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Disk Drive with Mode Canceling Actuator

FIELD OF THE INVENTION

This invention relates generally to mode canceling actuators used for canceling vibrational modes experienced by actuator-controlled arm assemblies employed in disk drives.

BACKGROUND OF THE INVENTION

Actuator-controlled arm assemblies employed in disk drives experience problems due to mechanical resonances. These resonances, or vibrational modes include the natural modes of the actuator system, e.g., a voice coil motor (VCM), and those of the arm assembly.

In a typical disk drive the arm assembly has a positioner arm equipped with a slider. The arm assembly is mounted about a pivot. The slider carries a transducer or read/write head and is located above the surface of a magnetic disk. The VCM is mounted opposite the slider and causes the arm assembly to rotate about the pivot as required to seek and track data on the disk. Specifically, the arm assembly moves the head to a desired data track in the disk and, once there, maintains the head's position over the track.

At high speeds and large track densities these seek and track operations are affected by the vibrational modes of the arm assembly and the actuator. Specifically, the vibrational modes limit the control loop gain of the actuator system, reduce the bandwidth of the actuator system, or both. This causes the head to experience excessive settling time after positioning, poor response to disturbances, poor tracking ability, or any combination of these.

General methods of achieving higher bandwidth include the use of high specific stiffness materials such as AlBC or Albumet for making the arms, employment of compound actuators and application of active and passive damping techniques.

The use of dual-stage actuation systems having a primary actuator, e.g., a VCM, for executing large movements and a secondary actuator, e.g., a piezoelectric element (PZT), for fine-tuning and tracking is well-known in the art. The small PZT miliactuator has higher vibrational modes than the VCM due to scaling. Descriptions of such systems are found in scientific and patent literature. Some representative references include Zhi-Min Y., et al. "Controller Design Criteria for the Dual-Stage Disk Actuator System", Proc. SPIE - International Society for Optical Engineering (USA), Vol. 2101, No. 1, 1993, pp. 305-8 and Guo, W., et al., "Dual Stage Actuators for High Density Rotating Memory Devices", IEEE Trans. Magn. (USA), Vol .34, No. 2, pt. 1 March 1998, pp. 450-5.

Prior art systems have also attempted to ensure stable operation of actuator systems by stabilizing the control loop. This has been done by inserting gain stabilizing filters such as electronic notch filters in the control loop path. These filters are placed in the downstream portion of the control loop to filter out the signal information within the band reject

frequency range of the notch and thus help minimize excitation of these vibrational modes.

Filtering techniques of this type have several drawbacks. First, rather than compensating the vibrations of the actuated arm assembly directly they rely on correcting the driving signals sent to the actuator to achieve compensation. Second, these techniques depend on the ability of the designer to accurately predict the frequency of the vibrational modes. This becomes increasingly more difficult when operating in high accuracy regimes. Third, compensation of the gain loop is not capable of eliminating the amplitude and phase effects of the vibrational mode, i.e., the effects of the mode can not be completely canceled.

Another technique for damping vibrational modes of a servo control system was presented by Masahito Kobayashi et al. in *"MR-46 Carriage Acceleration Feedback Multi-Sensing Controller for Sector Servo Systems,"* at the International Conference on Micromechtronics for Information and Precision Equipment, Tokyo, July 20-23, 1997. This proposed multi-sensing control system uses accelerometers to generate acceleration feedback. An acceleration feedback controller receives the feedback signals and compensates the servo to damp the mechanical resonance modes.

Although Kobayashi's technique has been demonstrated, it can not be efficiently implemented without the use of notch filters. Furthermore, designing the feedback controller requires the designer to model the very complicated transfer function $H_d(s)$ of the servo-controlled system. This is computationally challenging and requires a considerable amount of processing time. Also, since the poles and zeros of the compensator used in the feedback controller can not be predetermined, it is not

possible to guarantee the existence of a stable compensator. Most importantly, however, Kobayashi's technique does not achieve cancellation of the vibrational mode.

The prior art also teaches gain stabilization through low-pass filtering in the control loop. According to this method the components of the control signal having the resonance frequency are effectively prevented from exciting the vibrational modes of the actuator structure. This helps ensure system stability, but it also increases the phase shift at frequencies in the vicinity of the servo loop's unity gain crossing, thereby reducing the bandwidth of the servo system.

In U.S. Patent no. 5,459,383 Sidman et al. teach a feedback loop using a motion sensor located in the servo system at or near the point of control. The sensor is referred to as collocated because it is at or near the point of control. During operation the sensor detects both the rigid body motion and deformation of the actuator. The signal component from the rigid body motion is always much larger than that due to deformation. The collocated feedback loop operates in conjunction with the ordinary feedback loop and has the effect of making the servo system perform as if the mechanical structure of the system had a much higher mechanical damping than it actually possesses.

Although Sidman's system does improve positioning control it relies on gain compensation and does not actually cancel any vibrational modes. Furthermore, the signal derived from the sensor includes the large rigid body component, which is also processed by the feedback loop and affects undesirably the rigid body motion properties of the actuator.

The above-mentioned problems have prevented the development of a servo system capable of entirely eliminating the effects of

vibrational modes, including modes at the lowest frequencies which severely limit the bandwidth of the servo system. Moreover, the solutions relying on control signal compensation or passive damping introduce considerable complications into the control loop.

A high-performance disk drive should have, in addition to the ability to cancel at least one vibrational mode, e.g., the Butterfly mode, a low-mass arm assembly and its rigid body motion should remain unaffected.

OBJECTS AND ADVANTAGES OF THE INVENTION

Accordingly, it is a primary object of the present invention to provide a vibrational mode canceling mechanism for stabilizing a servo-controlled actuator system and overcome the disadvantages of the prior art. Specifically, the control mechanism of the invention is designed to cancel the amplitude and phase effects of at least one vibrational mode while minimizing the mass of the arm assembly and preserving its rigid body motion characteristics.

It is another object of the invention to provide for mode cancellation of major modes including the butterfly mode.

Still another object of the invention is to achieve mode cancellation with a low-cost and low-mass element, and to thus permit one to design efficient devices with a higher number of tracks per inch (TPI). In particular, it is an object of the invention to employ a mode-canceling actuator such as a piezoelectric (PZT) element.

The above objects and advantages, as well as numerous improvements attained by the system and method of the invention are pointed out below.

SUMMARY

These objects and advantages are attained in a disk drive in which an arm assembly is equipped with a primary actuator and a mode-canceling actuator. The primary actuator, preferably a voice coil motor (VCM), is mounted on the assembly at a first location while the mode-canceling actuator, preferably a piezoelectric (PZT) actuator, is mounted at a second location. The second location is selected in such a way that the strain in the mode-canceling actuator is in phase with sway deformations of the primary actuator produced by the vibrational modes. Specifically, the strain experienced by the mode-canceling actuator has to be in phase with sway deformations in the plane of the arm assembly caused by the major modes. A primary driver is provided to control the primary actuator such that it exerts a primary force, $\vec{f}_{pm}$, on the arm assembly. Meanwhile, a secondary driver ensures that the mode-canceling actuator applies a mode-canceling force, $\vec{f}_{mc}$, to the arm assembly such that one or more vibrational modes, e.g., the lowest frequency modes, are canceled.

