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Design of a Swinging Arm Actuator for a Disk File

Abstract: An integral disk enclosure that incorporates a novel head-positioning actuator concept is used in the IBM System/32 and in recently announced terminal controllers, such as the Retail Store Controller and the Communication Controller. Based on a swinging arm rather than the conventional linear carriage, the actuator presented the designers with opportunities to create a simplified mechanism and to gain advantages in reliability and cost. This paper describes the geometrical, structural, and electromechanical basis of the design.

Introduction

One of the predominant goals in the evolution of storage devices in computer technology has been to increase areal recording density in successive designs with minimal increase in cost. Among the design objectives was one for a disk store having a capacity of five to ten million bytes for terminal and small system applications. In the early stages of the development program for the IBM 3340, the design had been aimed at this modest capacity range, but the high performance servo system being considered could not be cost effective in a single-disk configuration. This development work has been discussed in some detail by Mulvany [1].

One approach to a disk store design was an exploratory development under the direction of Walter S. Buslik, in which a low-mass, low-load recording head (similar to that used in the IBM 3735 Programmable Buffered Terminal) was used on a single disk surface to give a store approximately five million bytes capacity. In this design [2] the drive was to be packaged as a sealed, self-contained unit, because the very low flying height of the head, 0.64 μm (25 μ in.), demanded high levels of cleanliness obtainable only by sealed enclosure or by expensive filtered purging facilities.

When a decision was made to seal the disk enclosure, two fundamental design questions had to be resolved. First, the bearing, windage, and electrical heat must be removed solely by conduction through the enclosure walls, and second, the contents of the enclosure cannot be serviced in the field because that would lead to particulate contamination of the internal space. Heat removal is not a serious difficulty, providing the electrical dissipation is reasonable, because the bearing and windage losses of a 14-in diameter disk at 3,000 rpm amount to only about 30 W.

The servicing problem however, is more challenging. An assembly containing a precision spindle, disk recording heads, head actuator, preamplifiers, and contamination control components becomes a very expensive spare part to replace if it cannot be serviced. It is, therefore, important to design for maximum reliability of the whole disk enclosure to reduce the frequency, and therefore the overall costs, of replacement. The reliability of the head and disk were known to be high, and the ability to land the head on the disk avoided the necessity of expensive and potentially troublesome head loading mechanisms. Spindle and filtration components can be designed to meet virtually any reliability requirement, and the preamplifier reliability is ample if its function is minimized. A configuration for the head actuator, however, would have to surpass the reliability of mechanisms such as the lead screw and detent in the Buslik design [2].

The disk store was required to operate without degradation over an ambient temperature range of 10 to 45°C and under vibratory conditions involving 0.5 g acceleration. It was realized that at the projected track density of 118 tracks per cm (300 per inch), it was impossible to register the head adequately with respect to written tracks by absolute mechanical precision alone. It was, therefore, intended to control the head position by servo control from tracks written on one disk surface, and to use a voice coil to drive the head carrier. This decision avoided the cost of absolute positioning devices such as detent mechanisms or optical sensing elements within the disk enclosure, but substituted the cost of writing the servo information on the disk, and also that of the head and channel to read the information.

The other main function required of the actuator mechanism was to support the head mountings at a fixed 389

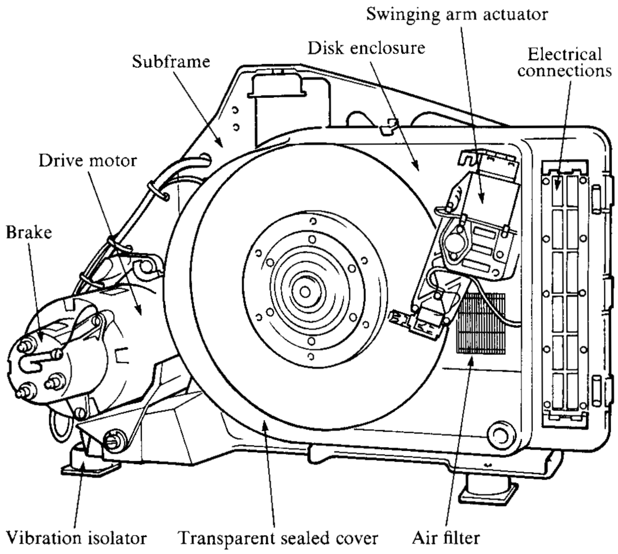

Figure 1 Configuration of integral disk enclosure designed for the swinging arm actuator.

