A High Speed, Small Size Magnetic Drum Memory Unit for Subminiature Digital Computers

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A MEMORY with dimensions compatible with microminiature assemblies is required for future computers to be used in missiles and aircraft. A drum memory is described which can fulfill this need. The bit rate of 546 kc makes possible a 20-bit serial word time of the order of 40 microseconds. For a computer with add and multiply times of 40 microseconds, the drum memory described is adequate. Moreover, the technique described can be extended to provide a 20-microsecond word time by doubling the rotational speed of the drum, and to 10 microseconds or less by reading out two or more bits in parallel. A memory capacity of 15,000 twenty-bit words is available in the 7.4 \( \times \) 3.7 \( \times \) 3.7 inch total unit size, which is adequate for the type of computations usually made in an aircraft or missile. The advantage of such a drum memory as compared with a ferrite core memory, for example, is in cost, size, and ability to perform over wide temperature ranges. The disadvantage of the lack of immediate access to any address can for the most part be overcome by suitable programming precautions.

The magnetic drum development was performed under contract for Wright Air Development Center of the USAF to determine whether recording densities of 500 to 1000 bits to the inch and more than 30 tracks to the inch could be achieved in a small unit which would meet the requirements of MIL-E-5400, Class 2.

**General Description of Memory Unit**

- **Size**: 3.7 \( \times \) 3.7 \( \times \) 7.4 inches over-all
- **Power**: 400 cps, 3 phase, about 30 watts
- **Weight**: 11.3 pounds
- **Motor**: Mounted inside recording drum
- **Tracks**: 30 per inch, total of 122 tracks
- **Recording density**: 350 bits per inch using Manchester phase modulated recording
- **Clock frequency**: 546 kc
- **Total storage capacity**: 300,000 bits plus timing tracks and spare tracks

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Fig. 2 shows a partly assembled unit. To achieve the high degree of stability required for high density recording over a wide temperature and vibration range, an especially rigid unit was constructed. The framework and most critical parts are made of stainless steel selected to have a coefficient of expansion to match that of the ball bearings. A cross section drawing of the rotating part of the unit is shown in Fig. 1.

Fig. 1—Cross section of rotor assembly.

Fig. 2—Photograph of partly assembled magnetic drum.

The recording drum is made up of an internally mounted, 400-cps, 3-phase induction motor whose stator (1) is attached to a fixed shaft (3). The squirrel cage type rotor (2) is fixed inside a steel cylinder (8) which provides magnetic shielding and forms a mounting for a nonmagnetic stainless steel cylinder (9). This cylinder is plated with nickel-cobalt by an electroless method to form the recording surface. A shoe holding 27 read-record heads can be seen rest-
ing on the recording surface in Fig. 2. This shoe is loaded with a 6–10 pound force against the recording surface when the drum attains full speed. Since the shoe and 27 heads weigh less than 1.5 ounces, accelerations of 10 g's have little effect on the head spacing (which is maintained by an air film between the shoe and the drum). The shoe is positioned radially by means of pivoted arms. The pivots are held in V-grooves to eliminate any possible play.

A gear wheel can be seen which turns cam shafts mounted down the length of the four corners of the framework. The cam shafts take the pressure off the shoes for starting or stopping. A very small motor (not shown) will be mounted to turn the gear wheel against a spring when the drum has attained full speed. Upon the removal of the driving power, the spring will turn the large gear wheel and take the load off the shoes.

The shoes are self-aligning and no adjustments other than spring pressure are required. The use of two independent arms loaded by a single cantilevered spring achieves this self-alignment.

Positions for four large-sized shoes are visible in Fig. 2. On the other faces of the frame similar mounting spaces for smaller shoes are provided. These shoes are intended to hold both read and record heads for circulating registers.

The electrical characteristics are summarized as follows:

**Recording.** Peak currents of 100 ma are required for recording. The current is built up linearly during half a bit time for the Manchester type recording. A silicon transistor push-pull circuit with 6 volts on the collectors is used for the recording amplifiers.

**Reading.** The read signal is about 10 mv peak-to-peak at 546 kc and about 30 mv at 273 kc. No noise is noticeable on the signal under test conditions. Using Manchester or variable phase type recording, no transients are apparent beyond one recorded bit before and after each word. Pattern sensitivity has been eliminated by the use of narrow pole piece heads described later. Typical read signals made up of single eight-bit words are shown in Fig. 3.

