uppermost mass unit (m_1) unless there is something connected to it (like an actuator).

The simulation diagram for this multiple mass system is easily derived [14, 15] (though the block diagram is rather complicated). Looking at the simulation diagrams in Figure 8, it can be seen how additional masses effect the system. In the series of simulation diagrams presented, the first represents a single mass system, the second, a two mass system, and the third, a three mass system. Although G1, G2, H2, and so on, are not easily derived, their relative relationships are. The nodes of the diagram represent the individual mass positions.

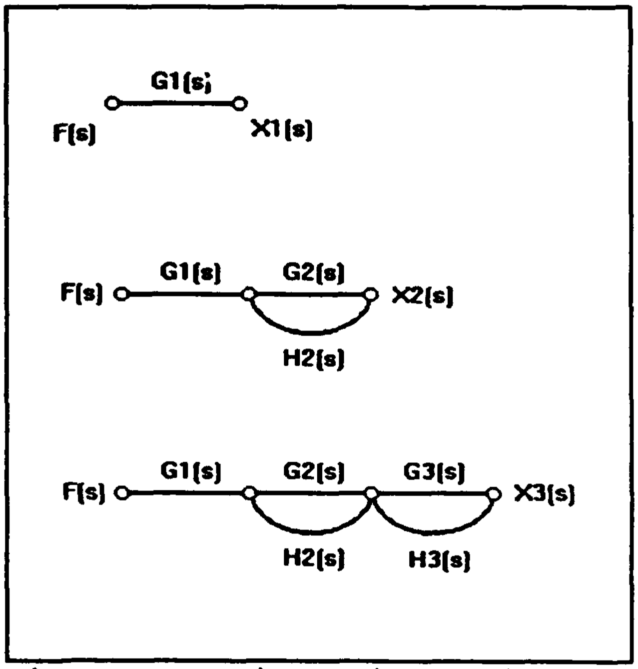

Figure 8- Simulation Diagrams representing the supports with multiple mass units.

With the system now properly modeled, the next step was to determine the natural frequency of the system. To save time, this was accomplished with the aid of a computer An impulse force was applied to the upper mass (denoted by the arrow labeled f(t) pushing down on m_1 in Figure 7) and the movement of each mass was graphed as a function of time. The frequency of the resulting oscillation is the natural frequency (also known as the resonant frequency) of the system. While more mass units tend to produce better results, computer speed and memory limitations make it impractical to include too many mass units in the model.

After mathematically modeling and testing the unmodified system, the actuators and sensors had to be added to the model. There are many ways to apply a controlling force to the system and several of them were investigated. In all cases, recall that the actuators had to exert an equal and opposite force at both contact points and that any constant (biasing) force would have no effect on the dynamics of the system. Figure 9 shows the setup for another resonant frequency experiment, this time with a constant (biasing) force being applied where the actuator is mounted. This constant force simulates both the loading bias and the magnetic field bias required to optimize the effectiveness of the Terfenol rods. The theoretical analysis determined that the system must still have the same natural frequency as it did without the constant force being applied, the only difference being that each mass unit would have an initial displacement around which it oscillated.

In Figure 9, the actuators are shown connected at both

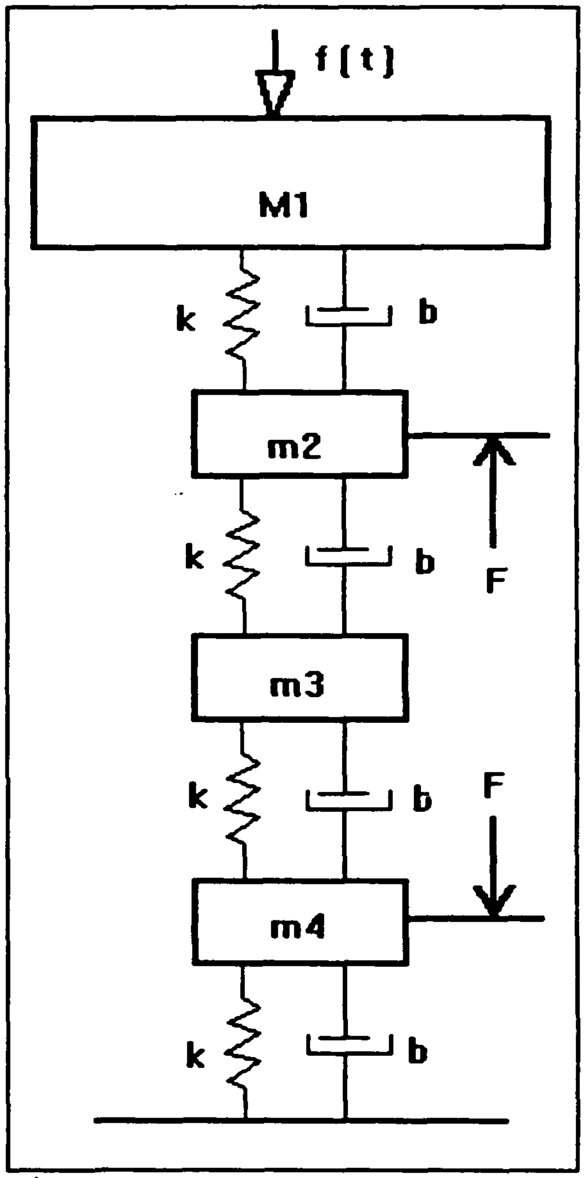

Figure 9 - The support-mounted actuator must act equally on the two mass units it is connected to.

ends to the supports only. Assuming that whatever feedback necessary could be obtained, the essence of the simulation is at hand: vibration elimination. By exactly countering the force on mass 3 from mass 2, it would follow that mass 3 would have a new net force of zero and therefore zero displacement. Using the configuration shown in Figure 9 with appropriate feedback, a plot of the upper mass' displacement was found, and as predicted, the lower masses were still while the upper masses were unstable. On real models, of course, this could not happen -- either non-linear effects would be dramatic, or less likely, the system would fail.

It could be shown that the same results occur for every possible combination of connection points for the support-mounted actuator. In order to avoid this particular problem, one of the masses connected to the actuator had to be smaller than the other. This could be done in one of two ways: 1) if the actuator were mounted such that one end was connected to the support and the other end was not (either connected to the motor platform or the foundation), or 2) if neither end was attached to the support (both ends directly contacting the larger (unequal) masses on top and bottom. Both of these latter cases warranted further investigation.

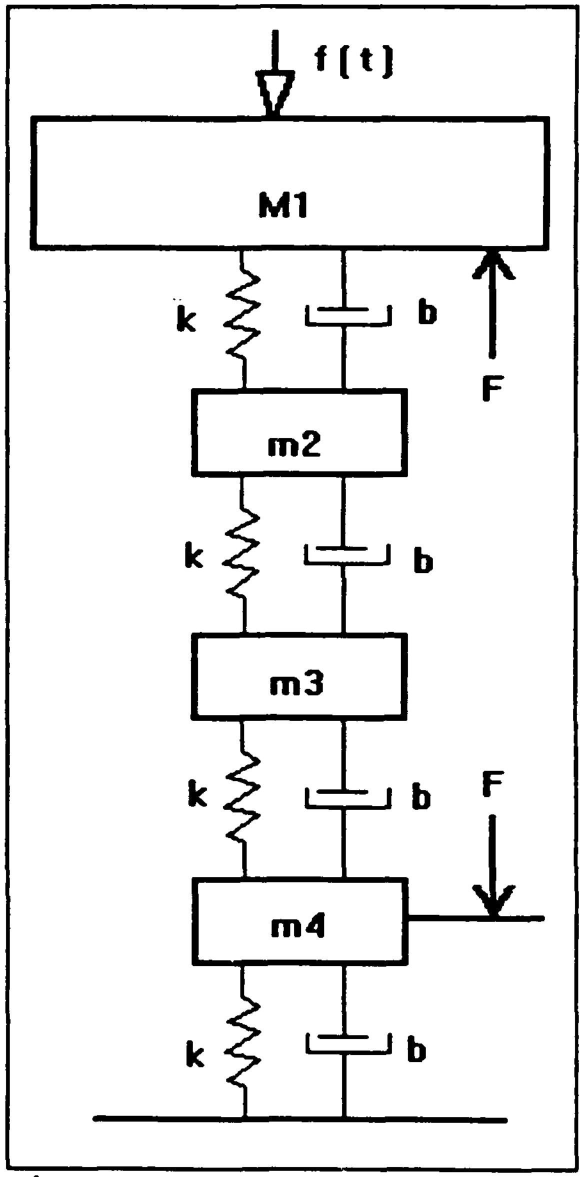

Figure 10 - The actuator can be mounted to the support at only one end, the other end at either the platform or the foundation.

The first of these cases to be studied involved the actuator connected at one end to the support and at the other end to the motor platform, as shown in Figure 10. When driven at the resonant frequency and controlled as necessary, the vibrations at the foundation were indeed eliminated and, interestingly, the nearer together the two ends of the actuator (fewer mass elements between them), the less vibration the platform itself experienced (contrary to what happened in the case of the support-mounted actuator).