The primary and secondary drives are conveniently controlled by a control circuit or device which sets the ratio r of the primary force, $\vec{f}_{pm}$, to the mode-canceling force, $\vec{f}_{mc}$ at a certain value. Specifically, the value of ratio r and the location of the mode-canceling actuator are set such that none of the vibrational modes produced by the mode-canceling force exceeds any of the corresponding vibrational modes caused by the primary actuator by more than 6 dB.

Preferably, the mode-canceling actuator eliminates at least the Butterfly mode. In some disk drives, elimination of one or more major modes other than the Butterfly mode is also desirable.

DESCRIPTION OF THE FIGURES

Fig. 1 is an isometric view of a disk drive according to the invention.

Fig. 2 is a plan schematic view of a portion of the arm assembly of the disk drive shown in Fig. 1.

Fig. 3 is a plan schematic view illustrating the operation of the arm assembly of Fig. 1.

Fig. 4 is a block diagram showing a control loop for use in a disk drive with mode-cancellation according to the invention.

Fig. 5 is an isometric view of a multiple positioner arm assembly equipped with mode-canceling actuators according to the invention.

Fig. 6 is a top plan view of an arm of an IBM Sailfin drive equipped with a PZT mode-canceling actuator.

Fig. 7 is a graph illustrating the gain cancellation performance of the mode-canceling actuator mounted on the arm assembly of Fig. 6.

Fig. 8 is a graph illustrating the phase cancellation performance of the mode-canceling actuator mounted in the Sailfin drive.

Fig. 9A includes two graphs showing the phase and gain of the Sailfin drive arm assembly without a mode-canceling actuator.

Fig. 9B includes two graphs showing separately the phase and gain of the Sailfin drive arm assembly with a mode-canceling actuator according to the invention.

DETAILED DESCRIPTION

Fig. 1 illustrates a servo-controlled actuator system **10** deployed in a hard disk drive **11**. Actuator system **10** has an arm assembly **12** equipped with a transducer or controlled element **14**, in this case a read/write head. Arm assembly **12** has a

positioner arm **16**. Entire assembly **12** is mounted on a pivot or hub **30**. Head **14** is positioned above a disk **22** with concentric data tracks **24** and accesses various tracks **24** during operation by pivoting or rotating around hub **30**. This general construction of arm assembly **12** is well-known in the art of hard disk drives.

A primary actuator **26**, in this instance a voice coil motor (VCM), is mounted in a cradle or coil support **28** opposite head **14**. VCM **26** moves arm assembly **12** by causing it to rotate about hub **30**. Consequently, actuator **26** is also known as a rotary actuator. The amount of rotation is controlled by a primary drive signal **50** (see Fig. 3) in the form of a current passed through coil **26**.

As better illustrated in the partial view of Fig. 2, a mode-canceling actuator **32** is attached to arm assembly **12** near cradle **28**. It is preferable that mode-canceling actuator be a light-weight device, e.g., a piezoelectric element (PZT). The use of a heavy mode-canceling actuator, e.g., a device weighing as much as or more than VCM **26** should be avoided, because it would affect the rigid body motion of arm assembly **12**. Also, a heavy mode-canceling actuator will undesirably increase the inertia of arm assembly **12**. Specifically, the weight of PZT **32** is typically about 1 mgr. In comparison, the weight of VCM **26** usually ranges from 4-6 grams.

The exact position of PZT **32** is selected such that it is sensitive to the in-plane sway deformations experienced by VCM **26**. Referring back to Fig. 2, the dashed lines indicate the in-pane sway deformation of VCM **26** resulting from the vibrations of arm assembly **12**. When arm assembly **12** experiences vibrational modes during operation, PZT **32** experiences strains resulting

from these modes. This is because the in-plane sway deformations of VCM **26** translate into deformations of cradle **28**.

In most instances, an empirical approach can be used to obtain a transfer function, T, describing the vibrational modes of arm assembly **12**. For purposes of illustration, the graphs of Fig. 9A show the transfer function T for a typical arm assembly used in a Sailfin type disk drive. The graphs show the gain and phase effects of the vibrational modes separately. The Sailfin drive has ten separate arm assemblies. The most prominent vibrational modes are the "butterfly mode" centered at about 3,500 Hz, the S mode located at about 4,200 Hz, the end-arm mode at about 4,800 Hz and the second butterfly mode at about 5,800 Hz. The gain of the "butterfly mode" is chiefly responsible for the limited operation bandwidth of a typical arm assembly. It is therefore crucial to cancel both the gain (amplitude) and the phase of these major vibrational modes, or, at least of the butterfly mode.

As is true of any mechanical system, when arm assembly **12** vibrates, the vibration can be decomposed into several vibrational modes which have their own unique vibration shapes called "mode shapes". The mode shape $\overline{\varphi_i}$ of the i-th mode can be expressed as:

$$\overline{\varphi_i} = \overline{\varphi_i}(x,y,z),$$

which is a function of location (x,y,z). The net in-plane sway deformation $\Delta_{\text{in-plane}}$ of coil **26** is then described by:

$$\Delta_{\text{in-plane}} = \int_{coil} \overline{\varphi_i}(x,y,z) \cdot \overline{F}(x,y,z)\,dxdydz\,,$$

where $\overline{F}$ is the force applied on coil **26**. Thus, the strain, ε_i, experienced by PZT **32** due to the i-th vibrational mode is also a function of the mode shape $\overline{\varphi_i}$ and is represented by:

$$\varepsilon_i = \varepsilon_i\left(\overline{\varphi_i}\right).$$

When the sign of the sway deformation $\Delta_{\text{in-plane}}$ is the same as the sign of strain ε_i at the location of PZT **32**, then PZT **32** is properly mounted with respect to the i-th mode. In this position the strain experienced by PZT **32** is in phase with the i-th mode.

Proper placement of PZT **32** is visualized in Fig. 2. For the sake of simplicity, only three test positions A, B, and C of PZT **32** are shown. Also, although the transfer function T consists of a multiplicity of modes only three major modes are illustrated. These may correspond to the first butterfly mode and the first and second end arm modes and directly limit servo performance and bandwidth because of their high gains and phase shifts. The remaining modes are minor modes, which have low gain and phase shifts and thus lie outside the range of interest.

In each position, PZT **32** effectively partitions all vibrational modes into two groups; one group of modes is in phase with the strain in PZT **32** and the other is out of phase. The final position of sensor **32** has to be such that all major modes i, j and k are in phase with its strain. The phase relationship with minor and negligible modes is not important and these may all

fall into the out of phase partition. In fact, minor modes generally remain stable due to original material damping.