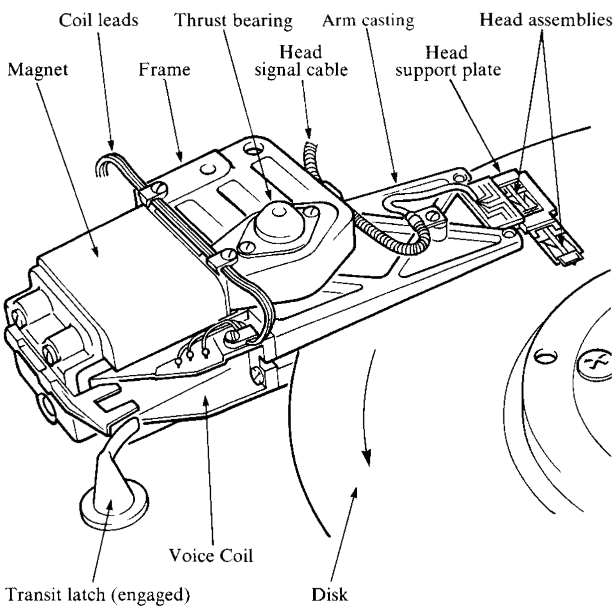

Figure 2 The swinging arm actuator.

distance from the disk surface while it moves radially across the disk under the power of a voice coil.

The main difficulty with the conventional wheel- or ball-supported linear carriage lies in the complexity of providing a single degree of freedom of movement with minimum friction. To reduce the complexity, a swinging arm actuator concept was proposed in which the heads move in an arc approximately radially across the disk. The concept was seen to offer several potential advantages besides reduced cost:

Table 1 Machine characteristics.

Capacity	9,308 million bytes
Data heads	2
Track density, nominal	118 tracks/cm 300 tracks/in.
Data tracks per head	303
Acess time:	
Track to track	14.2 ms
Average	72.5 ms
Data rate	889 KB/s
Rotational speed	2964 rpm
Bit density	2224 bits/cm 5650 bits/in.

1. Increased reliability through conservatively rated bearings and reduced bearing motion.
2. Low inertia.
3. Negligible friction and associated servo offsets.
4. A balanced assembly insensitive to translational machine vibration.

The concept was also seen to have two potential difficulties. The first was that read-write heads have a variable angular orientation to the local motion vector of the disk, i.e., yaw angle. The second was that structural coupling between heads and voice coil might introduce a resonance into the servo loop that could unduly restrict servo loop gain.

A machine based on this design concept was adopted as an integral part of the 3651 Retail Store Controller and subsequently of the 3661 Supermarket Controller, the 3791 Communication Controller and System/32. This paper discusses the nature of the design decisions for the machine configuration and describes the access performance.

Machine configuration

The configuration of the integral disk enclosure is illustrated in Fig. 1. The enclosure is mounted on a subframe at three points. The subframe supports the drive motor and brake and facilitates the removal of the disk enclosure by the service engineer.

A rigid aluminum casting is the major structural component of the disk enclosure, providing a housing for the spindle bearings, and a mounting for the head actuator mechanism, and forming one side of the sealed volume. An absolute filter is located in a passage in this casting through which the enclosed air is continuously circulated and filtered by the pressure difference existing between the inside and outside radii of the rotating disk