Finally, the last case to be looked at involved an actuator that completely bypassed the support and contacted the platform and foundation directly. Once again it was found that vibrations in the base were completely canceled, as expected, and the motion of the upper mass did not become unstable.

For all of the simulations performed so far, it was assumed that whatever feedback force was linear and attainable. In fact, the optimal feedback is very difficult to obtain and as a compromise, other forms of feedback must be used instead. Due to physical limitations, few accelerometers, strain gauges and force rings could be used, and certainly not in the middle of the support mechanism.

Modeling the force input from the motor was done using simple sinusoids and combinations of sinusoids. The project used a DC motor with some degree eccentricity, which, according to early experiments, followed this simple model sufficiently well once the harmonic vibrations caused by the brushes were filtered out (the magnitude of the high-frequency

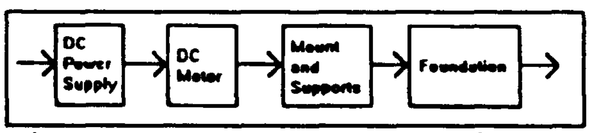

Figure 11 - The block diagram of the unmodified vibration system.

vibrations which were eliminated by the filters were insignificant compared to the magnitude of the lower frequency of the eccentric masses). The simple vibrating system is represented by the block diagram presented in Figure 11.

The final component to be modeled was the controller itself. Because the emphasis of this project was just to show the feasibility of eliminating vibration transmissions in rotary equipment, the size of the controller was not an immediate concern. For development of the controller, a Hewlet-Packard digital analyzer was used. The analyzer is capable of receiving two inputs and developing a transfer function defining their relationships. It also has a source function which is programmable with poles and zeros as necessary. The analyzer was the primary driver of the

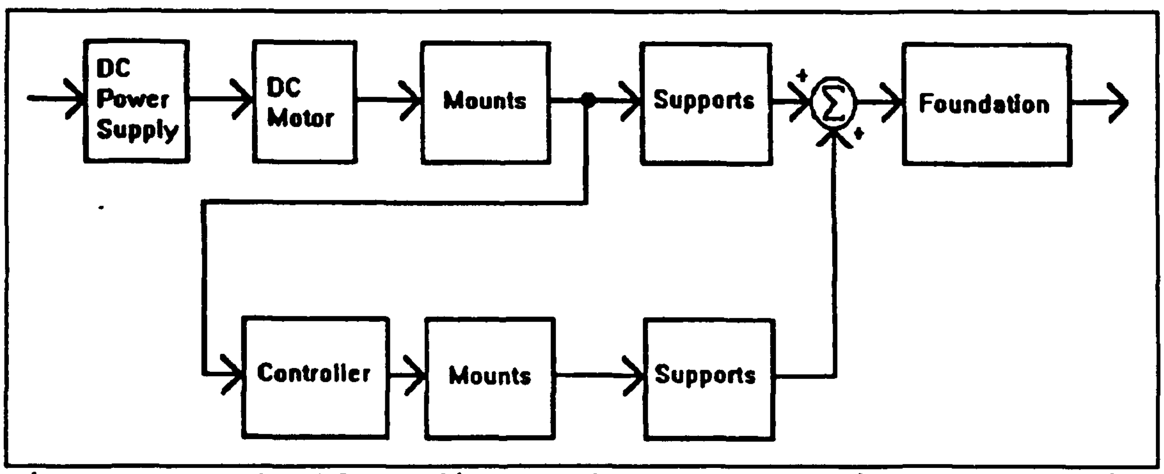

Figure 12 - The block diagram for a system with acceleration feedback.

actuators, but the signal was amplified by an external power amplifier which also added the necessary DC current biasing for the actuator. Experiments by Professor Reed showed that acceleration, velocity, and actuator strain feedback would be necessary [2, 3]. Other types of feedback might be beneficial but would only be discovered by experiments. Earlier in this section, mathematical model indicated that feedback from several states in the supporting mechanism would be beneficial, but there was no way to measure those parameters. Figure 12 shows just one possibility of feedback, and Figure 13 shows a more detailed example of a reasonable feedback loop.

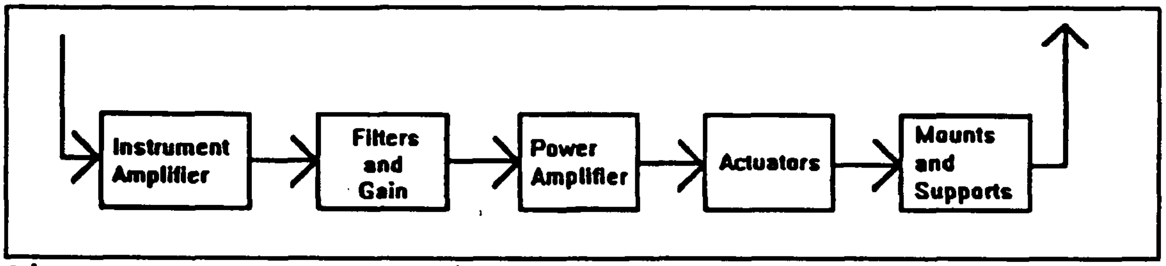

Figure 13 - A more detailed feedback loop.

MODEL TESTING AND EVALUATION

After building the physical model, it was necessary for it to undergo several tests. The first was to determine the

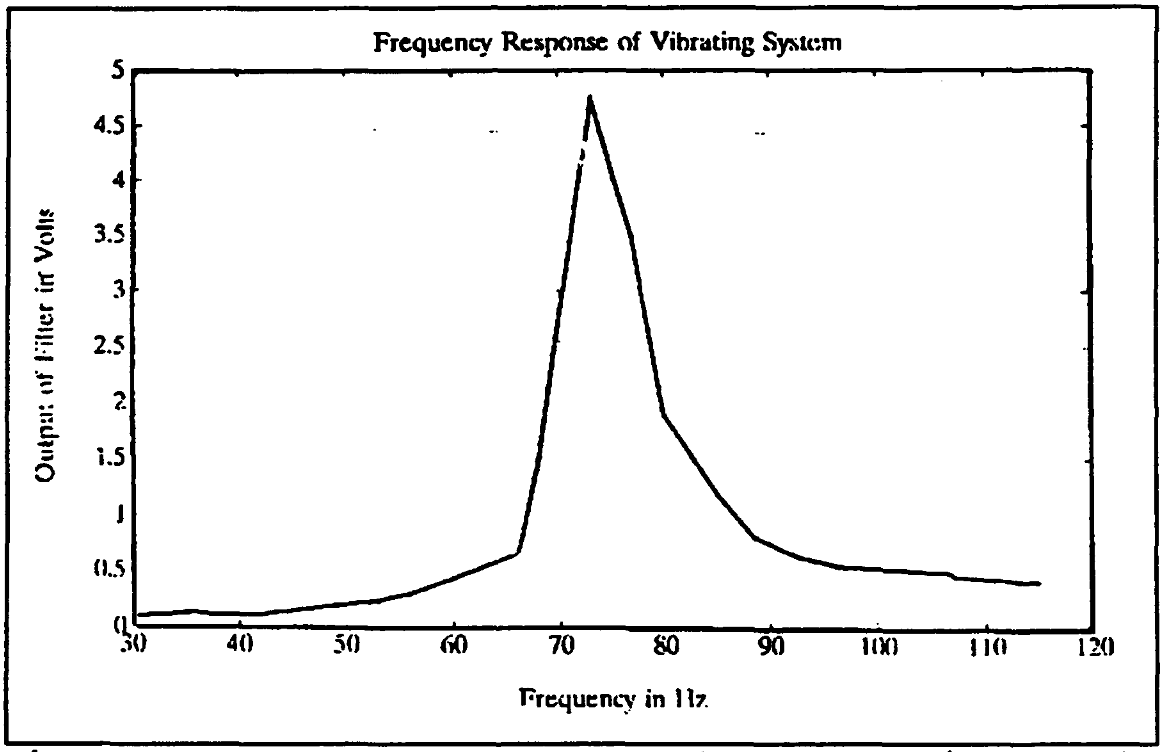

Figure 14 - Frequency response of the system driven by the motor only.

motor's tachometer constant. It was found later that the tachometer voltage feedback was beneficial to use in a comparator in order to determine when the controller should be activated and when it should be de-activated. Next, as shown in Figure 14, the motor was run through its full frequency range while the acceleration, represented by the the output magnitude of the filter, was plotted against the frequency of the motor. This graph revealed that the resonant frequency of the system was about 74 Hertz. Next, driven by an independent oscillator and power amplifier, the actuators were used to vibrate the system. Figure 15 shows the response of the system for the range of the motor but driven by the actuators.