The below table shows the phase relationship between strain experienced by PZT **32** and major modes i, j, and k with respect to sway deformation $\Delta_{\text{in-plane}}$ of VCM **26**. At location A the strain in PZT **32** is in phase with modes i, j, and k. At location B the strain in PZT **32** is out of phase with modes i and j, and at location C it is out of phase with mode k. Hence, the proper final location for permanently mounting PZT **32** is A.

	Mode i	*Mode j*	*Mode k*	...
sign of $\Delta_{\text{in-plane}}$	+	-	+	...
sign of ε_i for PZT location A	+	-	+	...
sign of ε_i for PZT location B	-	+	+	...
sign of ε_i for PZT location C	+	-	-	...

Preferably, the ultimate position of PZT **32** is fine-tuned even more precisely. Specifically, the final position of PZT **32** is selected such that the gains of major modes excited by PZT **32** in its mode-canceling operation through application of mode-canceling force, $\vec{f}_{mc}$, do not exceed a certain gain level. More specifically, the gain of any major mode excited by mode-canceling force, $\vec{f}_{mc}$, has to be no more than 6 dB larger than the gain of that mode due to the primary force, $\vec{f}_{pm}$, applied by VCM **26**. Under this condition the phase of the major mode affected will change, but it will be guaranteed that the gain will not be bigger than the original gain.

In practice, the proper final position for PZT **32** can be found by empirical or analytic methods such as computer-implemented algorithms. When the mechanical structure of arm assembly **12** is specially designed, e.g., when arm assembly **12** belongs to a rotary actuator system used in hard disk drives, the general vicinity of the potential final position is easier to predict and hence the empirical approach is viable. For example, in the IBM Sailfin actuator the right side of the second to bottom arm is the proper general area to search for the final PZT position.

The analytical approach involves modeling arm assembly **12**. This is done, for example, with the aid of finite element analysis software. The analysis pinpoints final position A on arm assembly **12**, specifically on cradle **28**, where sensor strain is in phase with the major modes. This is the position at which PZT **32** is permanently attached. A fine-fine tuning of this position may be required due to an additional requirement as described below.

The schematic diagram of Fig. 3 shows the necessary control elements for operating arm assembly **12** of disk drive **11**. VCM **26** is a current-driven device and is connected to a primary driver **54** which is a current amplifier. PZT **32** is a voltage-driven device and is connected to a secondary driver **52** which is a voltage amplifier **52**. Both amplifiers **52**, **54** are controlled by a control circuit **56**.

During operation, control circuit **56** determines from the vibrational data of arm assembly **12** and the position of head **14** proper primary drive signal **50** or drive current for VCM **26**. Specifically, drive current **50** has to be such that it causes VCM **26** to exert a primary force, $\vec{f}_{pm}$, on arm assembly **12**. At the same time, the mode-canceling drive signal **53** or voltage delivered by amplifier **52** has to be such as to cause PZT **32** to

apply a mode-canceling force, $\vec{f}_{mc}$, to assembly 12 such as to cancel out the desired vibrational mode, e.g., the butterfly mode. Cancellation of the mode, or elimination of the gain and phase associated with the mode, will take place when the ratio r of primary force, $\vec{f}_{pm}$, applied by VCM 26 to the mode-canceling force, $\vec{f}_{mc}$, applied by PZT 32 is set and maintained at a specific value.

The calculation of ratio r is based on the transfer function T, which can be expressed as:

$$T = \frac{R_k \psi_i^k \psi_j^i f_i}{m_i\left(\lambda_i - \omega^2\right)}$$

where y_i, m_i, l_i are the displacement vector, mass, and eigenvalue of the i-th vibrational mode. The vector R_k describes the measured point and f_j are the components of the driving force vector. For economy of notation, the Einstein convention is used and thus the repeated indices are summed over. In vector notation, this equation can be rewritten as:

$$T = \frac{\left(\vec{\psi}^i \cdot \vec{R}\right)\left(\vec{\psi}^i \cdot \vec{f}\right)}{m_i\left(\lambda_i - \omega\right)}$$

where it is convenient to remember this expression as the product of two inner products. To eliminate from the transfer function a major mode we produce a modified driving force, f_k, such that the inner product vanishes. In other words:

$$f_k \psi_1^k = 0 ,$$

or in vector notation:

$$\vec{\psi}_k \cdot \vec{f} = 0 .$$

Of course, this modified force must still obey the relation:

$$\vec{\psi}_0 \cdot \vec{f} \neq 0$$

where ψ_0 is the rigid body mode.

Referring now back to Fig. 3, mode cancellation of, e.g., the butterfly mode is achieved by the modified driving force f_k which is composed of the primary force, $\vec{f}_{pm}$, and the mode-canceling force, $\vec{f}_{mc}$. These two component forces are adjusted such that the ratio of the force applied by VCM **26** and PZT **32** on arm assembly **12** is given by:

$$\vec{\psi}_1 \cdot \left(\vec{f}_{pm} + \vec{f}_{mc}\right) = 0 .$$

In this case major mode ψ_1 is the butterfly mode. Of course, any other major mode could be selected for cancellation or a number of modes could be canceled at the same time. For this purpose multiple mode-canceling actuators like PZT **32** can be used in concert.

The ratio of the primary and mode-canceling forces is computed by control circuit **56** using standard logic circuitry. After performing the computation circuit **56** appropriately adjusts the gain of amplifiers **54**, **52** to apply the computed primary and mode-canceling forces via VCM **26** and PZT **32**.

In the present embodiment circuit **56** bases its adjustments of amplifiers **54**, **52** on the ratio r of primary force to mode-canceling force expressed as:

$$r = \frac{\left(\vec{\psi}_1 \cdot \vec{f}_{pm}\right)/\left(gK_f\right)}{\left(\vec{\psi}_1 \cdot \vec{f}_{mc}\right)/h}$$

where g is the voltage to current gain of current amplifier **54**, K_f is the torque constant of arm assembly **12** with actuators and any other devices, and h is the gain of voltage amplifier **52**. In the case where circuit **56** produces output voltages to regulate amplifiers **52**, **54**, the ratio can be expressed in terms of driving voltages V_{pm} and V_{mc} as:

$$V_{mc} = \frac{V_{pm}}{r}.$$

Generally, for a particular design of arm assembly **12** and placement of PZT **32**, this ratio r is well-defined and may be determined through simulation or experiment. Of course, if this ratio does not satisfy the above equation then this constitutes evidence that other modes are affecting assembly **12**.

In addition to maintaining proper force ratio r the actual magnitudes of the primary and mode-canceling forces have to be controlled as well. Once again, this is done in order that the gains of major modes excited by PZT **32** in its mode-canceling operation do not exceed by more than 6 dB the gain levels caused by VCM **26**. This is done by fine-tuning the location of mode-canceling actuator **32** while satisfying the above-explained in-phase condition.

The proper gain relationship is visualized in Fig. 7 where graph **202** indicates the result of mode-canceling force only and graph

204 indicates the result of primary force only for mode-cancellation according to the invention deployed in an IBM Sailfin drive. The gains of the end arm mode due to mode-canceling force are within a $\Delta \geq 6$ dB of the end arm mode gains caused by the primary force. The result of both forces acting together is indicated by graph **206**. Clearly, the butterfly mode is canceled under these circumstances and the remaining major modes are not aggravated. The phase effects are shown in the graphs of Fig. 8. The same reference numbers are used for the arm assembly's response to primary force only, mode-canceling force only, and both forces being applied together.

