Head actuator dynamics of an IBM 5¼-inch disk drive

by D. P. Fazzio
M. A. Moser
C. J. Polson
J. N. Scheffel

The IBM 5¼-inch disk drive contained in the IBM 9345 DASD Module provides high track density and storage capacity, dynamic test failure rates below three parts per million, and low sensitivity to assembly variations. The design techniques used to achieve the required vibrational characteristics of the head actuator assembly are described. Dynamic stability specifications are derived from drive performance requirements and the actuator servomechanical system design. Modeshapes of the actuator are determined by encoding magnetic patterns onto a disk and using the read/write heads as position transducers in an operational drive. Structural changes in the carriage assembly that might lead to design improvements are explored with models derived using finite element analysis. Taguchi orthogonal matrix experiments are used to reduce the sensitivity of the actuator to dimensional tolerances and assembly processes. The achievement of actuator assembly design objectives is verified from production yields and statistical data obtained during dynamic tests.

Introduction

Since the invention of the hard disk drive at IBM in 1957, many hard disk drive models with differing mechanical designs have been used in direct access storage devices [1]. One factor which has limited the data density (bits per square inch) and storage capacity of a drive has been vibration of its components. Considerable efforts have been made to design servo control systems that reduce the effect of vibration on operation of a hard disk drive [2–4], but servo systems cannot compensate for all mechanical vibrations. Structural design improvements are required to minimize vibrations in order to optimize drive performance. Several test methods have been used to study and resolve vibration problems. These have included frequency response, transient response, and, more recently, modal analysis. In some drives, vibration is so complex that these tools are not adequate, and 100% inspection of the drives produced is required to ensure their quality. This paper describes the successful application of four innovative techniques which have been applied to reduce mechanical vibration in a hard disk drive.

The high-performance 5¼-inch IBM disk drive contained in the IBM 9345 DASD Module (hereafter referred to as the 9345 disk drive) is used in IBM Models 9341 and 9343 low-

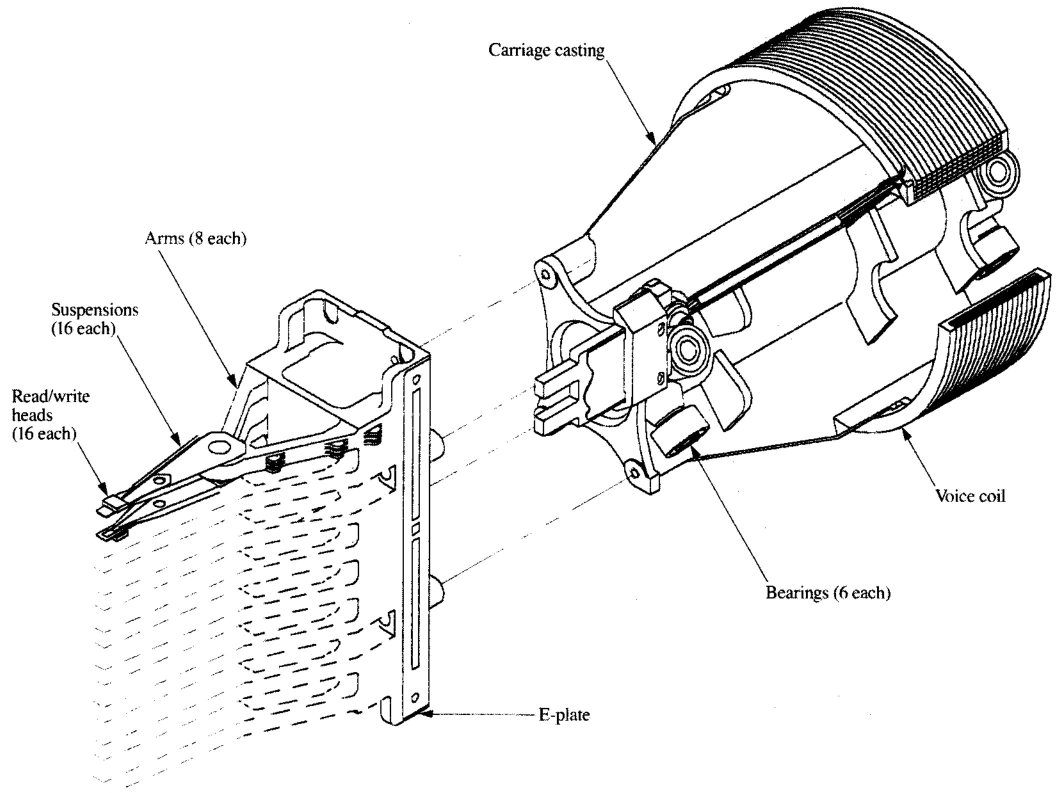

Figure 1

Carriage assembly of a 5¼-inch disk drive in an IBM 9345 DASD Module.

end mainframe storage devices and has been used in IBM Model 9570 high-speed (sustained 50+ megabytes per second) array devices used for scientific computing and visualization. Each drive unit has a capacity of 1.5 gigabytes. Each drive has nine disks rotating at 5400 revolutions per minute which provide a data rate of 4.5 megabytes per second. The 9345 drive has 16 functional disk surfaces. One of the disk surfaces, known as the servo surface, is encoded with a set of concentric magnetic patterns (tracks) that are used solely to determine the relative position of the actuator and disk stack. The remaining 15 disk surfaces are used for data storage and contain no positioning information. This is referred to as a dedicated servo architecture.

Figure 1 shows the structure of a portion of the linear actuator [5] used in the IBM 9345 disk drive. This assembly consists of a voice coil in a magnetic field with an E-plate to support the read/write heads and suspensions on arms which move between the disks. A carriage casting connects the coil to the E-plate and guides the assembly via six bearings which roll on cylindrical rails. These parts constitute the moving mass of the actuator, known as the carriage assembly, or carriage. Its purpose is to position the read/write heads relative to data tracks present on the disks. A servo system controls the carriage position by applying a current to the voice coil and producing a force to move the carriage. The resulting position change is then determined by reading the pattern on the dedicated servo surface.