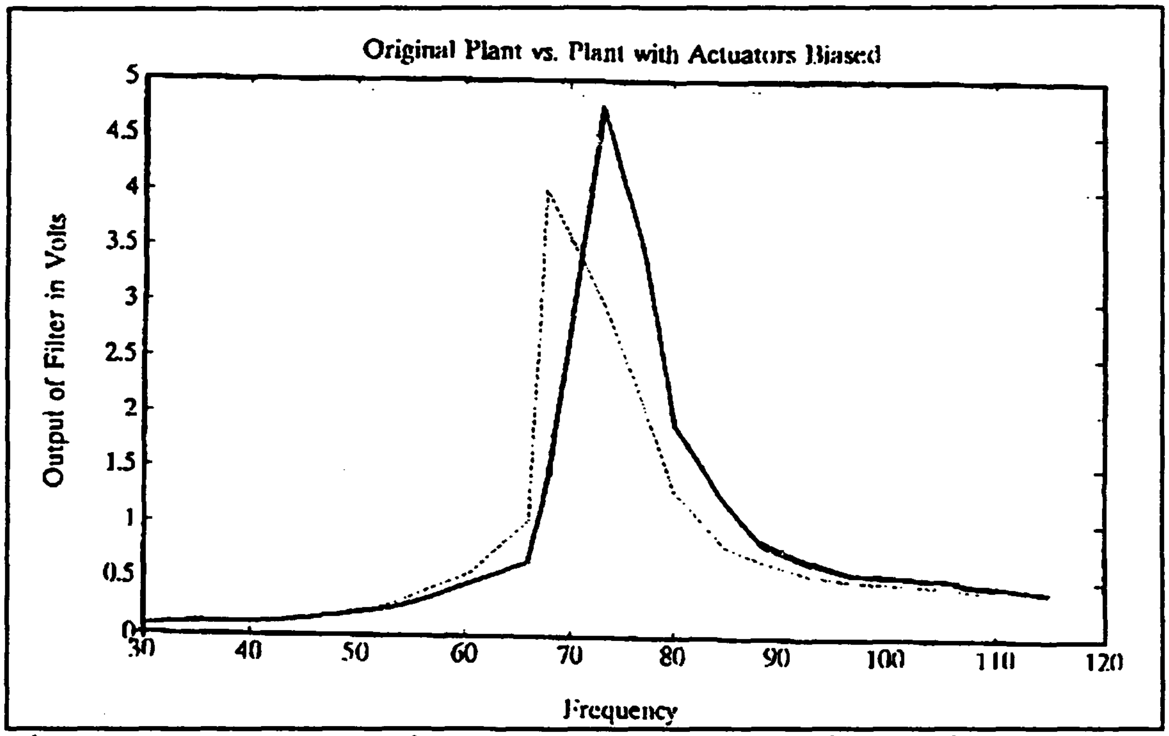

Figure 15 - A comparison of the system being driven by the motor and the actuators (with biasing).

As Figure 15 shows, the system's resonant frequency in the range of the motor was shifted somewhat to the left (lower than 74 Hertz) when driven by the actuators, and the output magnitude was decreased as well. This is not accounted for by the extra force applied by the actuators (recall from the theoretical analysis that any biasing force does not change the natural frequency of the system). It is suspected that this is a result of the back electromotive force which is generated when the rod is deformed in the presence of a magnetic field (refer back to Figure 6). Further tests, including one driven by the motor with the actuators biased, also showed this to be the case; with the actuators unbiased, the natural frequency of the system was unaffected.

After this series of tests, driving the system with both the actuator alone and the motor alone, the two forces were applied simultaneously (a simple form of open loop control). It was found that for any frequency above 30 Hertz, vibrations from the motor could be matched by the actuators and be completely canceled. Below 30 Hertz, the magnitude of the vibrations in this system was too small to notice any affects. The only instrument measurements being taken at the time were from an accelerometer whose signal was filtered and then displayed on an oscilloscope.

Inspired by the open loop results, direct feedback was led from the accelerometer output to the actuator. Although initially discouraging, once a defective power amplifier was replaced, there was a significant indication of success, which improved with a slight change in the filter and gain circuit. The final filter compensator design used for the project is shown in Figure 16 and again in more detail in Appendix 4.

Because of the potential benefits of having only one

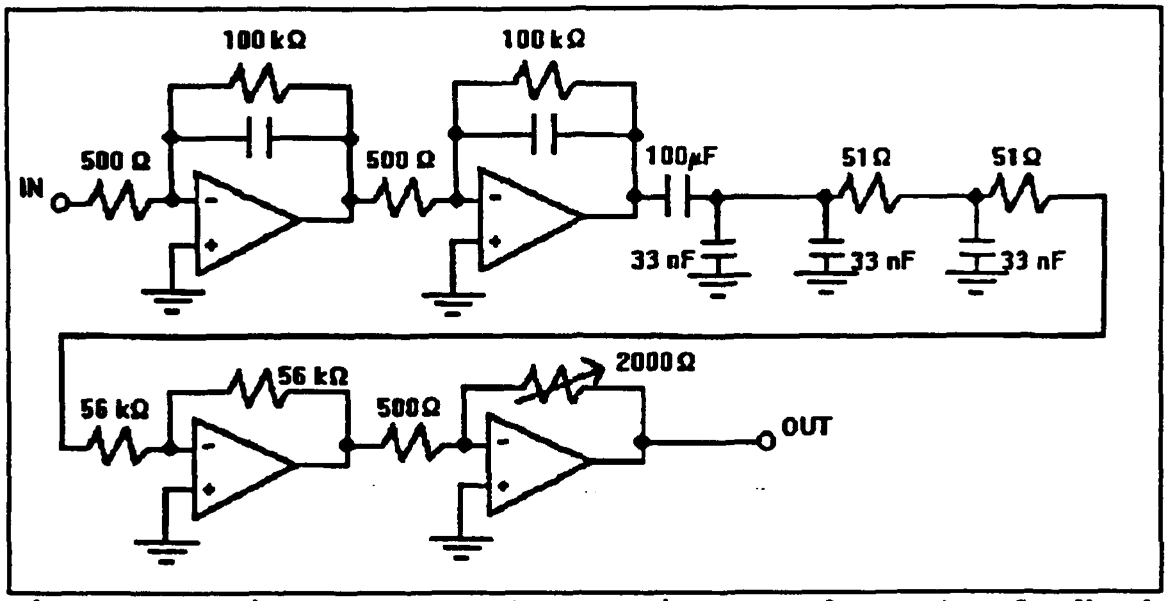

Figure 16 - Filter compensator receives accelerometer feedback and provides the actuator driving signal.

sensor necessary to monitor and provide feedback to a system, no other feedback was added as better solutions were sought using just this one accelerometer. Lead compensators were added to try to compensate for some of the delay time associated with low-pass filters, but due to the nature of the filters, too much phase shift was occurring in the frequency range of the greatest interest. Eventually, all of the lead compensators added for that purpose were removed. A Bode diagram relating the characteristics of the actual filter used for the project is found in Appendix 5 and reveals the large phase shift which occurs throughout the operating frequency range.

By running the motor through its full range of frequencies, it was determined experimentally, as is shown in

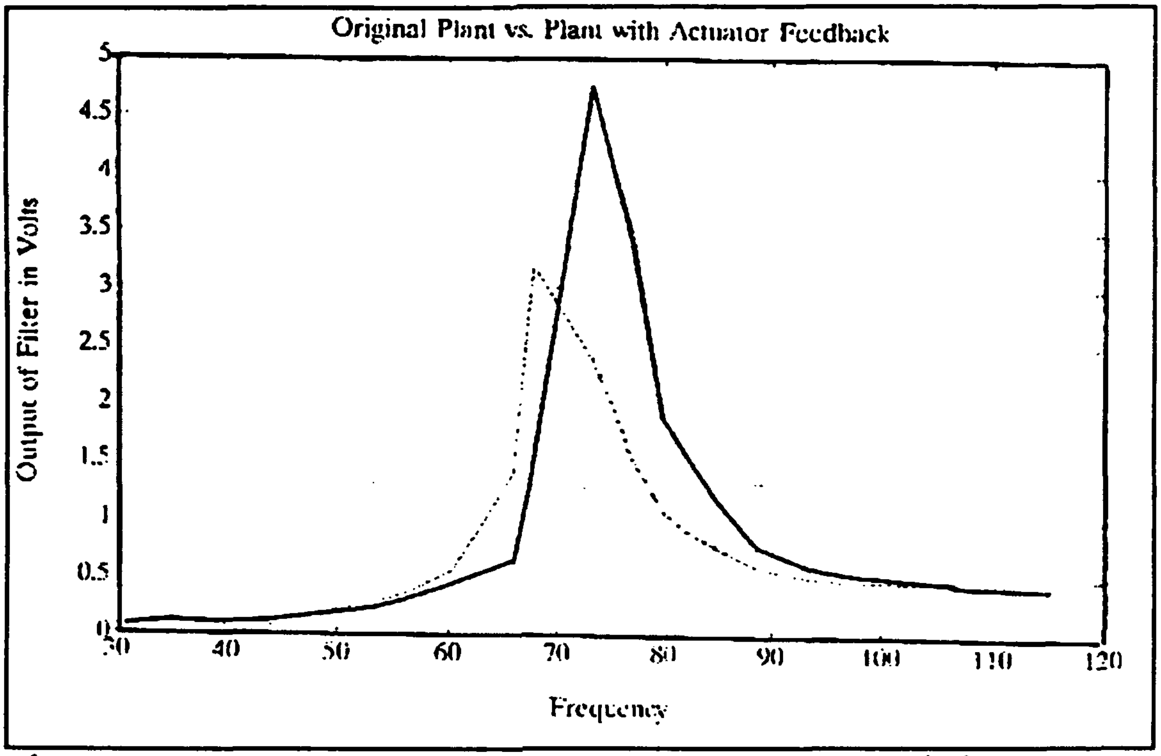

Figure 17 - The natural response of the unmodified system compared to the response with the accelerometer feedback control.