### Selection of Magnetic Coating

For digital recording the head-to-drum spacing should be of the order of one-tenth or less of the length of the recorded bits to achieve customary margins of operations. For 350 or more bits per inch, a head-to-drum spacing of less than 300 microinches is indicated. For both temperature ranges of $-50^\circ$ to $+125^\circ$C and high shock and vibration, the small head-to-drum spacing required of 300 microinches cannot be maintained unless the head is made to bear on the recording surface. Gas lubrication is satisfactory for the maintenance of spacing in this range. For practical reasons it is desirable that a small particle of dirt (or accidental mistreatment during assembly or service) not do appreciable harm to the recording surface. This puts a requirement on the durability of the magnetic coating. For this reason magnetic plating is preferred to oxide films. There is an optimum plating thickness (which in practice turns out to be of the order of 100 microinches) for 350 bits per inch as is shown later. Since oxide coatings are usually ground after application, there would be an especially difficult problem in grinding them down to a uniform thickness of 100 microinches. Thus it became necessary to develop a suitable plating. Electroplated nickel-cobalt alloys have been tried and work perfectly well magnetically. They can be plated as thinly as desired and have been tested at thicknesses of 60 microinches and less. Mechanically this plating is not the best that can be obtained since it is not especially hard and has not been made to have both a high coercive force and adhesion strength comparable to the bulk material strength. This type of coating is magnetically satisfactory, but slight damage may put several 0.03 inch wide tracks out of operation due to local peeling of the coating. Nickel deposited by the Brenner electroless process forms a very hard coating which has excellent adhesion and hardness after suitable heat treatment. This coating markedly improves the wear resistance of almost any material that might be used to make the drum. A modification of the Brenner process to include cobalt produces an alloy which has good recording characteristics. This alloy is satisfactory magnetically without heat treatment but can be made harder with heat treatment.
DETERMINATION OF MAGNETIC PROPERTIES AND THICKNESS

The signal read from the recording surface will be

\[ E_{\text{peak-to-peak}} = 2\phi \omega \times n \times 10^{-8} \text{ volts} \]  

where

\[ \phi = \text{maximum number of flux lines in the head} \]
\[ n = \text{number of turns on head} \]
\[ \omega = \text{frequency in radians per second} \]

This assumes that the readback signal is essentially sinusoidal. The parameter \( \phi \) will be less than the flux lines remaining in the recorded dipoles after magnetization since not all the lines can be made to link the head. It will be proportioned to track width. It will be dependent on \( B_r \) and \( H_e \) for the magnetic coating.

For a thin magnet which is very wide, it can be shown that

\[ H = H_0 - \left( \frac{2t}{\pi L} \right) (B-H) \]  

where

\[ H_0 = \text{applied field} \]
\[ H = \text{effective magnetizing field} \]
\[ t = \text{plating thickness} \]
\[ L = \text{length of the recorded dipole} \]
\[ B = \text{magnetic induction} \]

A nickel-cobalt plating having a coercive force of 320 oersteds and a saturation induction of about 6000 gauss was selected. The ratio \( t/L \) can be varied so that a demagnetizing \( H \) just intersects the corner of the \( B-H \) loop for the material. Since \( L \) is fixed by the recording density, \( t \) is selected so that the residual induction is near the maximum induction, thus taking advantage of the squareness of the hysteresis loop of the nickel-cobalt alloy. A greater thickness would provide no greater residual flux because of demagnetization, but would require a greater recording magnetomotive force and would magnetize more slowly due to eddy current effects. Thus both magnetic plating material and its thickness can be optimized for the drum memory.

Fig. 4a shows an actual \( B-H \) loop of a nickel-cobalt plated film to show the effect of thickness on the residual induction due to demagnetization. A line is drawn of slope determined by \( t/L \) which intersects the \( B-H \) loop at the point of residual induction.

Fig. 4b shows a similar \( B-H \) loop for a heat-treated, nickel-cobalt alloy chemically deposited by the Brenner process. The squareness is not as good as that obtained by electroplating, but it is expected that this could be improved.

The \( B-H \) loops were taken on actual plated 2½ inch diameter by 4½ inch long stain steel cylinders before they were mounted on the drum assembly. (See Fig. 5 for photograph of the \( B-H \) tester.)

The \( B-H \) loops were taken by magnetizing the plating axially in a solenoid whereas recording takes place around the periphery of the drum. There was some doubt as to whether or not anisotropic effects would invalidate this measurement, and so several disks were plated and tested along various axes in the \( B-H \) tester. Very little change in \( B-H \) characteristics was noted as the direction of magnetization was changed. The disks were purposely ground so that the effect of grinding marks would be observed if they set up an easy direction of magnetization.