In a particular application of the invention, mode cancellation may be turned off while head **14** is seeking a new track **24**. That is because higher voltages are usually required for seeking. During that time circuit **56** can be switched off and then resume operation once a particular track **24** is selected to be followed.

The above-described system uses an open-loop type control for driving VCM **26** and PZT **32**. Open-loop control assumes *a priori* knowledge of the frequencies of the modes to be canceled. Small changes in ratio r due to temperature effects can be mitigated by adjusting or re-calibrating control circuit **56**. However, in a high-performance system small and continuous variations in ratio r will adversely affect the efficiency of mode-cancellation. Hence, in a preferred implementation of mode-canceling a closed loop control system **100** is used as shown in the block diagram of Fig. 4.

Control system **100** is a general closed-loop system. A person of average skill in the art will appreciate that in practice this type of system may be modified to best cooperate with any given disk drive. System **100** has a primary driver **102** for driving a VCM **108** and a secondary driver **104** for driving a PZT **110**. The

logic of system **100** is contained in drivers **102**, **104** rather than in a separate circuit. Secondary driver **104** is connected to the output of primary driver **102** to receive a primary drive signal **106** which is also delivered to VCM **108**. Secondary driver **104** uses signal **106** in generating mode-canceling drive signal **112** delivered to PZT **110** to ensure the proper force ratio r, as described above.

In particular, the effects of VCM **108** and PZT **110** applying their respective forces on a schematically indicated arm assembly **114** are measured by disk drive mechanics **116**. Preferably, mechanics **116** will simply include the read/write head which derives a position error signal (PES) **120** from pre-recorded servo information on the disk. PES **120** informs primary driver **102** by how much the read/write head has to be moved to be in the correct position over a selected data track. PES **120** can be further processed to derive a signal **118** indicating the vibrational response of assembly **114**. Alternatively, PES **120** can be used directly as signal **118**. In another embodiment, mechanics **116** may include transducers in addition to the read/write head, e.g., accelerometers. Such transducers are typically mounted close to the read/write head and are capable of delivering a measured signal **118** indicating the actual vibrational response of assembly **114**. The read/write head can still deliver PES **120** as necessary. Mechanics **116** of this type are well-known in the art.

System **100** is wired to deliver signal **118** to secondary driver **104** and PES **120** to primary driver **102**. (Alternatively, a head position signal derived from markings disposed on the disk can be used instead of PES **120**.) Signal **118** is used to determine or at least estimate the performance of system **100** or the accuracy of the model. This determination is performed with the aid of known methods, such as adaptive control or on-line/real-time

system identification. More detailed information about these methods for determining system performance is contained in the textbook Adaptive Control, 2nd edition, by Karl J. Astrom and Bjorn Wittenmark published by Addison-Wesley.

Signal **118** is used to adjust the parameters of secondary driver **104** and cause it to issue correspondingly adjusted mode-canceling signal **112**. In accordance with known principles of control engineering, signal **118** may or may not be filtered, to adjust the parameters of secondary driver **104**. The control techniques which can be applied fall into two broad categories. The first category includes techniques that adjust the parameters based on the performance of the system **100**, such as direct adaptive control, learning control with fuzzy logic and/or neural networks, auto-tuning, and the like. The second category includes techniques that adjust the parameters based on the accuracy of the model, such as indirect adaptive control, on-line system identification, self-tuning, and the like. In essence, all of these techniques achieve the same objective by using signal **118** to directly or indirectly (adaptively) adjust the feedback control, i.e., to determine signal **112**. This adjustment is performed on the time scale on the order of changes taking place in the disk drive system. For example, the adjustment time scale matches the time scale on which the torque constant F_t or characteristics of PZT **110** vary. Thus, the adjustment ignores disturbances and noise, which typically have significantly shorter time scales, but takes into account known parameters which contribute to the variation of ratio r thus guaranteeing improved mode-canceling performance.

The mode-canceling system of the invention provides an effective mechanism for canceling the gain and phase of any given vibrational mode and stabilizing a servo-controlled actuator system. Furthermore, the mechanism of the invention minimizes

the mass of the arm assembly and preserves its rigid body motion characteristics because of the low-weight of the PZT element. The low-cost of PZT elements further opens the possibility to design efficient and low-cost disk drives with a higher number of tracks per inch (TPI).

An adaptation of the mode-canceling system to a multiple positioner arm assembly **150** is shown in the isometric view of Fig. 5. In this embodiment a single primary actuator **152** in the form of a VCM lodged in a cradle **158** applies the primary force to assembly **150**. Six positioner arms **154** equipped with read/write heads **156** are thus rotated about a hub **160**. One mode-canceling actuator **162** in the form of a PZT element is mounted on each positioner arm **154**. The position of each PZT **162** is selected according to the placement criteria discussed above. During mode-canceling operation each PZT **162** contributes a portion of the total mode-canceling force applied to assembly **150**.

Fig. 6 illustrates in top plan view a mode-canceling PZT **170** mounted on a representative arm assembly **172** belonging to an IBM Sailfin drive. The mode-cancellation performance data in the graphs of Figs. 7 and 8 are based on arm assembly **172** in which PZT **170** applies a mode-canceling force, as described above, to cancel the butterfly mode centered at about 3,500 Hz. Specifically, graph **206** in Fig. 7 illustrates the gain reduction for the butterfly mode achieved by applying the mode-canceling force with PZT **170**. Corresponding graph **206** in Fig. 8 illustrates the phase cancellation achieved for the butterfly mode.

For a clearer evaluation, Fig. 9B shows the performance of the Sailfin drive using a PZT mode-canceling actuator for canceling the gain and phase of the butterfly mode in two separate graphs.

The top graph shows the phase response and the bottom graph the gain response. For comparison purposes, Fig. 9A shows the performance of the same drive without a mode-canceling actuator.

The system and method of the invention are simple and permit the user to achieve mode-canceling of any major mode affecting a disk drive. Of course, it will be clear to one skilled in the art that the above embodiment may be altered in many ways without departing from the scope of the invention. Accordingly, the scope of the invention should be determined by the following claims and their legal equivalents.

CLAIMS

What is claimed is:

1. A disk drive comprising:
   a) an arm assembly experiencing predetermined vibrational modes;
   b) a primary actuator mounted on said arm assembly at a first location;
   c) a mode-canceling actuator mounted on said arm assembly at a second location at which the strain in said mode-canceling actuator is in phase with sway deformations of said primary actuator produced by said predetermined vibrational modes;
   d) a primary driver for causing said primary actuator to exert a primary force, $\vec{f}_{pm}$, on said arm assembly;
   e) a secondary driver for causing said mode-canceling actuator to exert a mode-canceling force, $\vec{f}_{mc}$, on said arm assembly such that one of said predetermined vibrational modes is canceled.