Figure 17, that the only zone for which the analog filter compensator would be of use was that of the resonant frequency. Since the resonant frequency of the biased actuators occurred prior to the natural frequency of the unmodified system, feedback from the tachometer was added in order to activate and deactivate the controller biasing around the resonant frequency. The only portion of the controller manipulated was the current (magnetic field) biasing; acceleration feedback was always sent to the actuators although it had little effect except for within the resonant frequency range. For frequencies not near the resonant frequency, gain was very limited due to instabilities

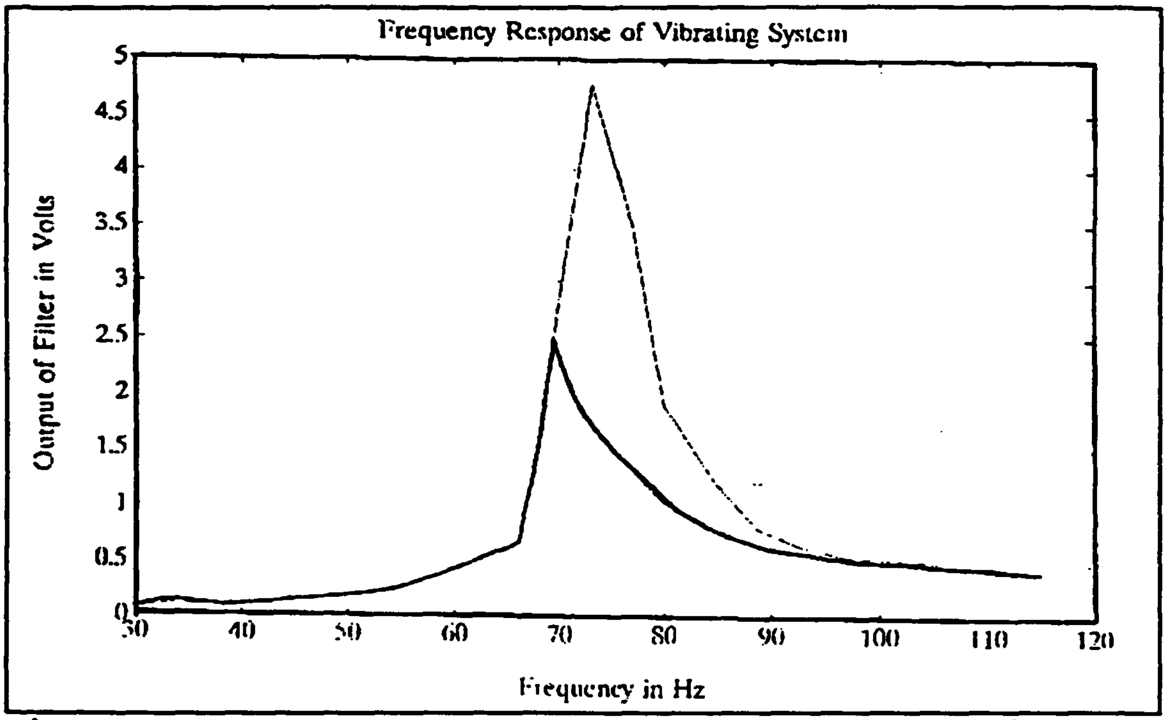

Figure 18 - The frequency response of the system with both acceleration feedback and frequency activation control.

(manifested as "chattering") caused by the closed loop system, while a slightly higher gain was possible within the resonant frequency range. The overall effectiveness of the new controller in the region of the resonant frequency was noteworthy, as can be seen in Figure 18, a graph of the "Smart" controller.

As an interesting side note, it should be mentioned that "chattering" was not a problem when the actuators were driven by an independent oscillator at the same frequency and magnitude. In fact, at any frequency, and for all but extremely large signals, "chattering" was no problem with the open loop driver at all. This would suggest that a digital controller which creates and sends its own signal to the actuators would not have similar problems due to its own ability to filter out the noise and send pure sinusoids. (A recommendation for a digital controller will be included the "Future Research Possibilities" section.)

CONCLUSION AND FUTURE RESEARCH POSSIBILITIES

While this project was primarily concerned with finding an analog controller to reduce vibrations in rotational equipment, many findings along the way indicated other paths one could take to solve the problem. One method, the use of a digital controller, was already mentioned earlier in the paper. Another approach might be to develop a voltage controlled oscillator which drives the actuators at a frequency determined by the voltage of the tachometer output, while the phase of the signal could be triggered by a gate on the shaft of the motor. Also, while this research was initially targeting the periodic vibrations of rotational equipment, a very fast controller would most likely be able to handle low-impact shocks.

Vibrations caused by rotational equipment are very common, and more effort should be placed on developing active controllers to thwart its effects on materials and humans alike. Many large industries which depend on providing luxuries are very interested in this type of research and this project demonstrated that even the simplest of feedback dramatically reduces the vibrations transmitted through one system to another. While this analog controller was limited to the resonant frequency of the system, such limitations are not predicted for more advanced controllers, such as micro-processor controller actuators.

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Appendix 1
Construction Details

The physical model used to test the theoretical results obtained used a DC motor with symmetrical shafts and a tachometer mounted on a specially designed aluminum table. The motor was mounted in the center of the platform and two eccentric masses were placed on the motor (one on both shafts) such that any vibrations created would primarily be uniform and vertical to the plane of the system.

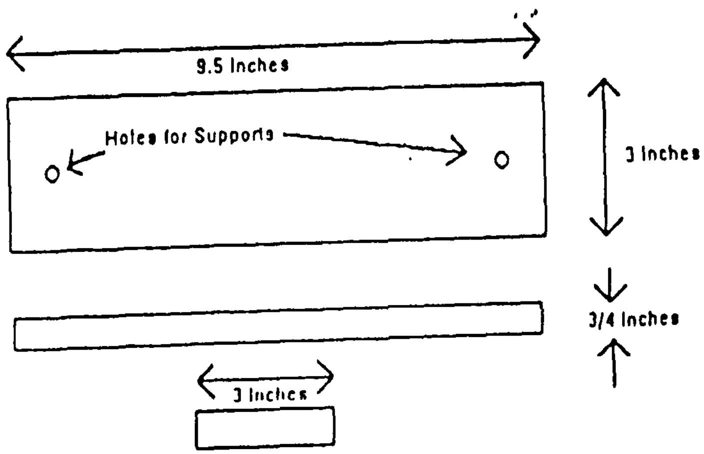
The table consisted of two pieces of aluminum measuring 3.0 by 9.5 by 0.75 inches (one for the platform and one for the foundation). The supporting mechanism was a single "all-thread" screw passing through the center of each edge of the table. It was originally planned that the supports would be aluminum too, but the resulting resonant frequency was out of the range of the motor, which could only manage 150 Hertz. The whole table assembly was then mounted to a piece of aluminum representing the recipient of the vibrations transmitted by the foundation. It measured 12 by 15 by 0.5 inches.

The actuators were constructed from PVC and were 3 inches tall with a 0.5 inch (outer diameter) center shaft and 2 inch diameter ends. The center shaft inner diameter was slightly larger than 0.4 inches to allow the Terfenol-D rod and strain gauges to be pass through. Hand wrapped wire around the spools allowed for the magnetic field necessary to control the output force created by the Terfenol-D rods.

Appendix 1
Construction Details

Other equipment used in the finished project was an oscilloscope to monitor the vibrations detected by the accelerometers, a 20 Volt, 20 Amp power amplifier to drive the actuators and a 35 volt DC power supply to drive the DC motor. The controller itself used a +/- 12 Volt power supply. The feedback accelerometer required a small instrument amplifier in addition to the gain in the controller circuit.

MOTOR PLATFORM



* The Foundation is the same size and shape, but it has four holes drilled for mounting it to the "Surroundings," in this case a piece of aluminum measuring 15" x 12" x 1/2".