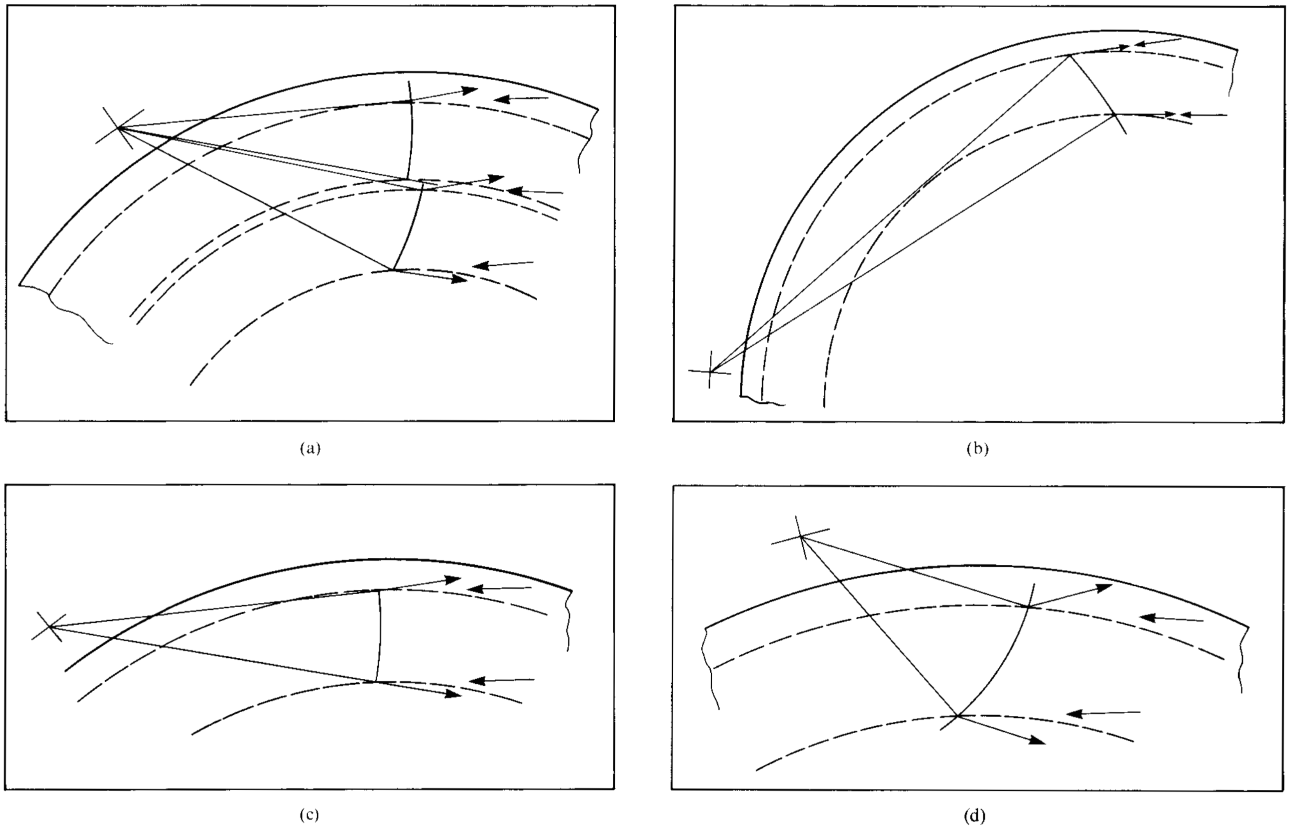

Figure 3 Head and track geometries for the swinging arm actuator. The recording head orientation and the ideal zero yaw orientation are indicated by the direction of the arrows. (a) Data surface. (b) Ideal geometry with long arm. (c) Actual outer data head. (d) Gross yaw with short arm.

[3]. The enclosure is completed by a molded plastic cover having receptacles on the outside for the circuit cards for write driver and read preamplifier.

The actuator illustrated in Fig. 2 carries three recording heads rigidly linked, two for the surface of the disk away from the base casting (both of which can read and write data) and one for the surface next to the base casting, which reads track positional information prewritten on this surface during manufacture.

This positional information is used to control the motion of all these heads at all times, including seeks between tracks. The method of encoding and demodulating the positional information is very similar to that described in Ref. [4]. The two data heads service two separate bands of 303 data tracks, each track formatted to store 60 sectors of 256 bytes per sector.

The principal characteristics of the machine are given in Table 1. The construction and characteristics of the recording head and its suspension are described in Ref. [1].

• *Actuator geometry*

The objectives in defining the geometrical dispositions of the basic elements of the actuator mechanism were to minimize the yaw angle between the read-write element alignment and a true tangential line on the disk, and to minimize the arm length. That these two requirements are contradictory can be easily seen in Fig. 3, in which the geometry finally selected is compared with two extreme cases of arm length. The long arm case is similar to that commonly adopted on phonographs, which are designed to minimize yaw in the direction of pick-up sensitivity with respect to a radial line, i.e., tracking error. A maximum yaw of only 0.2° could be obtained by adjustment of the arm length, and offset angle of the read-write element. However, as is discussed subsequently, the maximum bandwidth of the servo is dramatically dependent on arm length for practical, low-inertia arm structures. Early in the development program, when it was necessary to define the geometry, it was not clear what maximum yaw angle or arm length could be tolerated. It was known that yaw up to 15° reduces head flying height by a few percent and that effective bit density increases inversely as the cosine of yaw, but the effect on the life of the disk and head during starting and stopping in contact could not be determined until completion of extensive life tests. A decision had been made not to land the

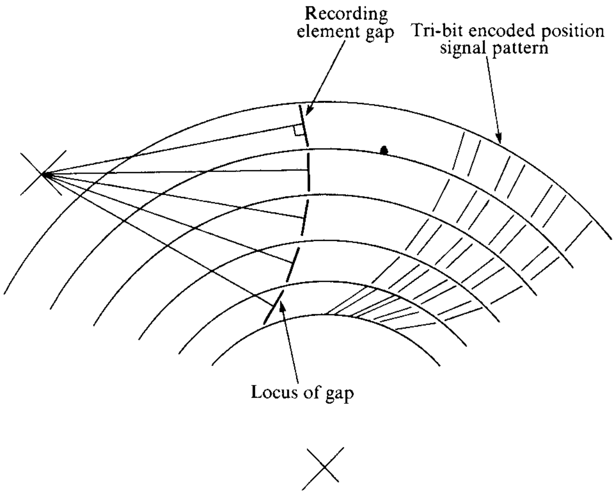

Figure 4 Effect of yaw on track and bit density for equal angular increments. The servo head is shown on an exaggerated scale.

heads on written data and that the landing zones should be on the inside of the data zones to minimize the resulting loss of recording surface area. A further decision was to locate the actuator pivot clear of the disk outside diameter to simplify the pivot design and maximize rigidity.