Access time is a measure of how quickly an actuator can move to a desired track and begin accessing data; it is one of the principal indicators of drive performance [6]. The 9345 disk drive has an 11-millisecond weighted average access time.

The relationship between the head position (output) and the voice coil current (input) depends on the frequency of the input current. There are two reasons for this. First, the force due to the voice coil current produces a carriage acceleration equal to the second derivative of the carriage displacement. Compensation systems are used in disk drives to eliminate this second derivative effect and make the output-to-input relationship independent of frequency for low frequencies (below 300–400 Hz for this disk drive). The use of a closed-loop feedback controller for the servo system improves the tracking accuracy, increases the frequency range over which the output-to-input relationship remains constant, and improves the disturbance rejection capability of the system. The second reason for the frequency dependence occurs at higher frequencies. The mechanical vibration of the actuator produces amplitude and phase differences between the output and input. If the resonances are not properly accounted for in the servo controller design, they lead to instabilities which are manifested as vibrations of large amplitude.

This oscillation can move the data heads off the data tracks and cause errors in reading data or can partially erase adjacent tracks during writing. Consequently, actuator vibration limits the achievable track density and storage capacity of a disk drive. To compensate for severe actuator vibration, it may be necessary to design the servo system for slower access times and lower performance levels. Designing a competitive disk drive requires balancing the servo system design, the actuator mechanical design, the track density, the access time, and the power consumption. This paper discusses the techniques used to improve the actuator's mechanical characteristics.

The primary actuator design objectives are 1) to minimize the carriage inertia and 2) to improve the dynamic response of the actuator structure. Minimizing carriage inertia decreases access time and voice coil power requirements. Improving the dynamic response of the actuator by reducing vibration amplitude produces better disk drive performance. The carriage inertia is a function of the density and volume of the materials used in the carriage. The best materials are those with high stiffness and low density, but material selection is limited by cost, manufacturing, and technical considerations. Once materials have been selected, the only control of carriage inertia is the volume of its parts. Reducing the volume decreases the inertia, but this can also reduce the carriage stiffness and may degrade the dynamic response of the actuator. For this reason, minimizing carriage inertia and improving the dynamic response of the actuator structure are competing objectives which must be balanced.

The dynamic response of an actuator is determined by applying a variable frequency current excitation to the voice coil (input) and measuring the off-track acceleration of the servo head radial to the disk (output). The ratio of output to input as a function of frequency produces a frequency response function (FRF), which compares the behavior of an actuator to that of a rigid body or ideal system. Rigid body behavior yields a constant FRF independent of frequency, and deviations are due to mechanical resonances in the actuator. The frequency response from an early prototype of the 9345 disk drive is shown in **Figure 2** with the significant modes identified. The vertical axis measures the magnitude of the servo head acceleration relative to the acceleration of the coil (gain). The maximum acceptable amplitudes for various frequencies depend on the servo system and the drive design characteristics. The amplitudes must fall below a specification line to guarantee actuator stability while moving (seeking), arriving (settling), and transferring data (track following); see Figure 2. Determination of a specification line is discussed in the next section.

The dynamic stability of actuators in mass production varies from drive to drive. If FRF failure rates are high, 100% testing of disk drives is required in order to screen out bad actuators and ensure quality. However, if projected actuator FRF failure rates are below three parts per million, statistical process control techniques [7] can be used to test a sample of disk drives and ensure dynamic stability of all the drives produced. Since performing FRF testing is a costly and time-consuming process, achieving this low failure rate is a primary objective.

A specification margin is the difference between a local maximum on the FRF plot and the specification line at a given frequency. A positive margin indicates that a resonance peak is below the specification, and that the actuator characteristics at that frequency are acceptable. By measuring margins on a statistically significant number of drives and assuming a normal distribution, a mean and standard deviation (σ) is calculated [7]. If the 4.5σ spread of the margins is entirely above zero, the failure rate is less than three parts per million, and dynamic stability of the population is ensured. This is based on the fact that, for a normal distribution, 99.99966% of the population is under the curve to one side of the 4.5σ line [8].

As indicated by test results, the target failure rate was achieved for the IBM 9345 drive by applying four techniques:

1. To obtain dynamic stability specifications for the FRF test, a new methodology derived from control system theory [9, 10] is applied.
2. A new procedure is introduced to measure actuator modeshapes in a functional disk drive. This procedure eliminates the need for actuator removal and aids in problem isolation and model validation.
3. Finite element analysis (FEA) is used to identify the source of two high-frequency resonances and generate design modifications to improve them.
4. Taguchi matrix experiments are used on the assembly process to identify and optimize the source of a resonance which is tolerance- and process-sensitive.