NOTES

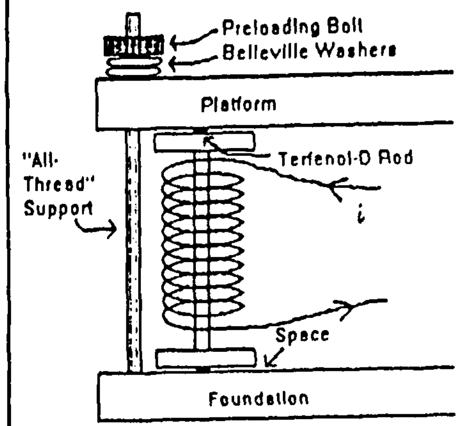
1) The holes drilled for supports are threaded in the foundation but not in the platform mount.

2) The supports were 1/4" steel "All-Thread."

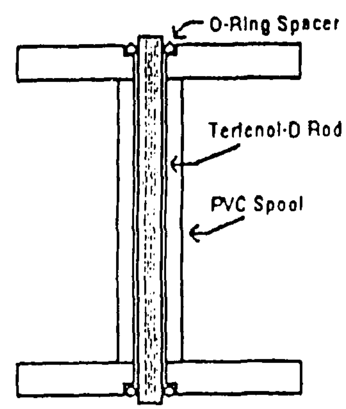
3) Belleville washers were used between the platform and the pre-loading bolt.

4) The space between the platform, the actuators, and the foundation was maintained by a firm sponge.

ACTUATOR ASSEMBLY

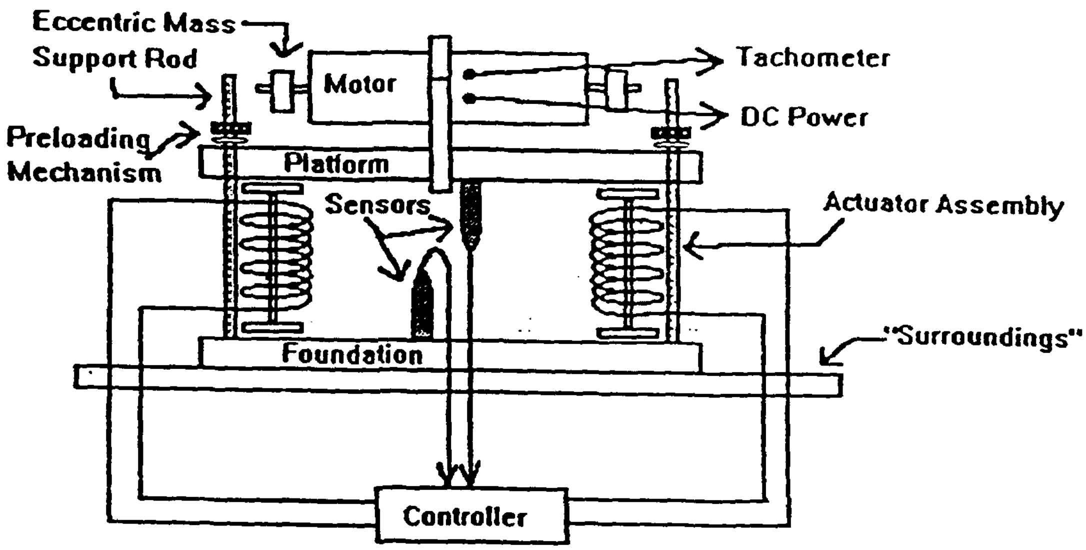


SPOOL

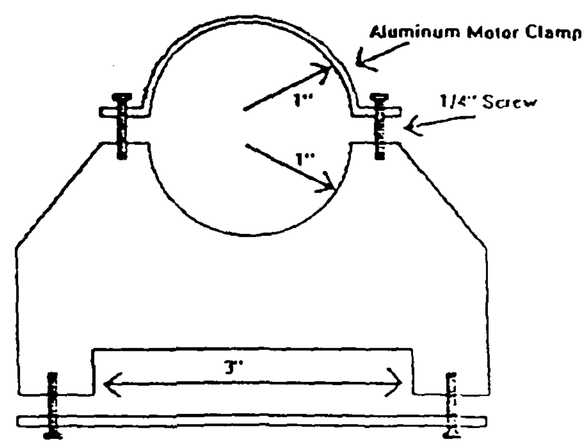


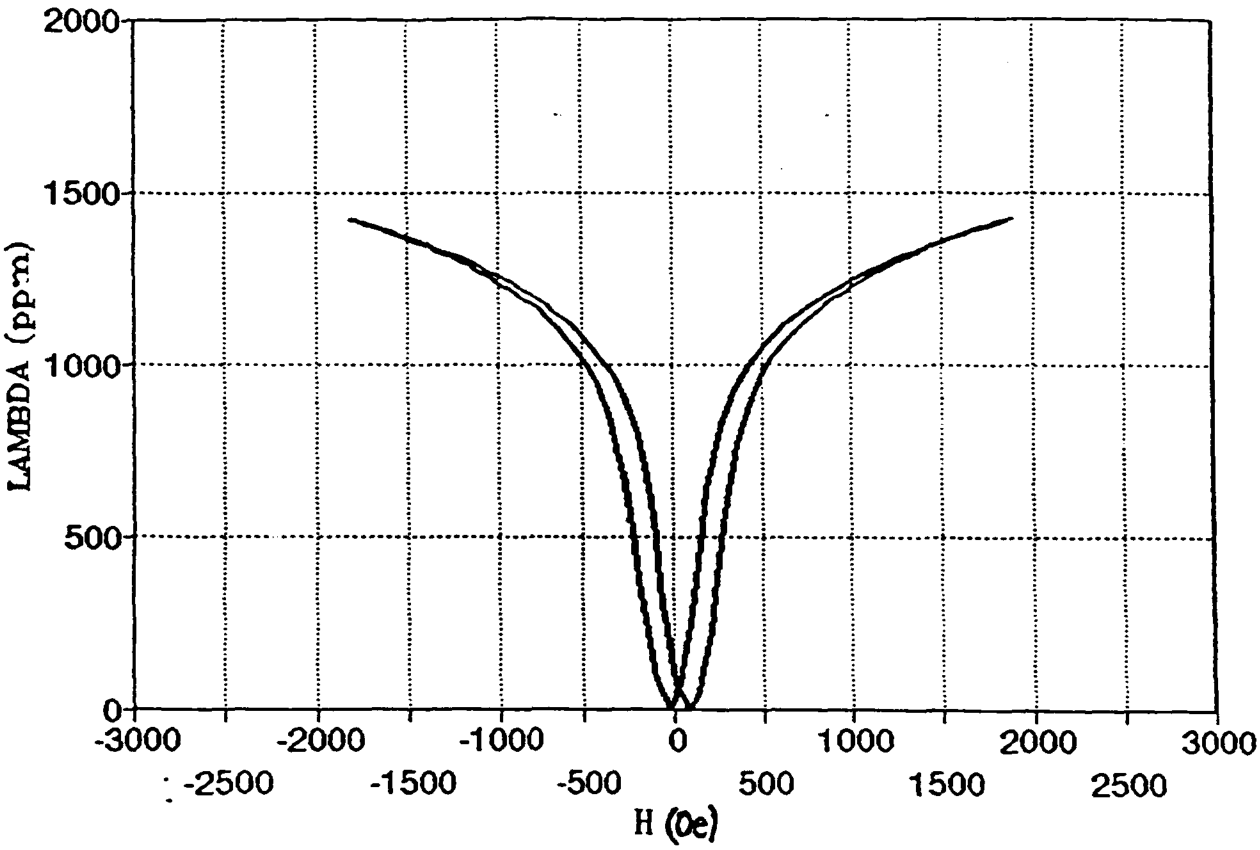
Appendix 1
Construction Details

COMPLETE SYSTEM



MOTOR CLAMP





A plot of Terfenol-D's Strain vs. Field Strength. Note the non-linear relationship and hysterises.