With these constraints a series of possible geometries was defined which demonstrated that the yaw of any head could be minimized by mounting it at an angle on the arm that gives equal and opposed yaw at the limits of movement. This was implemented for the data heads, but not for the servo head, because for the tribit position encoding method it is necessary to read well-defined clock pulses at intermediate positions between tracks. The requirement demands either that the read-gap alignment be tangential to the actuator pivot and that clock transitions be written on the disk as arcs of circles with center at the actuator pivot (Fig. 4) or that a variable timing reference be used in generating the written pattern to align the clock transitions. The latter was regarded as difficult to achieve. However, for the servo head, there exists the further freedom to define the disk radii between which it will swing because there is only one head on that disk surface, and it has the same swing angle as the data heads, which each cover only one-half of the surface. The best position for the servo head was found to be under the outer data head.

To fully determine the geometry the assumption was made that a yaw angle of about 14° would be acceptable from the standpoint of life, and the arm length was adjusted so that this angle would never be exceeded. The resulting geometry is defined in Table 2, the radii referring to the center of the read-write gap.

Constant track density with a swinging arm mechanism gives a variation across the disk of the angular increment of arm motion between adjacent tracks. This is undesirable because it introduces a variation with track position in the dynamics of a seek. However, because the length of the read-write gap projected in the radial direction decreases as the cosine of the yaw angle, it is possible to equalize all track-to-track angular motion increments while still maintaining a constant ratio of track width to track pitch across the disk. This is illustrated in Fig. 4.

• *Actuator structure*

Two requirements dominate the design of the moving parts of the actuator mechanism. The first is to maximize the ratio of available torque to inertia in order to give the shortest motion time interval for the track-seeking operation. The second is to achieve a structure permitting maximum possible bandwidth for the position servo loop, to give the shortest possible settling time after a seek. A high bandwidth is also desirable to minimize the position error of the servo head in the presence of vibrations and external forces. The bandwidth of a head positioning servo is limited by the need to maintain stability in the presence of mechanical structural resonances [4]. In conventional actuator mechanisms the force generated by a voice coil is applied through a more or less linear structure to most of the carriage mass, the primary resonant frequency normally occurring at 3000 Hz or higher. In the swinging arm concept the inertial load is distributed along the arm but can be regarded as a lumped mass at the head. This equivalent mass is easily made smaller than would occur in an equivalent linear actuator, but the rigidity of the connection to the voice coil cannot approach that of the linear actuator, because the structural connection is in bending- rather than extension-compression.

The resonant frequency of a mass-stiffness system is proportional to the square root of the stiffness, but the stiffnesss of cantilever beams of given cross section is inversely proportional to the cube of the length. There is little point in expanding these observations beyond noting that the resonant frequency is strongly dependent on arm length, because only a full analysis of practical designs can give meaningful guidance to the designer.

After the arm length was determined from geometrical considerations already described, it was necessary to design a structure with a desirable combination of stiffness and inertia. There are many ways in which an arm structure having acceptable dynamic properties might have been designed. The rationale adopted was to develop a symmetrical configuration, to design for low manufacturing cost, and to use finite element modeling as a mathematical tool to explore various structural forms.

Table 2 Geometric data for elements of the actuator mechanism.

	Head radius to pivot (mm)	*Outer data track radius (mm)*	*Inner data track radius (mm)*	*Maximum yaw angle (degrees)*
Head:				
Outer data	111.4	167.8	138.8	9.1
Inner data	119.7	132.1	104.8	11.5
Servo	107.0	168.2	142.3	13.6

Spindle-to-actuator-pivot dimension: 193.7 mm
Track-to-track angular increment 0.001445 radians

Table 3 Finite element model to define the arm structure.

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     ∇ STRUCTURE
[1]   YL←0.001 × 18 ¯18 18 ¯18 22 ¯22 27 ¯27 0, 0, ¯27 25 ¯26 0 20 ¯20 20 ¯20
[2]   XL←0.001 × (-89 89 57 57 25 25 19 19 19 15), 0, 11 32 26 65 96 96 120 120
[3]   COILMATERIAL ⍝ DEFINE FORMER/COIL COMPOSITE
[4]   1 2 5 6 BEAMS 0.002 BY 0.014
[5]   GEPHENOLIC ⍝ DEFINE COIL FORMER
[6]   1 3 2 4 3 5 4 6 BEAMS 0.0065 0.0065 0.013 0.013 BY 0.0023
[7]   5 7 6 8 BEAMS 0.01 BY 0.01
[8]   MAGNESIUM ⍝ DEFINE ARM CASTING
[9]   9 10 7 9 8 9 8 11 BEAMS 0.02 0.0026 0.012 0.01 BY 0.01
[10]  10 11 7 12 BEAMS 0.006 0.007 BY 0.009
[11]  10 12 11 13 BEAMS 0.006 0.004 BY 0.008
[12]  12 14 13 14 BEAMS 0.004 BY 0.007
[13]  12 15 13 16 14 15 14 16 BEAMS 0.003 BY 0.006 0.006 0.005 0.005
[14]  7 9 10 12 PLATE 0.002
[15]  10 12 14 10 13 14 12 14 15 13 14 16 FACETS 0.002
[16]  ALUMINUM ⍝ DEFINE HEAD SUPPORT PLATE
[17]  15 16 15 17 17 18 16 18 BEAMS 0.005 0.008 0.0033 0.01 BY 0.0018
[18]  15 16 17 18 MASSES 0.0005
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Statements 1 and 2 define the x and y coordinates of node locations in numerical order. Statements 3, 5, 8, and 16 define the material properties to be used the the finite element functions. The other statements cause stiffness and mass matrices to be built up from beam, plate, and triangular face finite elements. The left-hand argument identifies the model topography and the right-hand argument the section dimensions of each element defined by the statement.