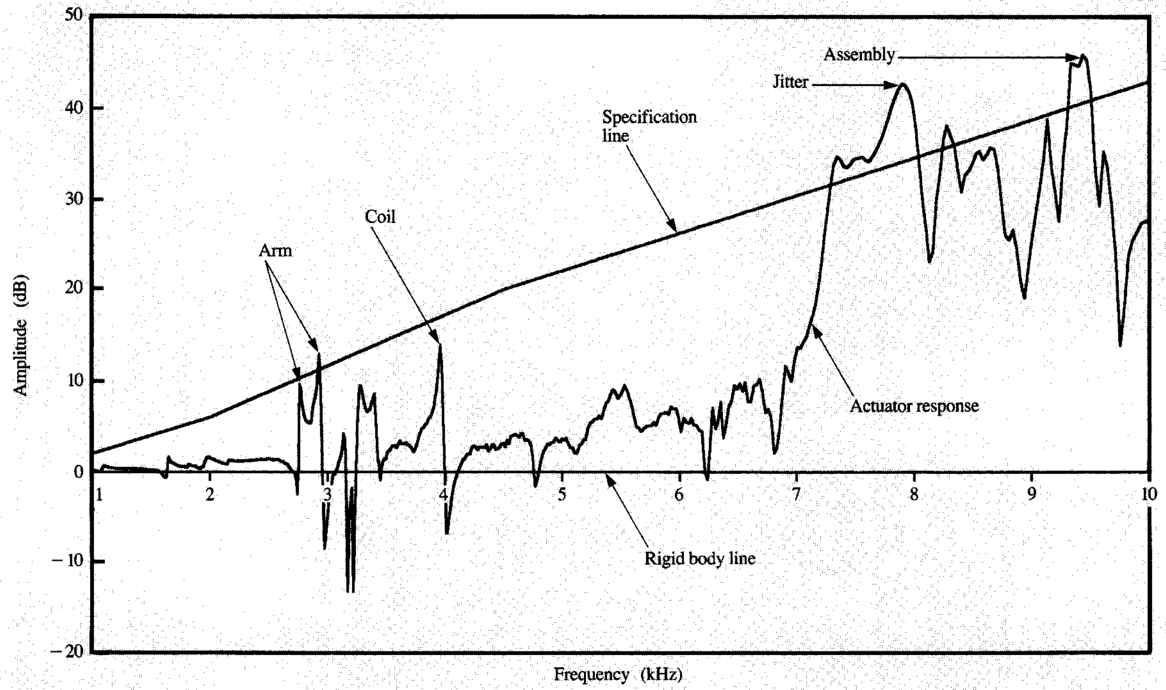

Figure 2

Measured frequency response function of a sample actuator compared with original specification.

These techniques used to design the drive's actuator could also be applied to design of other mechanical systems in which dynamic behavior is important.

Development of a stability specification for the actuator

The ideal behavior for the actuator is that of a rigid body. However, this cannot be achieved over the frequency range of interest, since it would lead to an actuator with excessively high mass. To obtain an actuator with acceptable mass, a compromise is made between reducing dynamic performance and lowering mass. For most of the frequency range, a specification for the dynamic characteristics of the actuator is determined from stability bounds on the combined servomechanical system. This specification defines the compromise boundary between the actuator mass and its dynamic performance. A superior actuator design is one in which only the extremes in population distribution fail the specification at significant resonances. If there are either no failures or a large number of failures, it is an indication that an incorrect trade-off has been made among actuator mass, the amplitude of the resonances, and the servo system performance.

A stability specification for the actuator can be derived by first conceptually subdividing the mechanical system into a nominal system and an "error" system, as shown schematically in **Figure 3**. The nominal system for the 9345 disk drive includes only the rigid body characteristics of the actuator. The error system represents deviations in drive operation from a nominal performance. Bounds are placed on the error system to ensure that the combination of actuator and servo system controller is always stable and that perturbations do not cause the system response to become unbounded.

If a continuous, linear, time-invariant system is stable, its response to a perturbation decays exponentially with time. Conversely, an unstable system has a response that increases exponentially with time. The type of behavior the system exhibits is determined from the location of the system poles, the singular values of the transfer function. For the disk drive, the system referred to is a combination of the actuator, the controller, and the feedback loop. The transfer function relates the amplitude and phase of the output to the input of the system as a function of complex frequency. Measurements of the transfer function can only be made along the imaginary frequency axis; this is the frequency response function.

When the complex frequencies of the poles of the closed-loop system all have negative real parts, the system is stable. If any pole has a positive real part, the system is unstable, and the response grows exponentially with time. Determination of the stability of a system is reduced to locating its poles, which requires a mathematical description of the system. Since such a description can be difficult to obtain, indirect methods such as the Nyquist stability criterion [9–11] can be used to determine the system stability. The Nyquist criterion also provides a method for deriving a frequency response specification for the system.

In deriving the stability criterion, a contour is drawn to extend along the imaginary frequency axis and extended to enclose the positive real portion of the complex frequency plane (see **Figure 4**). Any poles of the closed-loop system that lie in this portion of the plane lead to behavior that

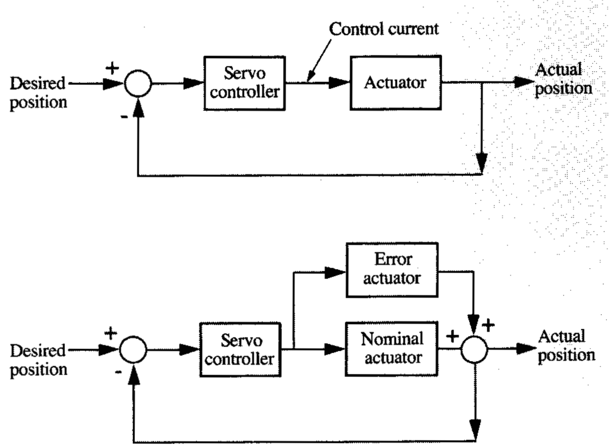

Figure 3

Decomposition of actuator dynamics into nominal and error systems.