Appendix 2
Properties of Terfenol-D

TYPICAL MATERIAL PROPERTIES

	TERFENOL-D		CERAMIC (PZT)	NICKEL
	METRIC	ENGLISH	ENGLISH	ENGLISH
1. MECHANICAL				
Density	$9.25x10^3 Kg/m^3$	$0.33\ lb/in^3$	$0.27\ lb/in^3$	$0.32\ lb/in^3$
Bulk Modulus	$9.0x10^{10} N/m^2$	$13.1x10^6 lb/in^2$		$27.2x10^6 lb/in^2$
Young's Modulus	$2.5\text{-}3.5x10^{10} N/m^2$	$3.6\text{-}5.0x10^6 lb/in^2$	$10.5x10^6 lb/in^2$	$31.7x10^6 lb/in^2$
Tensile Strength	28 MPa	$4.1x10^3 lb/in^2$		$44.9x10^3 lb/in^2$
Comp Strength	700 MPa	$101.5x10^3 lb/in^2$		
2. THERMAL				
Expansion Coefficient	$12x10^{-6}/°C$			$13.3x10^{-6}/°C$
Specific Heat Capacity	0.32 to 0.37 J/g/°K			
Thermal Diffusivity	0.035 to $0.03\ cm^2/s$			
Thermal Conductivity	10.5 to 10.8 J/°C/m/s			$613\ Btu/ft^{-2}h°F$
3. ELECTRICAL				
Resistivity	60 μΩcm	$2.36x10^{-5}\Omega in$		$.272x10^{-5}\ \Omega in$
4. MAGNETIC				
Magnetization	1.0T	$1.0x10^4 Gs$		
Curie Temp.	380°C	716°F		
5. MAGNETOSTRICTIVE				
Magnetostriction	1500-2000 ppm		*100-300 ppm	-40 ppm
Energy Density	$14000\text{-}25000\ J/m^3$	$.38\text{-}.67 Btu/ft^3$	$.026\ Btu/ft^3$	$.00081\ Btu/ft^3$
6. MAGNETOMECHANICAL				
Relative Permeability	5-10			
Coupling Factor	0.7-0.75		0.65	0.30
Sound Speed	1720 m/s	$5.6x10^3$ ft/s	$10.2x10^3$ ft/s	$16.1x10^3$ ft/s

*Electrostrictive

2/90

Appendix 3
Price List for Etrema Terfenol-D Products

ETREMA PRODUCTS, INC.
Subsidiary of Edge Technologies, Inc.

April 1, 1991

LIST PRICES FOR ETREMA TERFENOL-D®
ALL PRICES IN U.S. DOLLARS
$Tb_{0.27-.3}\ Dy_{.73-.7}\ Fe_{1.9-1.95}$ *

STANDARD RODS

FREE STAND ZONE MELT (FSZM) PROCESS

Diameter mm	Diameter inch	Price Per Linear inch
4	0.16	110.00
5	0.20	125.00
6	0.25	125.00
7	0.275	125.00

The patented ETREMA FSZM process produces the highest performance Terfenol-D commercially available. Contact ETREMA Products, Inc. for typical performance data.

MODIFIED BRIDGMAN (MB) SOLIDIFICATION

Diameter mm	Diameter inch	Price Per Linear inch	Diameter mm	Diameter inch	Price Per Linear inch
8	0.32	110.00	28	1.10	610.00
9	0.35	120.00	30	1.20	685.00
10	0.40	130.00	32	1.25	775.00
12	0.50	150.00	34	1.35	875.00
14	0.55	185.00	36	1.40	975.00
16	0.60	215.00	38	1.50	1075.00
18	0.70	275.00	40	1.60	1185.00
20	0.80	325.00	42	1.65	1295.00
22	0.85	385.00	44	1.75	1400.00
23	0.90	420.00	46	1.80	1505.00
24	0.95	460.00	48	1.90	1610.00
26	1.00	525.00	50	2.00	1715.00

Discounts are available on large quantities. Contact ETREMA Products, Inc. for a quotation.

*STOICHIOMETRY, PERFORMANCE DATA AND ROD SPECIFICATIONS

ETREMA Terfenol-D® stoichiometry will be customized to your specifications for optimum shape of the strain vs field curves, control of hysteresis or temperature range. Terfenol-D is $Tb_xDy_{1-x}Fe_y$. For most applications x=0.27-0.3, y=1.9-1.95. Prices may be higher for x>0.3.

Performance curves accompany samples in each shipment. Where possible, these are produced under conditions specified by the user.

Most rods are available in lengths up to 20 cm (8 inches). Maximum lengths are less for smallest and largest diameters. Machining to close tolerences for length, diameter, parallelism of drive faces, etc. is available at extra cost.

Appendix 3
Price List for Etrema Terfenol-D Products

MODIFIED BRIDGMAN RODS WITH AXIAL (LENGTHWISE) HOLES

Modified Bridgman cylindrical units can be supplied with axial holes. Prices depend on the outside and inside diameters of the finished units. Contact ETREMA Products, Inc. for wall thickness limitations and tolerance requirements.

MACHINED SHAPES

Small ETREMA Terfenol-D$_{TM}$ elements are custom machined to order. Consult us for discussions of optimum stoichiometry, drive axis orientaton and shape. Minimum order quantities apply, but research quantities may be available. Square elements 1.0mm thick per side and larger can be produced. Lengths depend upon thickness. Other cross sections may be possible, such as wafers and slabs.

HIGH FREQUENCY DRIVERS

For control of eddy current at high frequencies Terfenol-D may be sliced and relaminated. Prices will vary with lamination thickness, final length, diameter and quantity. Contact ETREMA Products, Inc. for further discussions and quotation to your requirements.

COMPLETE ACTUATORS/TRANSDUCERS

Terfenol-D driven actuators ready for use are available. Data sheets for each type and a detailed brochure are available.

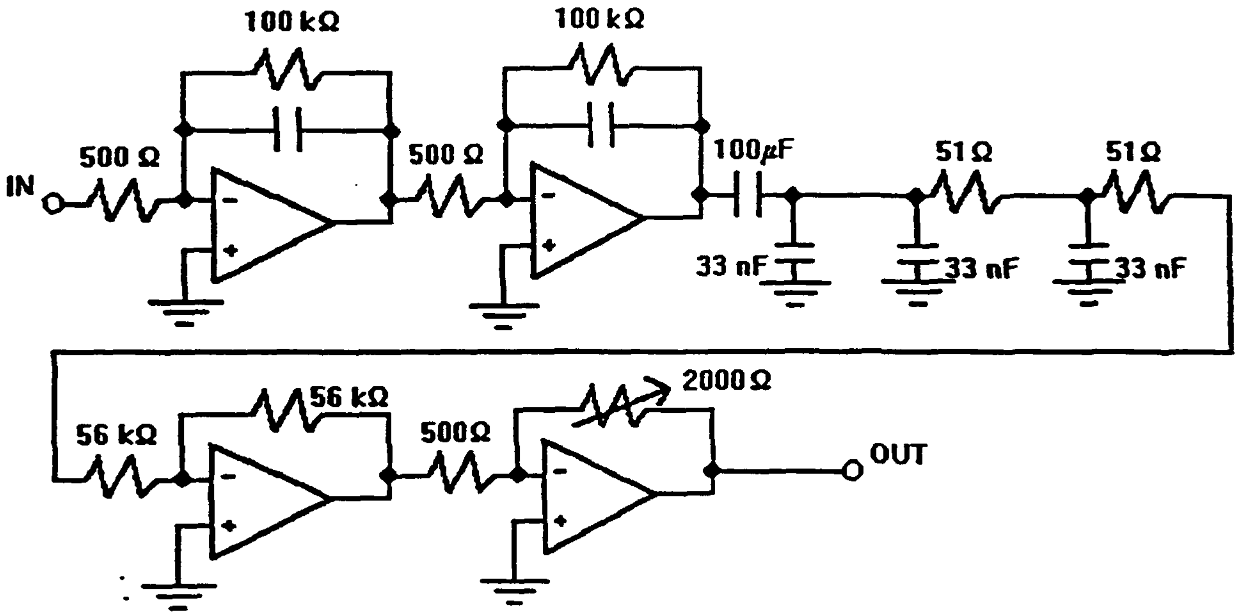
Model	Description	Price
LA-100	Linear Actuator	1,000.00
PN-50	Non Magnetic Bias Pusher	435.00
PM-50	Magnetically Biased Pusher	450.00
PHM-75	High Force Biased Pusher	825.00
PHM-110	High Force/Stroke Pusher	1,125.00
RA-101	Research Biased Actuator	500.00
RA-102	Research Nonbiased Actuator	500.00
RX-101	Research Biased Actuator (Anodized)	525.00

TERMS AND SHIPMENTS

Prices are F.O.B. Ames, IA. Shipping & Insurance charges will be prepaid and added to the invoice. Terms are Net 30 days after delivery. International shipments require an irrevocable confirmed letter of credit. Banking charges are the responsibility of the buyer. Payment is in U.S. dollars.