2. The disk drive of claim 1 further comprising a control means for setting a ratio r of said primary force $\vec{f}_{pm}$ to said mode-canceling force $\vec{f}_{mc}$.

3. The disk drive of claim 1 wherein said second location is selected such that none of said predetermined vibrational modes produced by said mode-canceling force exceeds any corresponding mode of said predetermined vibrational modes produces by said primary force by more than 6 dB.

4. The disk drive of claim 1 wherein said one vibrational mode is the butterfly mode.

* C L M S *

5. The disk drive of claim 1 wherein said predetermined vibrational modes are major vibrational modes.

6. The disk drive of claim 1 wherein said primary actuator is a voice coil motor.

7. The disk drive of claim 6 wherein said mode-canceling actuator is a piezoelectric actuator.

8. The disk drive of claim 1 wherein said mode-canceling actuator is a piezoelectric actuator.

9. The disk drive of claim 8 wherein said primary actuator is a voice coil motor.

10. The disk drive of claim 1 wherein said sway deformation is contained in a plane of said arm assembly.

11. A disk drive comprising:

a) an arm assembly experiencing vibrational modes;

b) a primary actuator mounted on said arm assembly at a first location;

c) a mode-canceling actuator mounted on said arm assembly at a second location at which the strain in said mode-canceling actuator is in phase with sway deformations of said primary actuator produced by said vibrational modes;

d) a primary driver for causing said primary actuator to exert a primary force, $\vec{f}_{pm}$, on said arm assembly;

e) a secondary driver for causing said mode-canceling actuator to exert a mode-canceling force, $\vec{f}_{mc}$, on said arm assembly such that at least two of said vibrational modes are canceled.

* C L M S *

12. The disk drive of claim 11 further comprising a control means for setting a ratio r of said primary force $\vec{f}_{pm}$ to said mode-canceling force $\vec{f}_{mc}$.

13. The disk drive of claim 12 wherein said second location is selected such that none of said vibrational modes produced by said mode-canceling force exceeds any corresponding mode of said vibrational modes produces by said primary force by more than 6 dB.

14. The disk drive of claim 11 wherein said at least two vibrational modes are major vibrational modes.

15. The disk drive of claim 11 wherein said primary actuator is a voice coil motor.

16. The disk drive of claim 15 wherein said mode-canceling actuator is a piezoelectric actuator.

17. The disk drive of claim 11 wherein said mode-canceling actuator is a piezoelectric actuator.

18. The disk drive of claim 17 wherein said primary actuator is a voice coil motor.

19. The disk drive of claim 11 wherein said sway deformation is contained in a plane of said arm assembly.

20. A system for operating a disk drive having an arm assembly experiencing vibrational modes, said system comprising:

a) a primary actuator mounted on said arm assembly at a first location;

b) a mode-canceling actuator mounted on said arm assembly at a second location at which the strain in said mode-

* C L M S *

canceling actuator is in phase with sway deformations of said primary actuator produced by said vibrational modes;

c) a primary driver for causing said primary actuator to exert a primary force, $\vec{f}_{pm}$, on said arm assembly;

d) a secondary driver for causing said mode-canceling actuator to exert a mode-canceling force, $\vec{f}_{mc}$, on said arm assembly such that at least two of said vibrational modes are canceled.

* C L M S *

DISK DRIVE WITH MODE CANCELING ACTUATOR

A disk drive (11) having an arm assembly (12) equipped with a primary actuator (26) such as a VCM and a mode-canceling actuator (32) such as a PZT mounted at a location selected in such a way that the strain in the mode-canceling actuator (32) is in phase with sway deformations of the primary actuator (26) produced by the vibrational modes of the arm assembly (12). The primary actuator (26) is driven to exert a primary force, $\vec{f}_{mc}$, on the arm assembly (12) while the secondary drive (104) ensures that the mode-canceling actuator (32) applies a mode-canceling force, $\vec{f}_{pm}$, on arm assembly (12) such that the gain and phase of one or more vibrational modes are canceled.