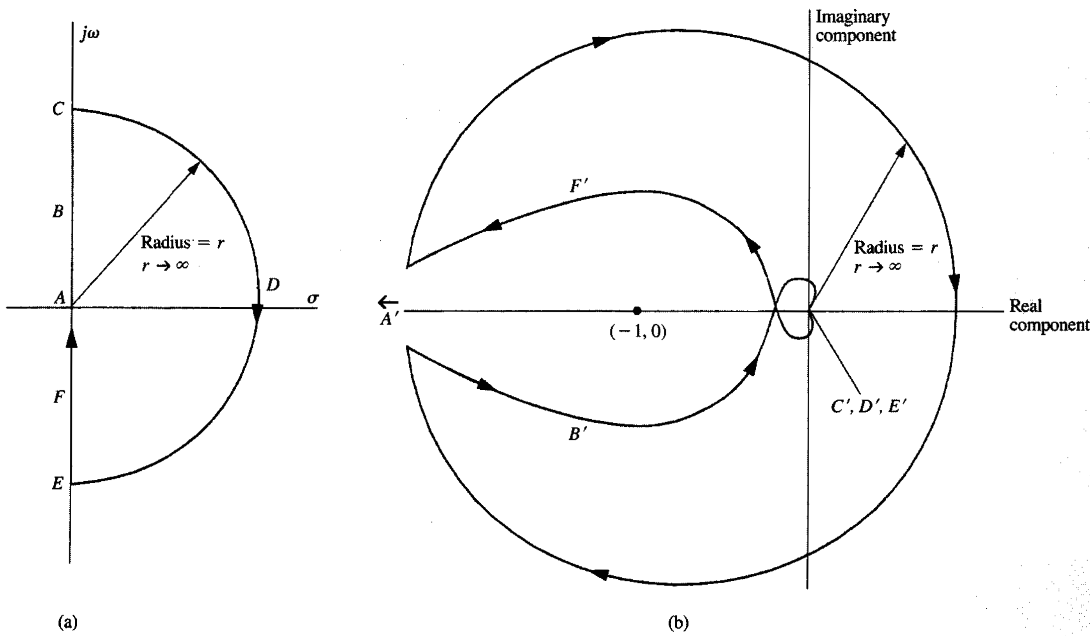

Figure 4

Integration contours used for computation of Nyquist stability criterion: (a) complex frequency plane; (b) open-loop controller/actuator response plane. Points on the complex frequency plane corresponding to open-loop controller/actuator response plane are labeled with letters A through F.

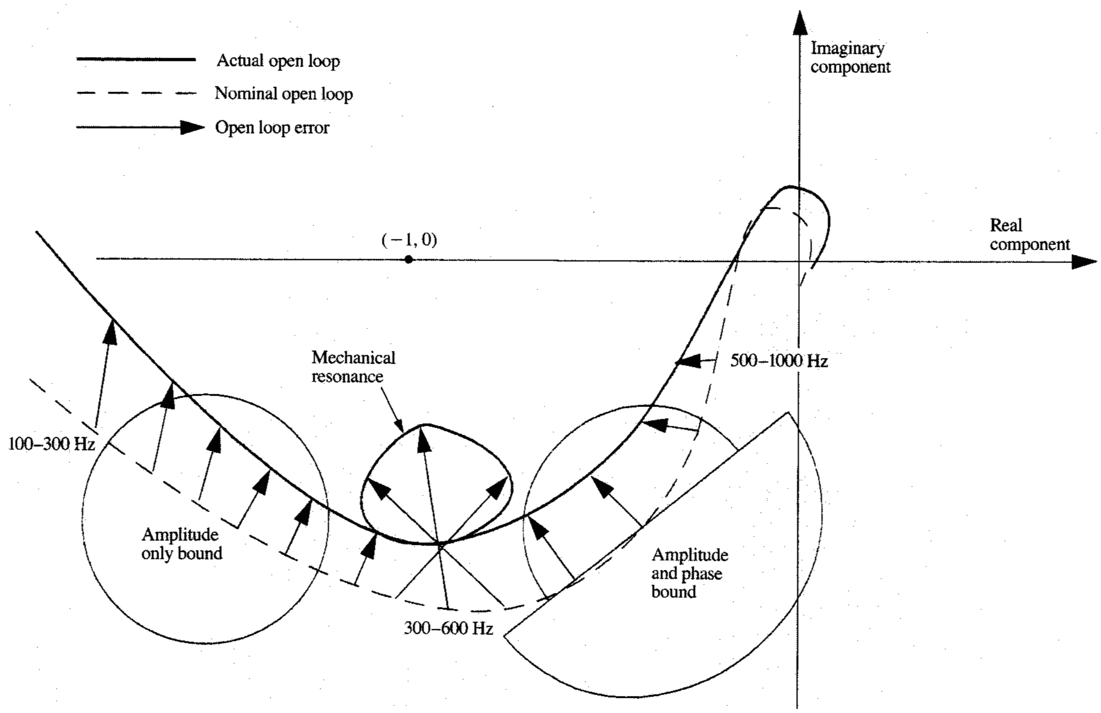

Figure 5

Schematic of half Nyquist contour, illustrating stability limits and specification bounds.

grows exponentially with time. By examining the response of the open-loop system as the frequency parameter is varied along the imaginary frequency axis, the stability of the closed-loop system is determined by counting the number of times that the point (−1, 0) is encircled on the open-loop controller/actuator response plane. For the type of controller used in the 9345 disk drive, the system is stable if there are no encirclements.

Since the stability characteristic determined is global, it is not suitable for deriving a test specification at particular frequencies. A method is presented below for establishing a criterion that can be applied at particular frequencies to determine stability. Since only sufficient conditions for stability are provided, the system is guaranteed stable if it satisfies these conditions, but is not necessarily unstable if it does not meet the criterion.

The nominal system is designed to be stable, and requirements are then placed on the error system to ensure that it does not destabilize the combined system. A sufficient stability condition is that the nominal and combined system contours lie on the same side of the (−1, 0) point, as shown in **Figure 5**. One way to satisfy this condition is to ensure that the vector connecting the nominal and actual systems at a given frequency does not reach the point (−1, 0) even after its direction (phase) is arbitrarily changed. A stability specification for the maximum amplitude of the error system is then established independently of its phase. In practice, the amplitude should be significantly smaller than this maximum, since the performance can be poor if the system deviates significantly from the nominal design. This reduced amplitude is implemented through a safety factor, which is the ratio of the length of the vector required to reach (−1, 0) to the maximum length that is acceptable for performance reasons. For the 9345 disk drive, the safety factor is 3.0 below 1500 Hz, since resonances in this range cause overshoot during arrival; from 1500 to 4000 Hz, the safety factor is lowered to 2.0, since the dynamics are well understood over this range and resonances do not lead to performance problems; above 4000 Hz, the safety factor is increased to 3.0, since resonances in this range are difficult to measure accurately within test time limitations.

The specification derived from the Nyquist plots has been further enhanced. If an error vector points away from

(−1, 0), it is less likely to destabilize the system and can have a larger amplitude. The resulting criteria are shown in Figure 5, where semicircles define the acceptable regions for each frequency. Such a diagram provides a pair of amplitude specifications in which the relevant amplitude curve is given by the phase of the error vector.

Measurement of actuator modeshapes in operational drives

To develop a reproducible mechanical system with minimal performance failures, modal analysis must be applied to an assembled drive. The technique presented in this section allows measurement at particular frequencies of the deformed shapes, or modeshapes, of an actuator which is installed in an operational disk drive.