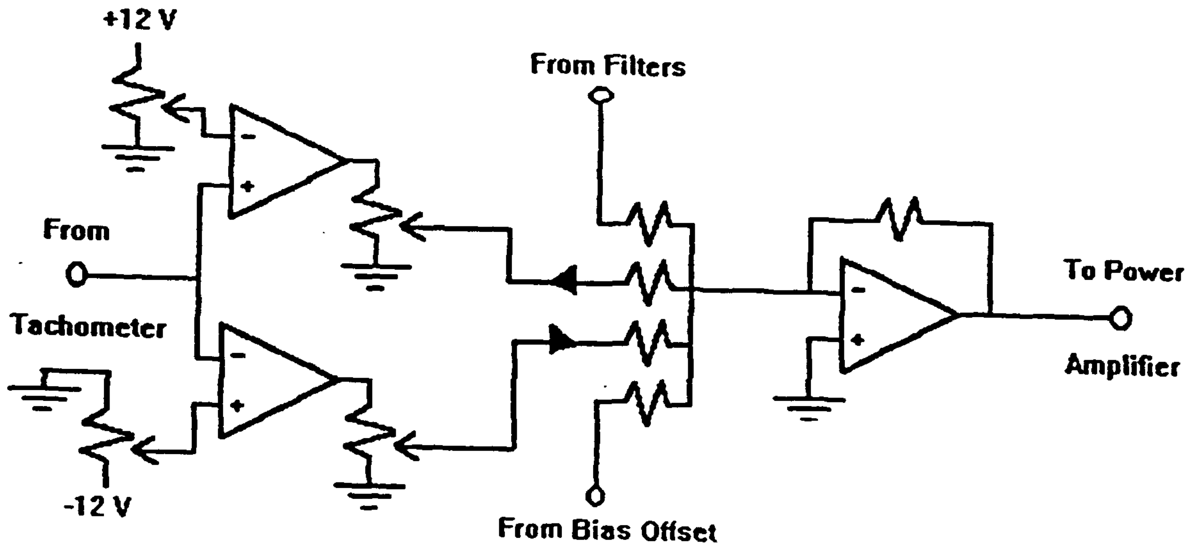
ETREMA Products, Inc. (515) 232-0820 or 1-800-327-7291

Appendix 4
Compensator Circuits



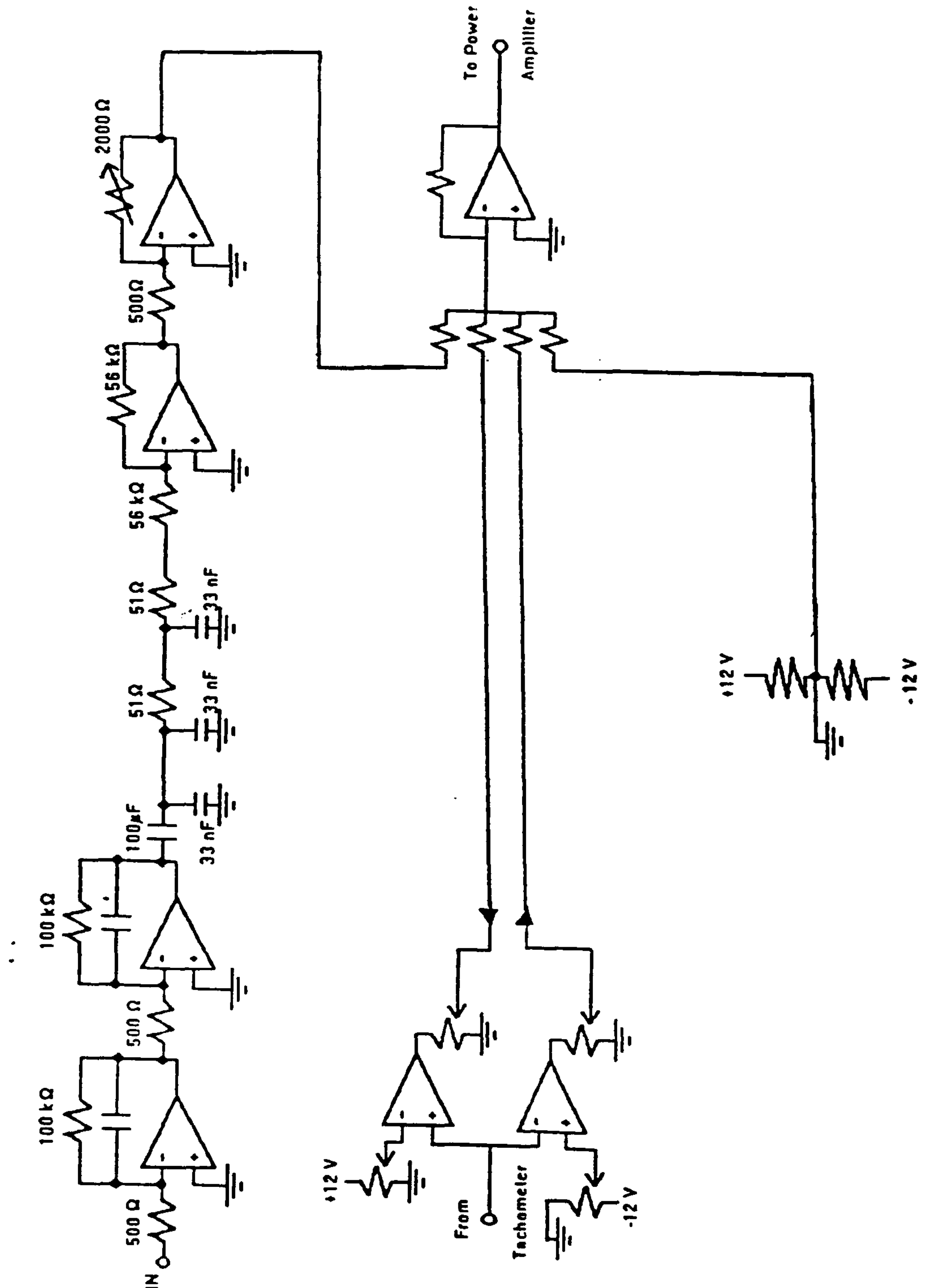
The compensator circuit for acceleration feedback. The signal is fed from the "Out" of this circuit diagram to the "From Filter" on the diagram of the frequency controlled biasing circuit.

Appendix 4
Compensator Circuits



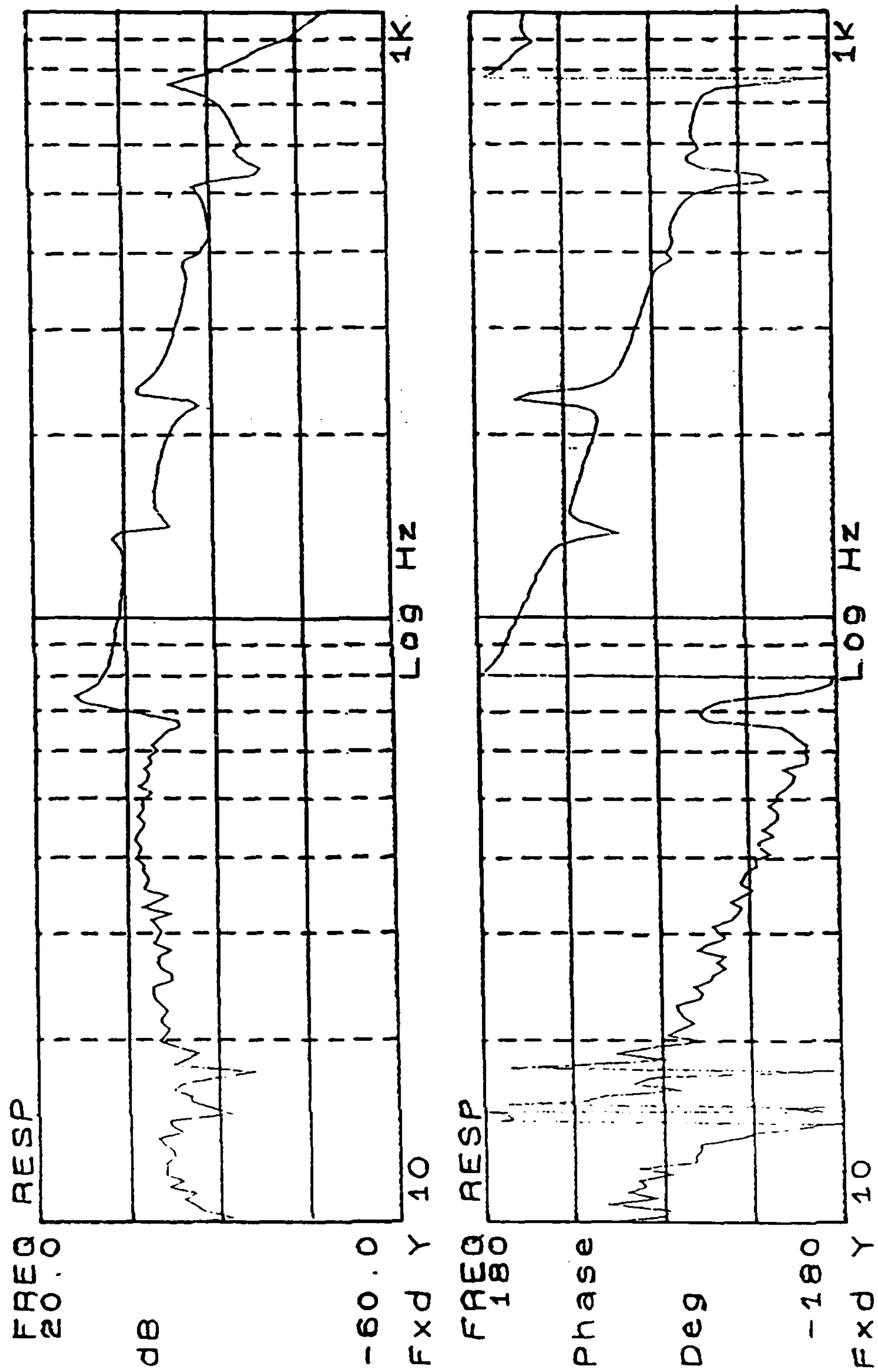
The frequency controlled biasing circuit. This circuit compares a voltage from the tachometer and activates or de-activates the pre-loading biasing. The output of the circuit is fed to the power amplifier for the actuators.