Fig 4

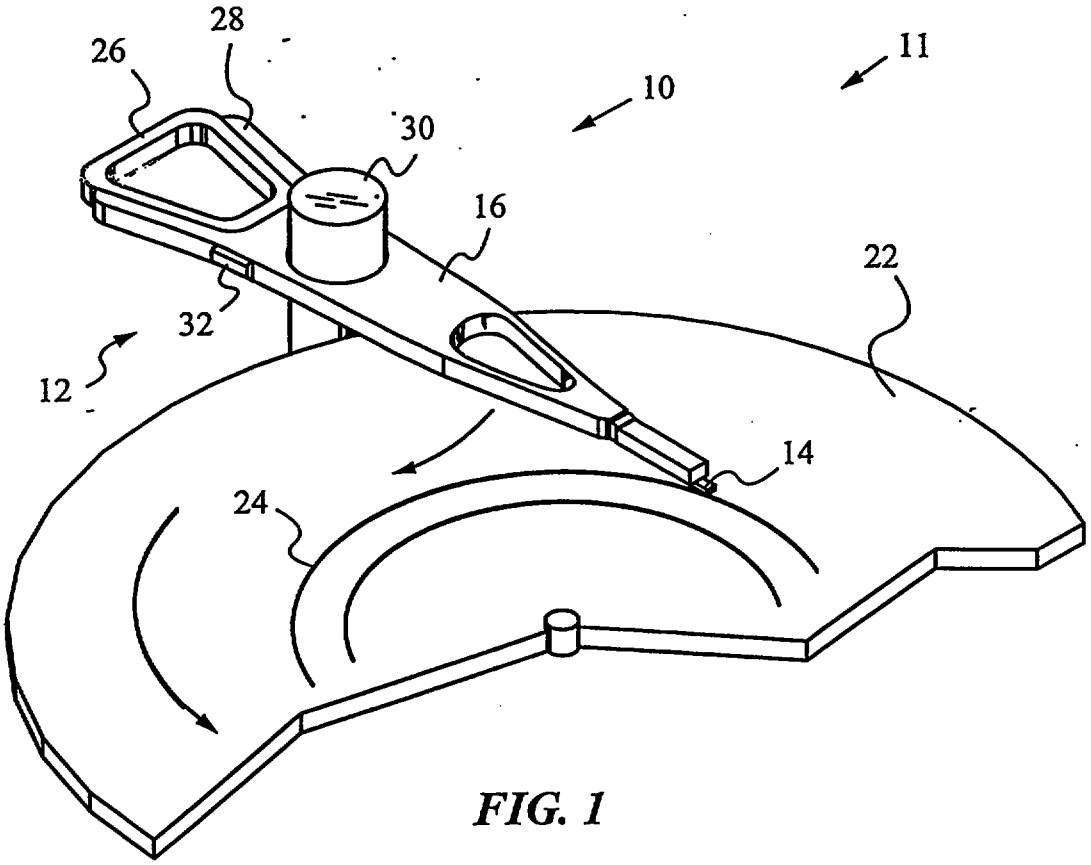


FIG. 1

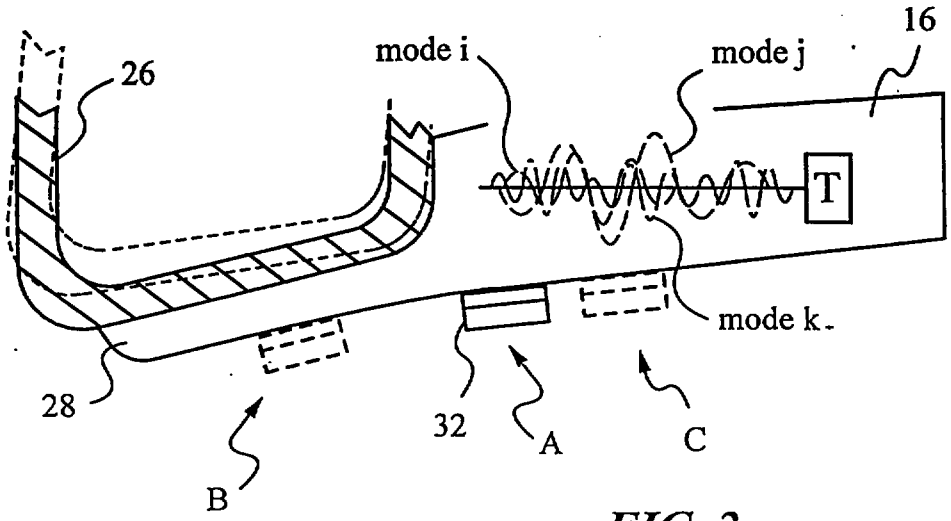


FIG. 2

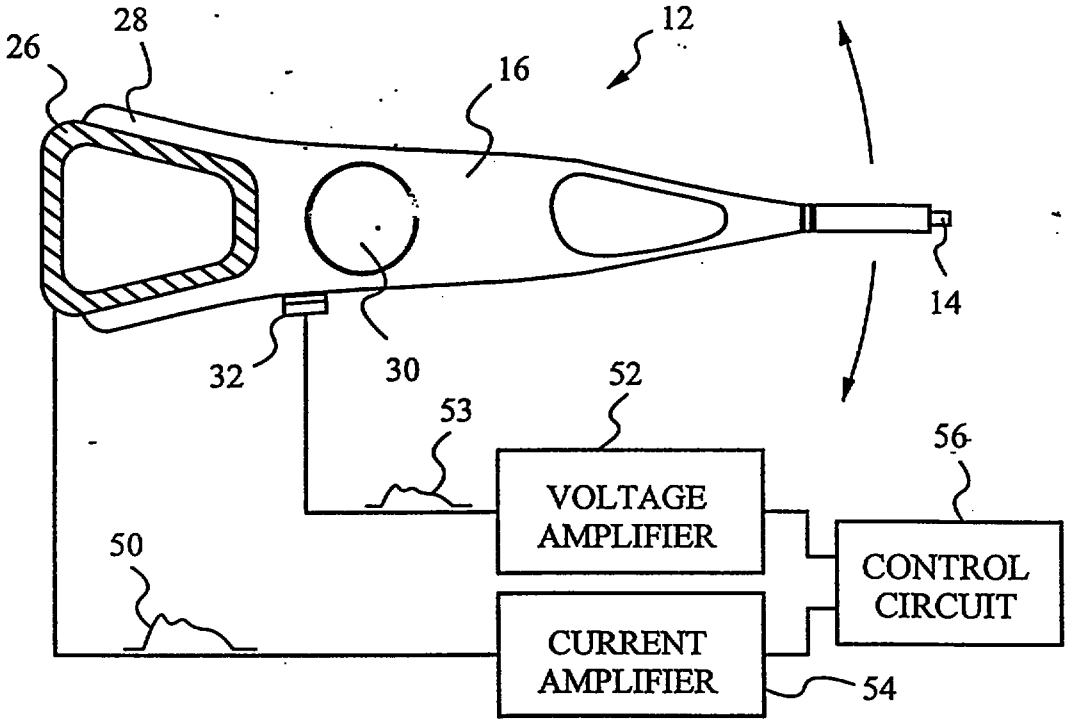

26
28
12
16
14
32
30
52
53
56
VOLTAGE AMPLIFIER
CONTROL CIRCUIT
50
CURRENT AMPLIFIER
54

FIG. 3

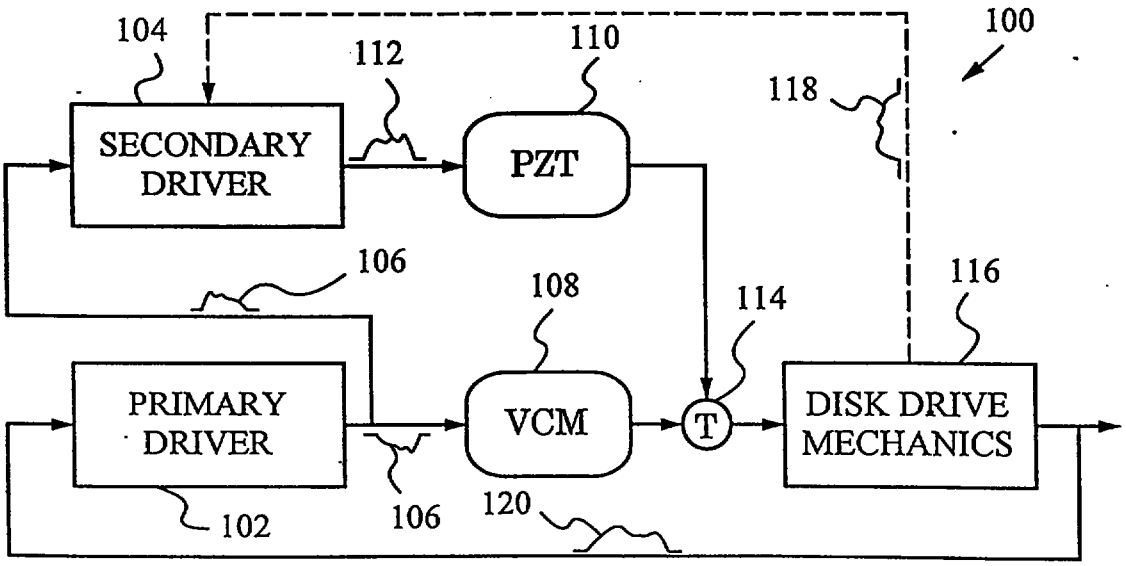

104
112
110
100
118
SECONDARY DRIVER
PZT
106
108
114
116