Dimensions are in meters or millimeters.

The reason that symmetry was considered important is that it decouples resonant modes from interactions between the plane of symmetry and orthogonal planes. This has two benefits: first, it simplifies resonant modes to in-plane and out-of-plane modes so that there is some hope of understanding and dealing with them; and second, the number of resonant modes that are excited in, or orthogonal to, a plane of symmetry is minimized.

Unnecessary resonances were avoided as much as possible, because they might cause either a gain or phase disturbance in the dynamic response characteristic of the arm, thus threatening stability, or they might cause high vibration levels at the read-write head suspension, with consequent modulation of the head flying height.

Twofold symmetry was achieved by adopting two identical arms and giving them symmetry about the plane containing both their center lines. Further symmetry did not prove possible, however, because of the limitations of the casting process, which had to be used in manufacturing. The symmetry of the structure was sufficient to simplify the resonant modes to the point where the mathematical analysis would have to cover only one arm and the in-and-out-of plane motions could be handled separately. It was, therefore, considered appropriate to write a Finite Element Structural Analysis program in APL/360 restricted to the analysis of planar structures in order to model alternative structures [5]. The results of this analysis showed that the highest first-order resonant frequency in-plane could be achieved by adopting a wide, flat girder structure composed of members with reducing cross section at larger radii, and with a thin diaphragm to resist shear. An attempt was also made to raise the frequencies of resonances out of plane in order to minimize vibration levels at the head suspension. In a later evaluation the latter requirement was found to be more significant than was first thought, because reso-

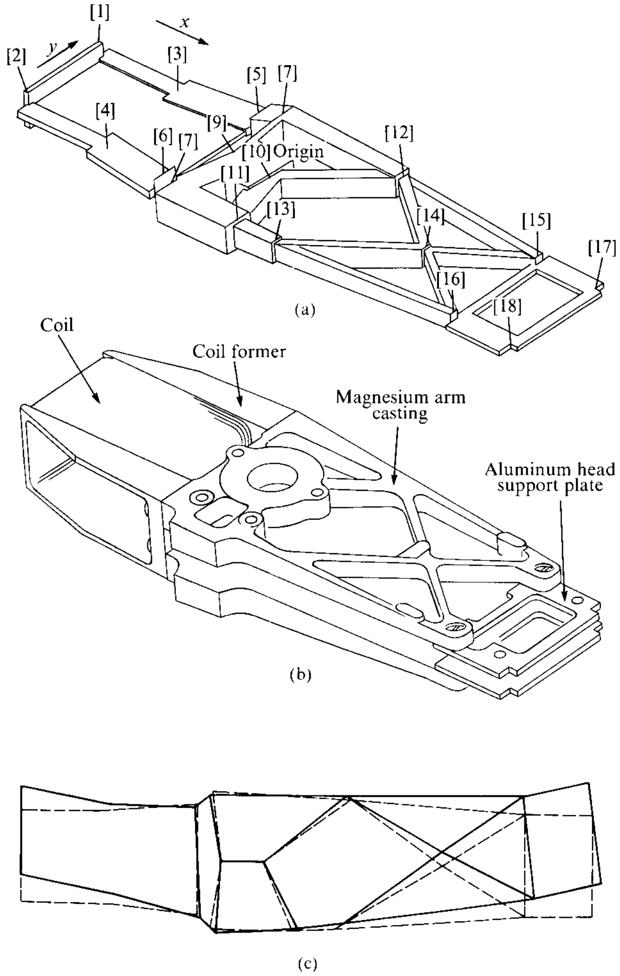

Figure 5 Diagram representing finite element model of arm structure. Results obtained from finite element model (a) Arm structure. (b) Details of actual structure. (c) Computed resonant mode shape at 1.88 kHz.

nances of the head-supporting structure coincident in frequency with resonances of the head suspension can pose a further threat to servo loop stability. This is due to the very high amplitude gains obtained in such a mechanical system. The finite element program was used to gain further understanding of the nature of the head-suspension resonances and to determine modifications of the arm structure to separate resonant frequencies.

The analysis also demonstrated the importance of integrating the voice coil rigidly into the arm structure. The design of the voice coil to optimize seek performance is described subsequently.

The component that emerged from the finite element modeling (Table 3 and Fig. 5) was clearly to be a casting, and to minimize inertia, a light-weight magnesium alloy was chosen. The two arms were cast integrally by an investment or diecasting process.

Operation of disk storage devices over a wide temperature range introduces misregistration of the recording heads with the written tracks because of differential expansions and distortions of the various components. The use of a track-following servo is a considerable asset in restricting the thermal effect to the disk itself and to the structure linking the heads together.