When dissipative forces in a structure are small and resonant frequencies are well separated, modeshapes correspond to deformations of the actuator structure at the resonant frequencies. Although these deformations can be measured with various optical or electromechanical sensors, hardware modifications require additional expense and time and can alter the system dynamics. When specialized patterns are encoded onto a disk with magnetic read/write heads, the heads become high-resolution position transducers sensitive to motion in the radial direction of the disk. Frequency response functions between the voice coil motor and each of the magnetic element locations are obtained by applying a sinusoidal forcing function with the actuator voice coil motor and varying the excitation frequency in a stepwise manner. Fitting techniques [12, 13] are applied to the data to determine the natural frequencies and modeshapes at the transducer locations. Deformations in the remainder of the actuator are estimated from kinematic assumptions about the relative motion of other points in the structure based on finite element simulations. An exaggerated deformation of the E-plate structure at the front of the actuator is shown in **Figure 6**.

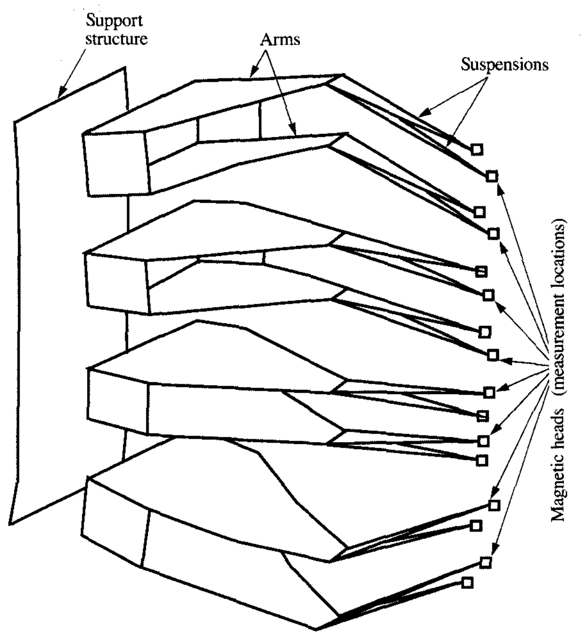

Figure 6

Sample mode shape of E-plate assembly extrapolated from head position measurements (amplitude exaggerated).

Although this approach was initially applied to establish whether the finite element simulations corresponded to the behavior of the real actuators, the ability to measure modeshapes without modifying the drive has led to other applications. When minor changes are made to components, qualification plans now include this method only; full instrumentation tests are reserved for testing major modifications. Modes not predicted by finite element modeling can be measured and used to refine the simulations. Process changes can also be tracked, and decisions on whether an alteration is acceptable are made rapidly.

Finite element analysis and structural design improvement

In finite element analysis (FEA), a mathematical model of a mechanical system is constructed on the basis of its geometry and material properties [14]. This model is used to simulate dynamic behavior, producing results similar to those from modal testing. A primary advantage of FEA over modal testing is that FEA can be used to evaluate different mechanical design options quickly, since model modifications are much faster than fabricating prototype hardware. Other advantages are that structural displacements are obtained from FEA at many more points than from testing, and the strain energy of a modeshape is also calculated from FEA data. This section describes how FEA was applied to refining the carriage assembly structure in the 9345 disk drive.

A finite element model of the carriage assembly excluding suspensions and heads was constructed and analyzed using two commercially available FEA software packages, CAEDS® and MSC/NASTRAN® [15, 16]. The original model is shown in **Figure 7**. Natural frequencies and modeshapes of the carriage were calculated with the software, and the frequency responses were analyzed. The validity of the FEA model was verified by comparing 1) the frequency response results from modeling to FRF testing and 2) the modeshapes from FEA and modal testing. The similarity between the model and test results verified that the FEA model could be used to investigate structural design changes.

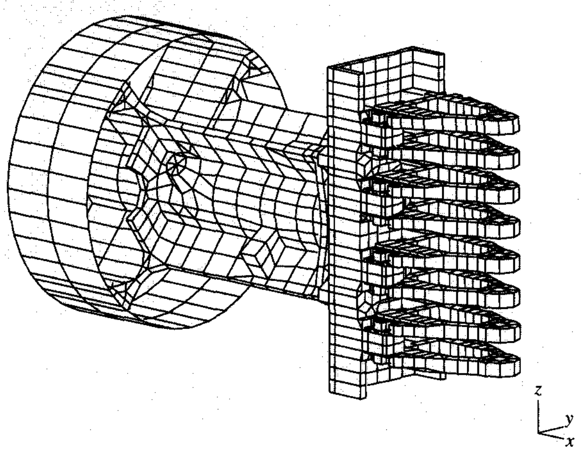

Figure 7

Finite element model of original carriage assembly design.

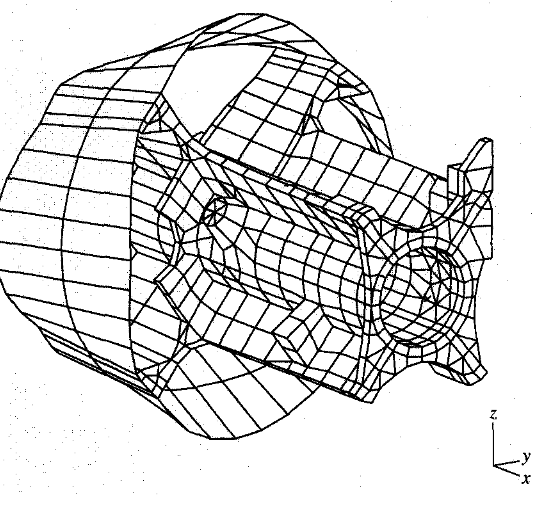

Figure 8

Coil mode shape at 4000 Hz (amplitude exaggerated).

Although each of the major modes shown on the FRF plot in Figure 2 was studied with FEA, it was most effective in reducing the coil mode (4000 Hz) and the assembly mode (9500 Hz). While the coil mode marginally passed the FRF specification on most drives, the assembly mode did not meet the specification on a significant number of drives. The deformed shapes of the two modes (with the E-plate removed for visualization) are shown in **Figure 8** and **Figure 9**. The deformations are exaggerated to aid visualization. The coil mode deformation was primarily in the voice coil, although the carriage casting was also distorted. The name "assembly mode" was applied because all carriage assembly components deformed at this frequency.