PRIMARY DRIVER
VCM
T
DISK DRIVE MECHANICS
102
106
120

FIG. 4

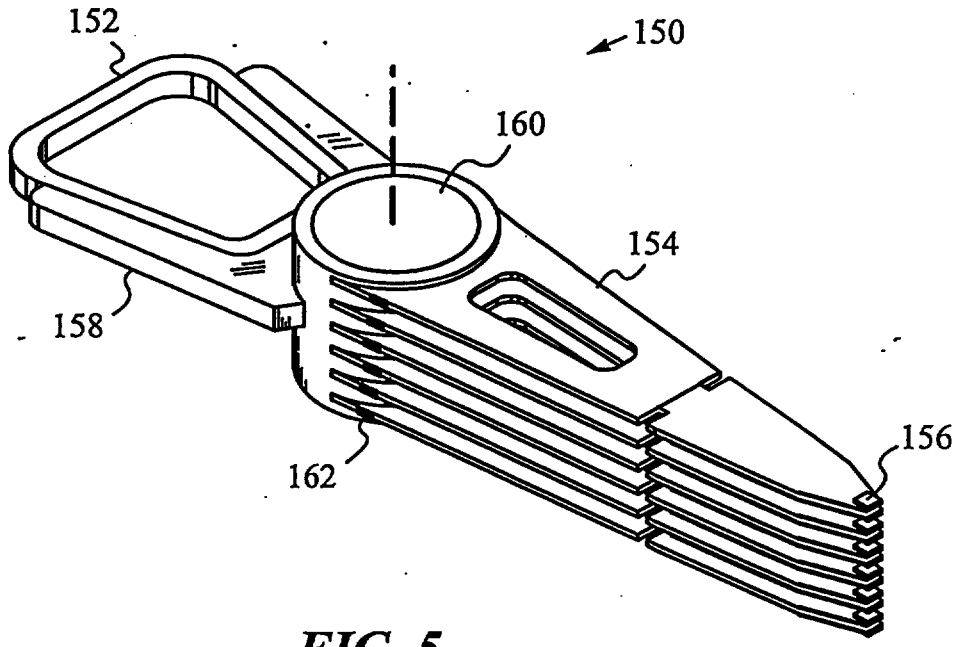


FIG. 5

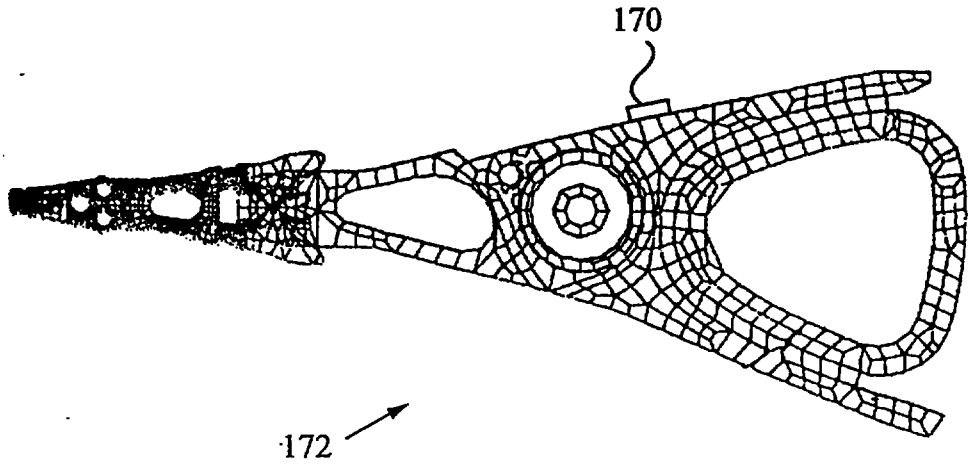


FIG. 6

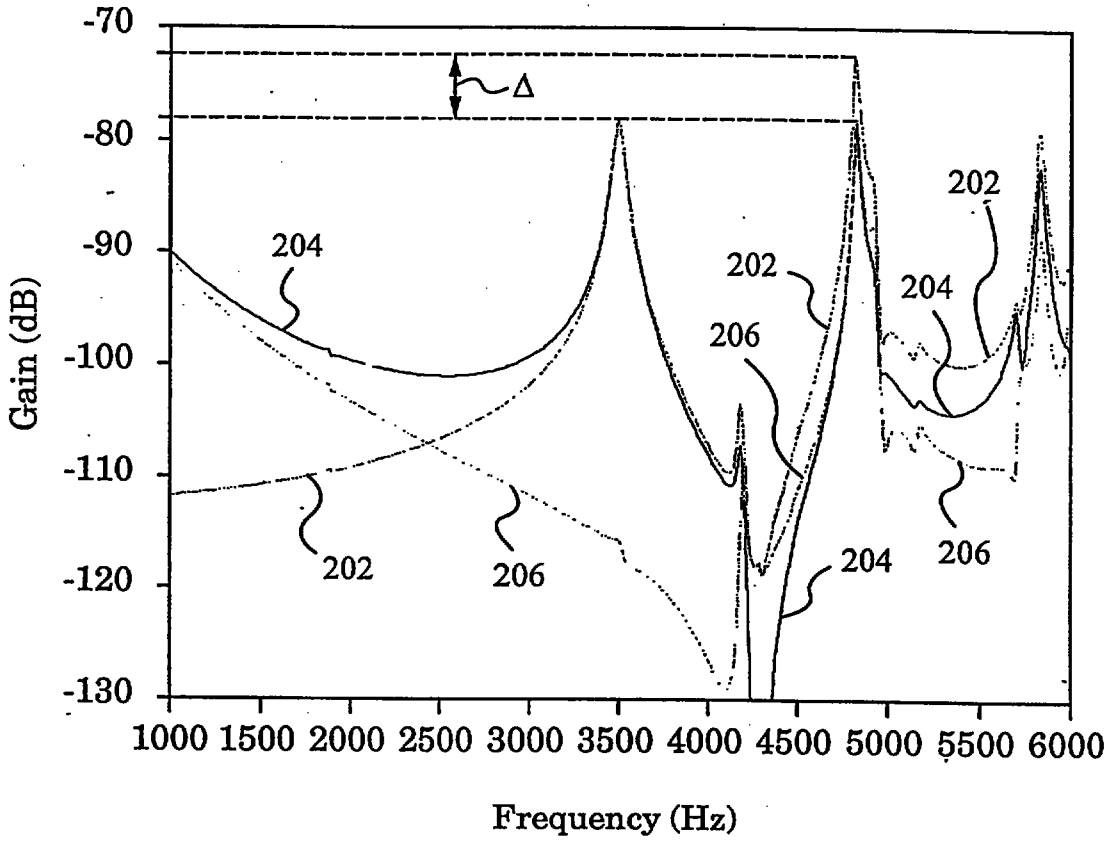

-70
-80
-90
-100
-110
-120
-130
Gain (dB)
Δ
204
202
206
202
204
206
202
206
204
1000 1500 2000 2500 3000 3500 4000 4500 5000 5500 6000
Frequency (Hz)