To control differential expansion between disk and head mountings, the heads were attached to plate subassemblies made of the same aluminum alloy as the disks. The attachment of these plates to the magnesium arms presented a problem, however, because the differential expansions would distort the plate between the fixing centers. To minimize the effect of this on the separation of the two data heads, the head support plate was formed as an open rectangle attached at two adjacent corners to the magnesium casting. The heads are mounted at two points, shown as holes in Fig. 5(c) that are as close as possible to the other two corners of the plate. The limited strength of the structure linking the outermost member to the attachment points permits its expansion to be largely unaffected by the rest of the structure. The side members can be regarded as links that at the head mounts divide by three the difference in expansion between the outer aluminum member and the inner composite member. Distortion between servo head and the opposing data head is minimized by maintaining symmetry of the two arm structures and head mounting plates.

The pivot bearings used in early models of the actuator were conventional angular contact ball races with axial preloading to remove radial clearance. When the mode shape of the primary resonance was measured, it was found that the arm structure was moving as a rigid body, but that displacement was taking place between the inner and outer races of the pivot bearings. When the radial stiffness of the bearings (5×10^5 N/m) was incorporated in the finite element model, it was found that the bearing stiffness was dominating the lowest mode of vibration and that an increase of bearing stiffness of about an order of magnitude would be necessary to give a primary resonance limited only by arm stiffness. The low radial stiffness of a ball race is a result of the Hertzian deformations at the points of contact between balls and raceways, and the stiffness increase required was found to be unobtainable with any reasonable increase of bearing size. It was, therefore, decided to adopt roller bearings with line contact to overcome this limitation, but this, in turn, raised a problem in removing radial clearance while retaining purely rolling contact. The eventual solution was to use needle roller bearings preloaded by a sprung insert incorporated in the shaft, as shown in Fig. 6. This arrangement gives a radial stiffness of about 8×10^6 N/m. Axial location was achieved by thrust pads of small diameter at the pivot shaft ends.

Needle roller bearings have a very high load bearing capacity in relation to their size. The dynamic capacity of the selected bearings is around 4450 N (1000 lbf) each. The preload force is a mere 22 N (5 lbf). The design margin on life is, therefore, astronomical. Grease lubrication is specified and the bearings are shielded to reduce to an absolute minimum the probability of bearing failure.

Figure 7 illustrates a typical actuator open-loop transfer function. The primary arm resonance is at 1.9 kHz which, as explained above, is a lower frequency than that of a typical linear actuator design. Its amplitude is also lower because of damping introduced by a plastic coil former. There is no doubt that with a purely metallic structure, the rise in open loop response at the primary resonance would exceed the zero-dB level and give cause for concern over loop stability unless phase could be controlled to give an adequate margin of stability.

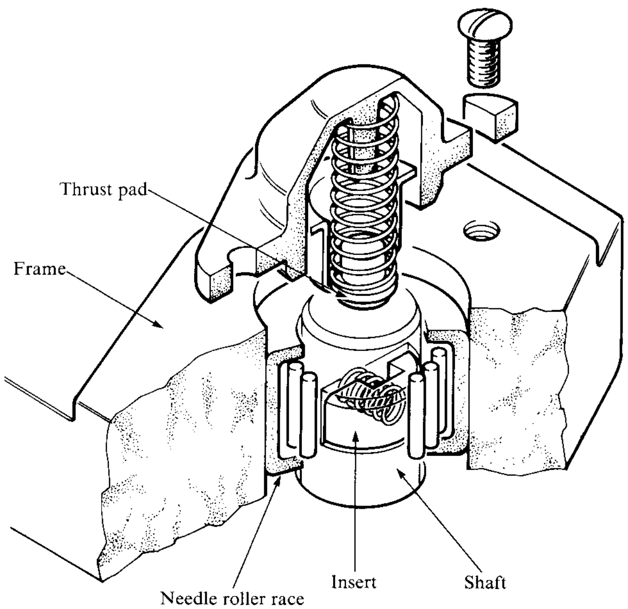

Figure 6 Pivot bearing, showing the needle roller bearings preloaded by a sprung insert incorporated in the shaft.

Access performance

It was believed that an average access time in the range 70 to 150 ms would be acceptable to most users, and an attempt was made to provide such performance at minimum cost by simplifying the electronic circuits.

In high-performance access mechanisms, the majority of the cost of achieving the required seek time is due to the voice coil drive transistors and the necessity for cooling; the velocity detection and control circuits; and the magnet assembly. The modest access performance required in this case appeared to be achievable economically.

The cost of drive transistor cooling was eliminated by restricting peak drive current to about 0.4 A, permitting the use of small transistor without heat sinks. The drive circuitry was simplified by winding a bifilar coil to avoid bridge driver connections or bipolar power supplies.

The access control method chosen was simply to accelerate for three tracks, to coast at constant velocity until a point four tracks from the destination, and then to decelerate to a lower velocity before position loop capture. Short accesses were dealt with as special cases. This method avoided the circuit complexity and cost of analog velocity profiling methods, but at considerable penalty in move time performance.