From animated studies of modeshapes of the carriage structure, twelve structural modifications were conceived and investigated in an effort to reduce the FRF amplitude at the coil and assembly mode frequencies. Each modification was simulated and evaluated in about one day using FEA, whereas prototyping and testing each modification would have required weeks. FEA eliminated consideration of modifications that yielded no significant improvement. Time-consuming prototyping and testing were reserved for modifications that appeared to reduce the resonances.

Three changes effectively reduced the coil and assembly modes and were combined into a "stiffened carriage" design. A comparison of the original design model in Figure 7 and the stiffened carriage drawing in Figure 1 provides an idea of the required modifications. Unfortunately, the stiffened carriage design increased moving mass by several grams; since this would increase access time, a reduction of the carriage mass was required. FEA was used to calculate the strain energy distribution during modal deformation, thereby determining which carriage areas underwent little strain at a mode and were therefore least important to maintaining carriage stiffness at that mode. Low-strain areas provided the best candidates for material removal without adversely affecting the carriage modes. These results could not have been obtained by any available test methods.

Since each mode produced a unique strain energy distribution, it was difficult to determine which areas could be modified without adversely affecting any of the significant carriage modes. A method was devised to combine the strain energy data for all of the significant modes into one data set within CAEDS to visualize and identify areas that would have minimal impact on those modes. Using this method, mass was removed without adversely affecting the amplitude of FRF results. The final design increased the total carriage moving mass by only 1 gram, or less than 1%, and had little effect on access time. The final stiffened carriage model predicted that the specification margin for the coil and assembly modes would increase by 6.0 dB and 6.5 dB, respectively. Statistical test data indicate that the margins were actually increased by 7.2 dB and 5.7 dB, indicating how effectively FEA can predict improvement in the dynamic behavior of a mechanical system. Similar improvements in the specification margins for the arm modes from 2700 to 2900 Hz were achieved by the design of a new one-piece

E-plate to replace the original design, which had welded arms. This design change was also analyzed and refined using FEA (see the Acknowledgments).

Taguchi orthogonal matrix experiment

An actuator resonance near 800 Hz which caused FRF failures as high as 15% in early prototype hardware was particularly difficult to control with the servo system. FEA of the actuator structure predicted modes near 800 Hz but could not duplicate the response amplitude measured by testing of the actuator. Consequently, FEA was ineffective for study of this mode. Furthermore, test measurements were not reproducible for this mode. Disassembly and reassembly of the same hardware would produce a range of 800-Hz responses. The response of an actuator would change after it was moved to a test fixture or disturbed in other ways. These observations led to the conclusion that this mode was sensitive to the assembly process.

Since application of the techniques discussed previously was not successful in reducing this resonance, matrix experiment design techniques were used to study process variations [17, 18]. This procedure requires definition of a set of input variables that affect a process, followed by measurement of the effect that distinct levels or values of these variables have on outputs of the process. Testing all possible combinations of the variables generally requires many trials, each of which may be expensive and time-consuming. However, the advantage of matrix experiment design is that it requires running a minimum number of trials to study all levels of all input variables and will filter out some effects due to statistical variation. The experimental design defines the levels to use for each trial, and on the basis of results identifies which variables are significant and which input levels should be altered.

Several process variables which could influence the 800-Hz response were identified and employed to define a matrix experiment using ADOE (Automated Design of Experiments) software.[1,2] The matrix produced by ADOE is an L18 orthogonal array [17] containing eight variables, one at two levels and seven at three levels. Since the variables were mechanical dimensions within a tolerance range and assembly methods, not the simple process adjustments traditionally used for matrix experiments, the experiment was unique and difficult to control. Requirements included special parts for tests at the tolerance range extremes and careful tracking of the units through the assembly process to ensure that correct parts and assembly methods were used.

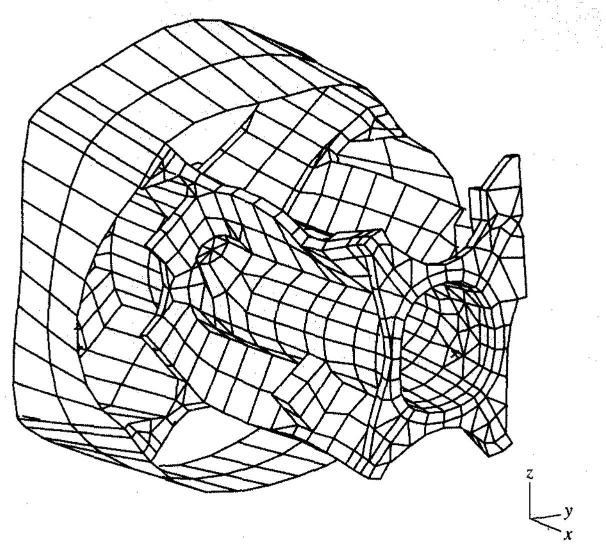

Figure 9

Assembly mode shape at 9500 Hz (amplitude exaggerated).

The matrix experiment quantified the relative effect of each variable on the amplitude of the 800-Hz response. Data analysis indicated that the baseplate coplanarity, which measures the planarity of the two actuator support surfaces, was the most significant variable. Surprisingly, the 800-Hz response improved as coplanarity increased (meaning that the two support surfaces became more out of plane). Isolator/rail skew is a variable which defines the alignment of the rails which guide the carriage. It also was significant and improved the 800-Hz response as misalignment increased. Any variable that increased distortion to the assembled actuator apparently decreased the 800-Hz resonance and shifted the frequency higher. The distortions presumably altered the damping in the actuator assembly and reduced the amplitude of vibration.

A second matrix experiment was designed to confirm the original results and study sensitivity to the baseplate coplanarity. The results of this experiment confirmed that the baseplate coplanarity does provide leverage to reduce the 800-Hz FRF resonance. An increase in the baseplate coplanarity to 13 μm increased the mean of the FRF specification margin at 800 Hz from about 5.5 dB to 10.5 dB.