FIG. 7

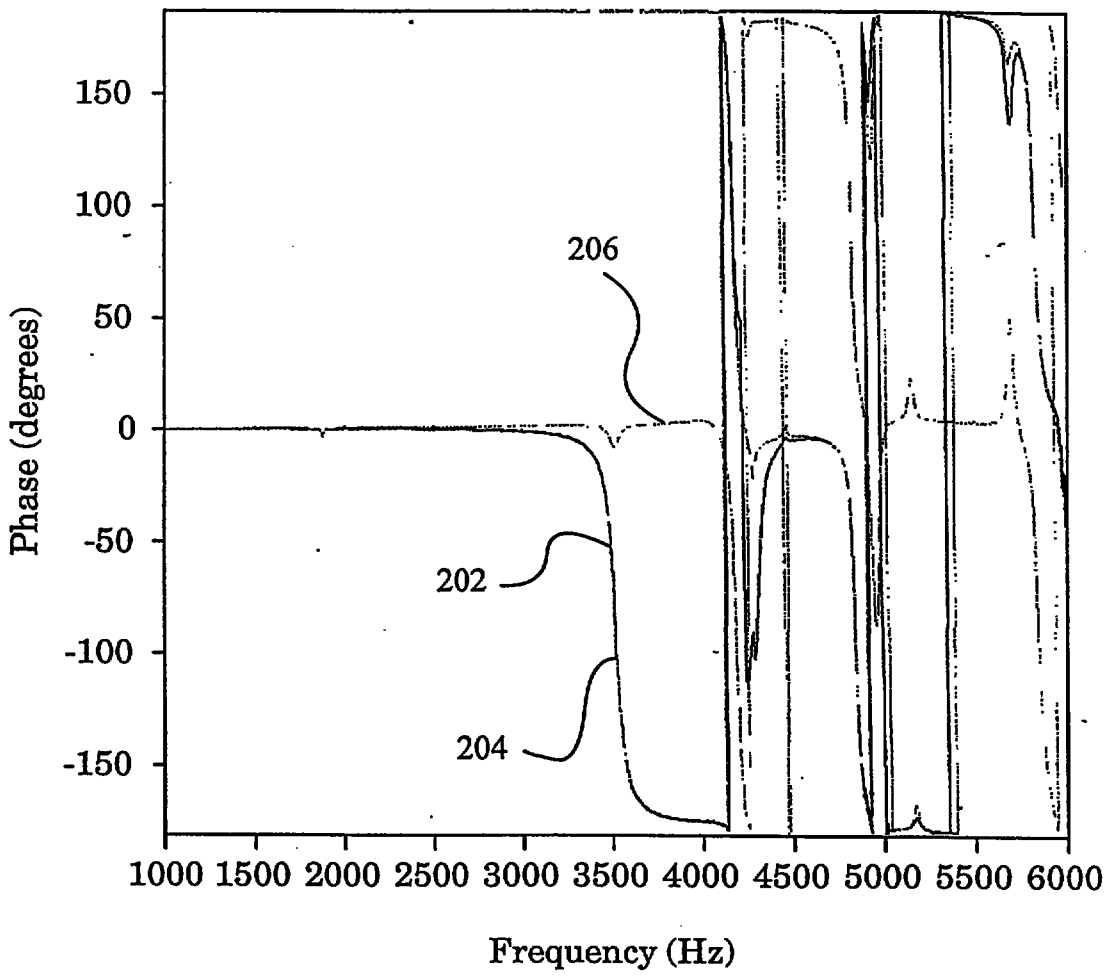

150
100
50
0
-50
-100
-150
Phase (degrees)
206
202
204
1000 1500 2000 2500 3000 3500 4000 4500 5000 5500 6000
Frequency (Hz)

FIG. 8

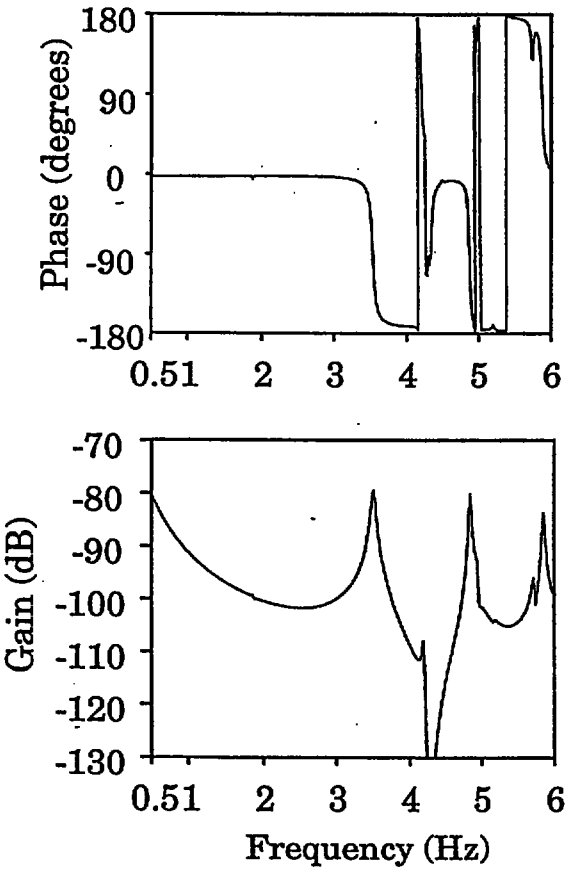

Phase (degrees)
180
90
0
-90
-180
0.51
2
3
4
5
6
Gain (dB)
-70
-80
-90
-100
-110
-120
-130
Frequency (Hz)

FIG. 9A

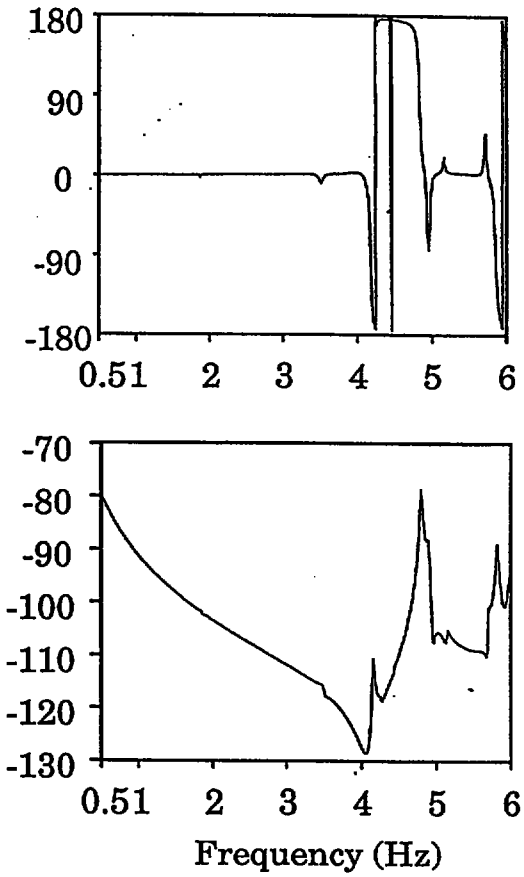

180
90
0
-90
-180
0.51
2
3
4
5
6
-70
-80
-90
-100
-110
-120
-130
Frequency (Hz)

FIG. 9B