These two simplifications of the electronic circuits, together with the target access time, place a constraint upon the mechanical design of the mechanism to achieve a minimum torque-inertia ratio at a given current. The structure and therefore its inertia, was defined as described earlier in this paper; thus the minimum acceptable voice coil motor torque to be achieved from 0.4 A maximum current was defined.

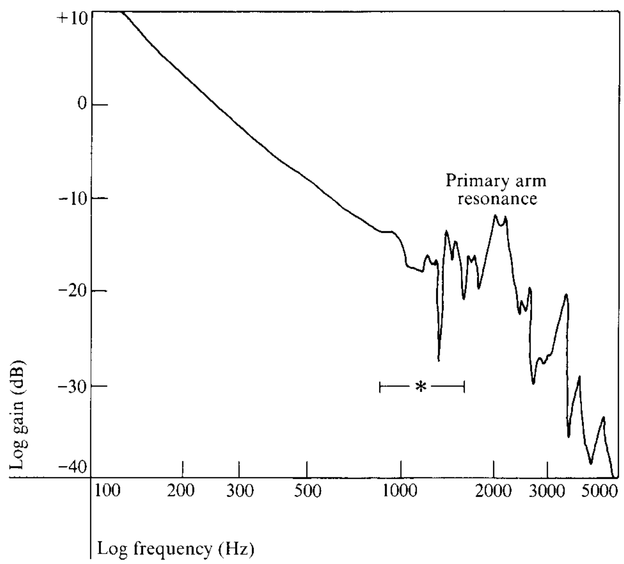

Figure 7 Typical measured open loop transfer function. Area between 900 Hz and 1700 Hz (*) shows effects of out-of-plane resonances and head resonances.

Voice coil

There are several ways in which a voice coil motor could be linked to a swinging arm, but in the interests of simplicity and reliability, it was decided to mount the coil rigidly to the arm. By locating the coil on the opposite side of the pivot to the heads, the moving parts would be balanced without an inertia penalty, and also

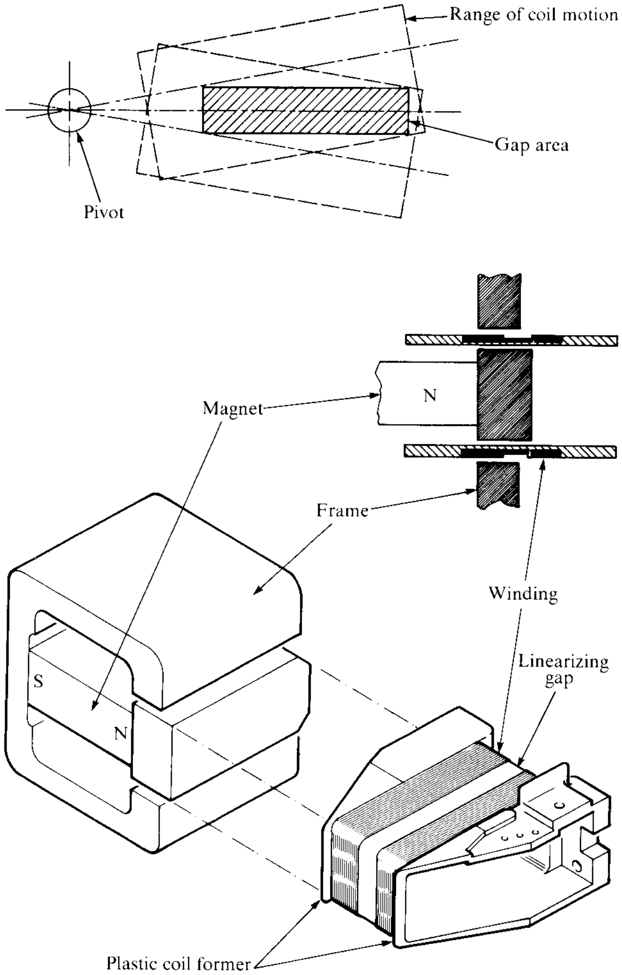

Figure 8 Configuration of swinging voice coil motor.

the magnitude of stray magnetic flux at the heads was minimized by the maximum separation.

With the coil constrained to rotate about a pivot, the conventional circular coil and magnet cross section would have to be toroidally developed to permit a close fitting, high-flux magnet, but the cost of manufacturing such a coil was clearly prohibitive. Therefore, a coil of rectangular cross section was chosen.

The criterion for the coil was a high ratio of developed force to conductor current for a given magnet-gap energy. This demands a large number of conductors closely packed in the magnet gap and can be achieved only by a winding process, for which conventional copper wire was chosen.

The use of a molded plastic bobbin avoided eddy current effects. However, the design was backed up initially with metallic bobbin and formerless alternatives, as there was some concern that dimensional stability and structural strength problems might be encountered with the plastics former. It turned out that glass-filled phenolic moldings not only had no major structural disadvantages but also had the important property, already mentioned, of introducing a substantial degree of damping into the arm structure. It is likely that without this damping it would not have been possible to close a servo loop around the swinging arm actuator at 300 Hz crossover frequency. This would have had a significant effect on the access performance over small numbers of tracks.