Appendix 4
Compensator Circuits



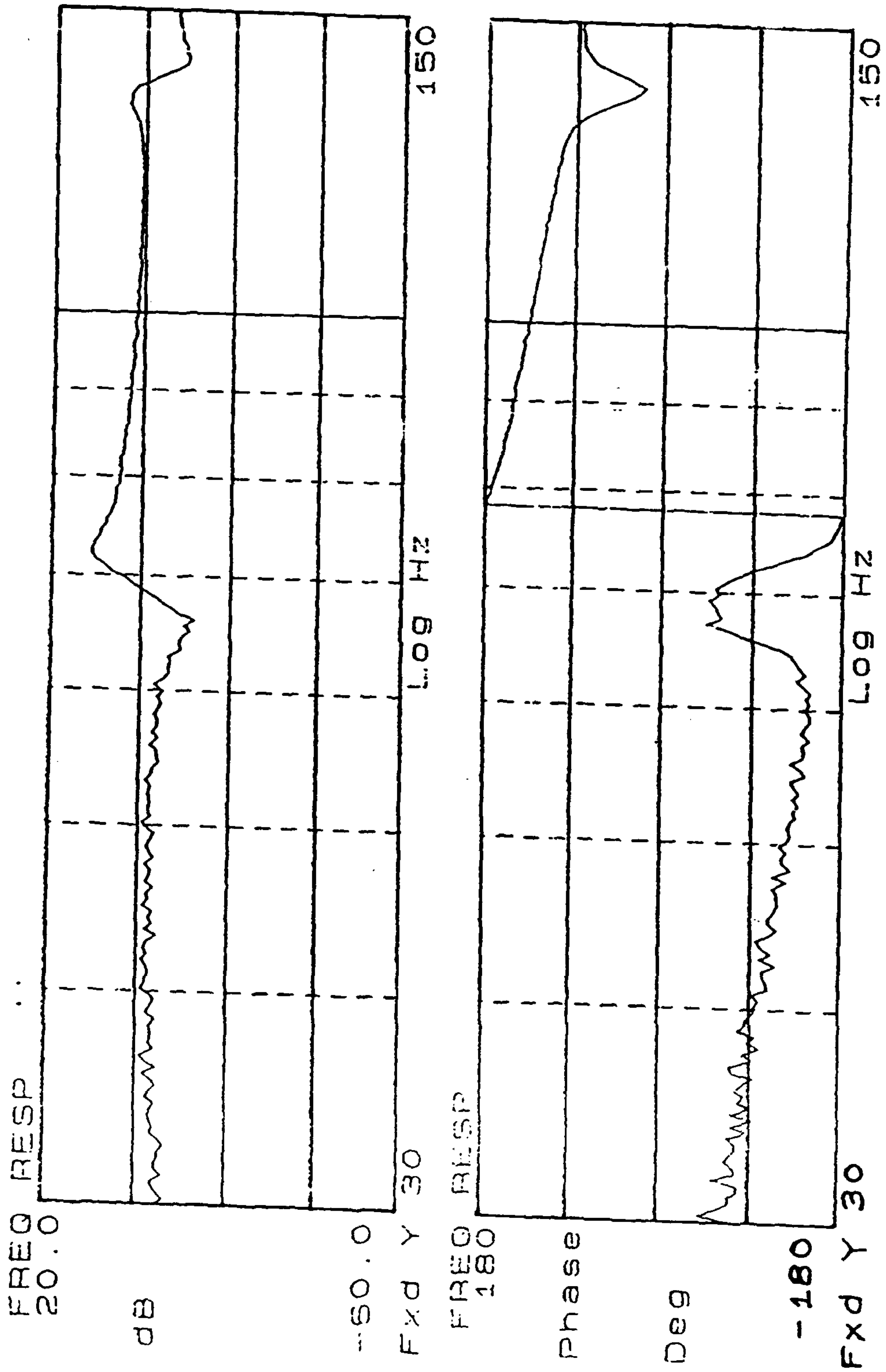
This is the entire control circuit, combining the acceleration feedback circuit and the frequency controlled biasing, as well as including a manual offset option.

Appendix 5
Bode Plots of the System and Filter



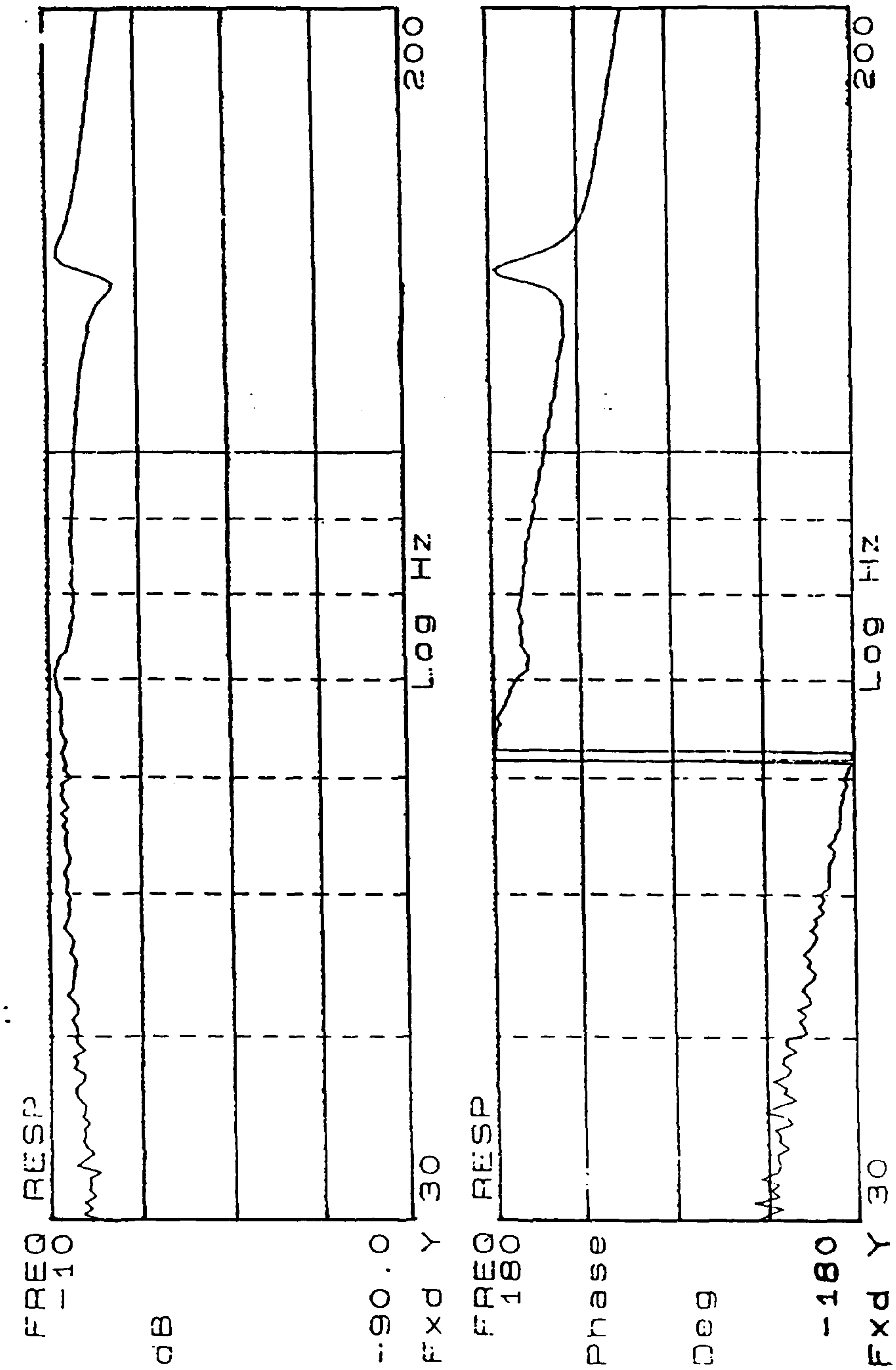
Bode Plot of the System and Filter from 10 to 1000 Hertz. This is the open loop transfer function of the upper platform, measured by an inverted accelerometer mounted to the underside of the platform.

Appendix 5
Bode Plots of the System and Filter



Bode Plot of the System and Filter from 30 to 150 Hertz for the upper platform as measured from the inverted accelerometer.

Appendix 5
Bode Plots of the System and Filter



Bode plot of the foundation from 30 to 200 Hertz, measured by an accelerometer mounted upright on the foundation.