Improvements in 5¼-inch disk drive dynamics

The effort to minimize vibrations in the IBM 9345 disk drive was initiated early in its development process. At that time, prototype disk drives were built and tested to verify the performance of the design. FRF test results on

[1] J. Rustagi, V. Singh, S. Ghosh, C. Carroll, L. Fleming, T. Nguyen, and K. Wong, "Introduction to Automated Design of Experiments: Taguchi Approach," Vols. I and II, July 1992 (IBM Internal).

[2] T. Nguyen, V. Singh, S. Ghosh, J. Rustagi, C. Carroll, L. Fleming, and K. Wong, "Automated Design of Experiments (ADOE) User's Guide (Version 2.0), April 1992 (IBM Internal).

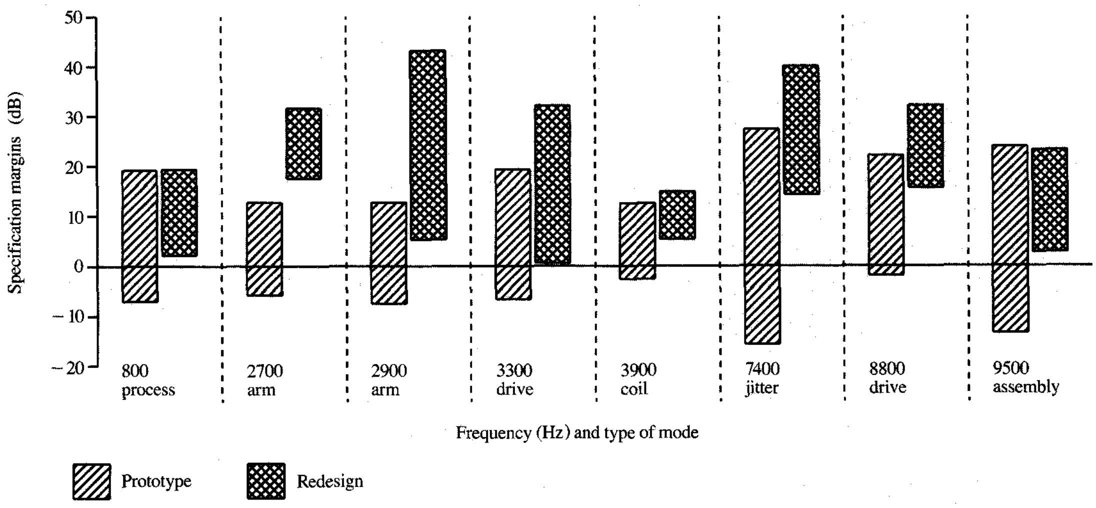

Figure 10

Improvement in FRF specification margins for the disk drive actuator in the IBM 9345 DASD Module.

these prototypes contained specification margins below zero in several frequency ranges, indicating mechanical vibration problems which could cause dynamic instability. Results from this prototype testing were the motivation for mechanical redesign of the actuator.

A sufficient number of prototypes were built and tested to statistically predict the 4.5σ range of specification margins. **Figure 10** illustrates 4.5σ ranges of the specification margin at frequencies with significant mechanical resonances. The ranges were calculated from FRF test data obtained from more than 100 prototype and final design drives.

The graph shows that the failure rates at FRF test were unacceptable for the prototype design. As previously noted, any portion of a 4.5σ range of specification margin that falls below zero represents a percentage of disk drives which would fail FRF testing at that frequency and require rework to correct the problem. If any of the 4.5σ range of specification margins for a given design fall below zero, the FRF failure rate of the design is greater than three parts per million, and 100% of the produced drives must be tested to ensure quality.

As Figure 10 illustrates, all of the specification margins for the final design are above zero, indicating that the objective of a failure rate of less than three parts per million was achieved. This is the design of the 5¼-inch disk drive, as shipped in the IBM 9345 DASD Module, and FRF performance of the disk drive is audited by statistical quality control techniques instead of 100% testing.

Summary

A successful design of the mechanical actuator for the 5¼-inch disk drive contained in the IBM 9345 DASD Module was achieved by using the following procedures:

- Generation of a stability specification based on performance requirements of an actuator, control theory, and dynamic behavior. The specification produced ensured that the desired balance between stability and performance was achieved.
- Encoding special patterns on the disks to measure radial head motion by using the heads as position transducers. This procedure, combined with kinematic analysis, produced dynamic measurements of actuator modeshapes in a functioning disk drive.
- Use of finite element analyses to visualize and evaluate structural changes that had the potential to reduce actuator vibration and to provide detail about the dynamic behavior of the actuator, including strain energy distributions. Application of these techniques resulted in the reduction of two significant modes by several decibels, with no adverse effect on access time.
- Use of Taguchi matrix experiments, which demonstrated that component tolerances and process variations have a significant influence on an actuator resonance at 800 Hz. Control of the inputs resulted in reduction of the resonance to targeted levels.

Application of the first technique provided a meaningful specification against which drives could be tested. When test results indicated that the drives were not meeting this specification, the other techniques were used to reduce the problem resonances to acceptable levels. Statistical analysis of test results proved that the techniques resulted in a disk drive design with virtually zero FRF failures.

The use of this combination of techniques to design the actuator in the 9345 disk drive was instrumental in producing a field performance with a failure rate of less than 100 parts per million for all disk drive problems. This approach virtually eliminated the need for sustained design engineering support after the drive went into production.

Acknowledgments

We would like to acknowledge the contributions of the following individuals to the success of this effort: Cathy Carroll, for help with running ADOE and setting up the matrix experiment; Pete Kanas, for extensive modal testing of the actuator structure and one piece E-plate design; Joe Lillig, for servo design and support; Greg Link, for modal testing; Jim Payne, for design modifications of the carriage casting; Victor Shum, for construction and analysis of FE models for the one-piece E-plate design; Vic Singh, for help with running ADOE and setting up the matrix experiment; Matt Stangl, for programming data collection routines for data head modeshapes; and Greg Yamamoto, for servo design and support.