Table 4 Summary of voice coil motor and access characteristics.

	Product design	*Design study*
Torque constant	0.79 Nm/A	1.03 Nm/A
Linearity	± 2% over ± 9°	–
Coil resistance	55 ohms	17 ohms
Peak current	420 mA at 23 V	1.35 A at 23 V
Number of turns	2 × 440	340
Gap flux density	0,55 T	0,65 T
Magnet gap	3 mm	3 mm
Pole dimensions	10 × 50 mm	16 × 43 mm
Mass of magnetically hard material	0.32 kg	0.45 kg
Mean radius from pivot	57 mm	58 mm
Total inertia of moving parts	700 kg mm^2	700 kg mm^2
Nominal head acceleration	5.8 *g*	23.8 *g*
Average access time (101 tracks)	72.5 ms	
Move time (101 tracks)	65 ms	

Life testing of coil components showed that distortion of the assembly was acceptable over the life of the product, provided that the temperature was restricted to 72°C maximum.

There remained a basic choice between a long magnet-short coil or a short magnet-long coil configuration. The former gives maximum utilization of drive power because all coil turns are linked by flux, but at the expense of a larger magnet gap-energy and therefore increased cost of magnetic material and pole pieces. The short magnet configuration gives maximum utilization of available flux, but gives less force for given coil current or coil dissipation. The short magnet was chosen because it gives adequate performance at low cost.

A wide arm structure is necessary to raise the frequency of the primary resonance as high as possible. This width must be carried through into the structure supporting the coil and can conveniently accommodate a coil of the required length.

The final stage of the design of the motor was to select the motor proportions that would just satisfy the access time requirement while minimizing the cost of the magnet. This optimization was achieved by a combination of design analysis and model testing. The magnetic material chosen was the grain oriented, anistropic, high-energy member of the Alcomax family. The characteristics of the voice coil motor design are summarized in Table 4 and the configuration in Fig. 8.

Control of head motion during a seek operation is facilitated if the dynamic properties of the electromechanical system are the same at any head position. This places a constraint on voice coil motor design to make the torque constant of the motor independent of coil position. Because of the effects of leakage flux, a coil of finite length does not give a constant torque over its range of linking with the entire gap flux, and since the leakage flux internal to the magnet assembly is greater than the external leakage, the torque-to-current ratio shows a broad peak displaced inward from the central position of the coil. To linearize the motor, a section of the top layer of the winding toward the outside was skipped, improving the linearity from ± 6 percent to ± 2 percent across the working range of movement.

Potential access performance

The dimensions of the product motor design were chosen to minimize the cost of the magnet while keeping performance at an acceptable level. Larger magnets and coils can be designed, however, in which the balance of the moving parts about the pivot is still maintained.

To make a comparison of potential access performance between the swinging arm actuator and linear actuators, one should consider the performance of a design in which product cost has not been such a dominant factor as in the present design. A design study was made retaining balance of the moving parts, the same ground rules for working margins and worst cases, and having the same inertia, but in which drive current is limited only by the effect of coil heating on the dimensional changes in the coil assembly. This implies a limitation on conductor temperature of around 72°C. The coil is single wound, requiring a bridge or bipolar drive circuit and is, again, of the short magnet-long coil configuration. The performance of this high-performance swinging arm actuator is summarized in Table 4. The best basis of comparison between electromechanical access mechanisms is probably the acceleration of the head from rest, because this figure is independent of stroke length or access control method. This value is cited in Table 4 for nominal conditions, and is comparable with current high-performance access mechanisms such as that of the IBM 3350 Direct Access Storage. Relevant to the comparison are the relatively low mass of magnetic material required and the fact that only eight heads could be supported by the existing structure.

Other design features

The movement of the heads across the disk has to be limited by stops that must be capable of arresting the motion of the arm with minimum shock under runaway fault conditions. The overtravel of spring-loaded buffers must be minimized to be accommodated within the travel of the heads across the disk. This also reduces the loss of available recording surface. The area under the graph of deceleration versus time to the point of reversal in direction is proportional to the velocity at stop contact. Thus it is clearly advantageous to design a stop in which this graph is approximately rectangular, and this is achieved by preloading the spring. However, a rapid rise time to this deceleration pulse can excite the numerous high-frequency resonances of the structure, and in particular of the relatively delicate and vulnerable head suspension. The principle cause of rapid rise times is the impact between the stationary and the moving components and the associated rapid transfer of momentum. The transfer of this shock to the moving arm was avoided by mounting the springs on the arm assembly. Tapered leaf springs were used, reducing the mass at the contact points to a fraction of a gram.

Acknowledgments

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To translate an idea into an effective manufacturable device requires the application of a wide range of skills and talent. The contributions of R. S. Blackwell, M. H. Bosier, T. A. Hatt, J. D. Lipscombe, C. A. Pollard, and J. C. Troke were essential to the successful outcome.

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