CAEDS is a registered trademark of International Business Machines Corporation.

NASTRAN is a registered trademark of the National Aeronautics and Space Administration. MSC/NASTRAN is an enhanced proprietary version developed by the MacNeal/Schwendler Corporation.

References

1. M. Bohl and A. J. Bonner, *Introduction to IBM Direct Access Storage Devices*, Order No. SR21-3208, Science Research Associates, Inc., Chicago, 1990.
2. R. Horowitz and J. McCormick, "Self-Tuning Control Scheme for Disk File Servos," Adaptive Learning and Control, ASME, Dynamic Systems and Controls Division, DSC v 21, New York, 1990, pp. 23–31.
3. D. K. Miu and S. P. Bhat, "Residual Vibrationless Track Accessing Control for Computer Rigid Disk Drives Using Laplace Transform Technique," *Adv. Info. Storage Syst.* **3**, 161–171 (1991).
4. D. K. Miu, G. M. Frees, and R. S. Gompertz, "Tracking Dynamics of Read/Write Head Suspensions in High-Performance Small Form Factor Rigid Disk Drives," *ASME J. Vibration & Acoust.* **112**, 33–39 (1990).
5. R. K. Oswald, K. Wasson, and J. A. Wagner, "The Disk File Linear Actuator: An Introduction to the Magnetics and Control," San Jose State Engineering Institute, San Jose, CA, 1981.
6. J. A. Wagner, "The Actuator in High Performance Disk Drives: Design Rules for Minimum Access Time," *IEEE Trans. Magn.* **MAG-19**, No. 5, 1686–1688 (1983).
7. *Statistical Quality Control Handbook*, Western Electric Company, Indianapolis, IN, 1982.
8. W. L. Hays, *Statistics*, Holt, Rinehart, & Winston, New York, 1981, p. 646.
9. R. C. Dorf, *Modern Control Systems*, 3rd Edition, Addison-Wesley Publishing Co., Reading, MA, 1980, pp. 262–309.
10. J. C. Doyle, B. A. Francis, and A. R. Tannenbaum, *Feedback Control Theory*, Macmillan Publishing Co., New York, 1992, pp. 33–41.
11. H. Nyquist, "Regeneration Theory," *Bell Syst. Tech. J.* **11**, 126–147 (1932). Also in *Automatic Control: Classical Linear Theory*, G. J. Thaler, Ed., Dowden, Hutchinson & Ross, Inc., Stroudsburg, Pa., pp. 105–126.
12. D. J. Ewins, *Modal Testing: Theory and Practice*, Research Studies Press Ltd., Taunton, Somerset, England, 1984.
13. *The Star System*, Version 3.0, Structural Measurement Systems, Milpitas, CA, 1990.
14. Thomas J. R. Hughes, *The Finite Element Method*, Prentice-Hall, Inc., Englewood Cliffs, NJ, 1987.
15. *CAEDS Graphic Finite Element Modeler User's Guide*, Order No. SH52-0263, available through IBM branch offices.
16. Harry G. Schaeffer, *MSC/NASTRAN Primer*, Schaeffer Analysis, Inc., Kendall Hill Road, Mont Vernon, NH, 1984.
17. M. S. Phadke, *Quality Engineering Using Robust Design*, Prentice-Hall, Englewood Cliffs, NJ, 1989, p. 291.
18. G. Taguchi and S. Konishi, *Orthogonal Arrays and Linear Graphs*, ASI Press, Dearborn, MI, 1986.

Received March 12, 1992; accepted for publication April 8, 1993

Dennis P. Fazzio *IBM ADSTAR, San Jose Laboratory, 5600 Cottle Road, San Jose, California 95193 (FAZZIO at SJEVM5).* Mr. Fazzio is a Senior Engineer currently working in a future disk drive development program. He joined IBM in 1970 in the Advanced Systems Development Division. He has worked in new product development for magnetic recording devices and technology over his entire career. Mr. Fazzio received his B.S. degree in mechanical engineering from California State Polytechnic University, San Luis Obispo, in 1968 and his M.B.A. from Santa Clara University in 1970. He is a member of the American Management Association.

Michael A. Moser *IBM Research Division, Almaden Research Center, 650 Harry Road, San Jose, California 95120 (MOSER at ALMADEN).* Dr. Moser is a Research Staff Member in the Engineering Analysis Software Department at the Almaden Research Center. He received an Sc.B. in mechanical engineering from Brown University in 1981, and an

M.S. in mechanical engineering in 1983 and a Ph.D. in applied mechanics in 1987, both from the California Institute of Technology. He subsequently joined the San Jose Development Laboratory, where he worked on the design of disk drives. In 1991 he received an IBM Outstanding Technical Achievement Award for his contributions to the dynamics of the 9345 disk drive. Dr. Moser is a member of the American Society of Mechanical Engineers.

Christopher J. Polson *IBM ADSTAR, San Jose Laboratory, 5600 Cottle Road, San Jose, California 95193 (POLSON at SJEVM5).* Mr. Polson is a Staff Engineer currently working in a future disk drive development program. He received his B.S. degree in mechanical engineering from the University of Minnesota in 1981, and an M.S. in engineering in 1986 from San Jose State University. He joined IBM in 1982 and has worked on the 3380, 3390, and 9345 disk drive programs, performing modeling, analysis, testing, design, and process engineering tasks. In 1991 he received an IBM Outstanding Achievement Award for his contributions in improving the dynamics of the 3380 disk drive. Mr. Polson is a member of the American Society of Mechanical Engineers.

John N. Scheffel *IBM ADSTAR, San Jose Laboratory, 5600 Cottle Road, San Jose, California 95193 (SCHEFFEL at SJEVM5).* Mr. Scheffel is an Advisory Engineer currently working in a future disk drive development program. He received his B.S. and M.S. degrees in mechanical engineering from Texas Tech University in 1980 and 1982. He subsequently joined IBM and has worked on the 3380, 3390, and 9345 disk drive programs, performing manufacturing engineering, testing, and analysis tasks. For the last six years, his focus has been on using finite element analysis to improve the quality of mechanical designs.