DESIGN OF SUITABLE READ-RECORD HEADS

The design goals called for 350 bits per inch re-
Recording density and at least 30 tracks per inch. Reading resolution of 350 Manchester cells per inch requires coupling as much flux as is possible from a 0.0014-inch-long magnetic dipole into a magnetic structure around which are wound a number of turns of wire. Coupling much of the flux requires a head gap of the order of 0.0004 inch and head to recording surface spacing smaller than 0.0001 inch. However, a compromise can be made which will cause a loss of signal but not necessarily loss of operational margins. Recording densities of more than 1000 bits per inch have been obtained in systems using a single floating head assembly. However, this usually is accomplished with very closely spaced heads and wider tracks than 0.025 inch. In the interest of economy and development time a compromise which utilized many heads mounted in a single air-floated pad was adopted. To make the construction problem easier, a head-to-drum spacing of 200–300 microinches was adopted. This limits a practical digital recording system to the region of 500 recorded bits to the inch. In the present system a recording density of 350 bits per inch is used, but this does not represent the practical system limit. The floating pads holding about 27 heads are of the order of 1.3 inches by 1.25 inches. Economies in space and cost are achieved by this mass mounting method which at present requires the use of recording densities of 500 bits per inch and less. The problems of recording and reading will be discussed separately although it is highly desirable that a compromise head be used which can both record and read. Apart from economy it greatly relaxes mechanical tolerance problems.

![Fig. 6—Ideal geometry for recording 350 bits/inch.](image)

**Recording**

Fig. 6 shows an idealized read-record head at its pole face. If the resistivity of the pole pieces were high so that eddy currents could be neglected, the amp turns required for recording and the read signal obtained per turn of the head winding could be quite closely calculated. Such a head is most difficult to make and the desirable spacing to the recording surface of 50 microinches or less is also most difficult to obtain in multiple head assemblies. The performance of the idealized head is of interest, however, for comparison with the compromise design which has been presently adopted but which clearly can be improved. To determine the recording amp turns required, let the $B-H$ loop (Fig. 4b) be assumed to be the $B-H$ loop for the recording surface. For the 0.00015-inch thick plating whose $B-H$ loop is shown on Fig. 4b assuming

$$L = 0.0014'$$

(350 bits per inch Manchester recording)

It can be seen from Fig. 4b that 500 oersteds are required to saturate the magnetic plating at 6000 gauss. From Equation (2) we find that

$$H_o = 900 \text{ oersteds approximately for 500 oersteds effective magnetizing force}$$

Two parallel lines are shown on Fig. 4b, whose intersections with the $B-H$ curve and $H$ axis give the residual flux density and the recording force required. This gives a flux density after magnetization of 3200 gauss. If the curve of Fig. 4a were used, magnetizing force of 600 oersteds would give a remnant density of 5500 gauss. However, because the electroplated coating is thinner (80 microinches versus 150 microinches), the remnant flux would be only 90 percent of that obtained for the electroless plating.

The overriding consideration for selecting the electroless plating was its hardness and resistance to wear.

The remnant flux for a recorded dipole 0.0014 inch long, 0.025 inch wide and 0.00015 inch thick would be about $7.7 \times 10^{-2}$ lines for a flux density of 3,200 gauss.

About 2.5-amp turns must be provided for magnetizing the plating. In the ideal head (Fig. 6) $14.5 \times 10^{-2}$ lines must be maintained across two gaps in series to saturate the coating at 6,000 gauss. The gap dimensions are 50 microinches in extent, 0.025 inch long and 0.0005 inch wide. This infers an average flux density in the air gap of 1,800 gauss, the maintenance of which will take about 0.36-amp turns.

The maintenance of flux in a very small continuous permalloy or ferrite circuit will take a negligible extra number of amp turns.

In practice, sufficient amp turns must be provided to generate a large number of fringing lines which form closed circuits around the side of the head and under and over the recording surface. If the ideal head as drawn in Fig. 6 were made, 3–4 amp turns would be sufficient for recording on the magnetic coating specified.

In practice, allowance has been made for the fact that the air gap may be 300 microinches instead of 50 microinches since this is much more readily achieved in a multiple assembly holding 27 heads. The best compromise for recording also includes making the silver shim gap larger than would appear ideal for small head-to-drum spacings since the flux density drops off rapidly in terms of the head gap dimension. A practical though not very efficient head would utilize 0.001 inch wide pole pieces with a 0.001 inch wide silver shim. (See Fig. 7.) Such a head records with 15-amp turns but gives a slightly greater read.
signal using 30-amp turns. Since these figures are large compared with the calculated 3-amp turns, it is clear that recording efficiency was sacrificed in order to make the head easier to fabricate and less sensitive to spacing than the ideal head. This inefficiency becomes important only if the recording circuitry becomes large or impractical. A head made to the dimensions shown on Fig. 7 has been driven at 546 kc with a silicon transistor circuit using 6 volts on the collectors and 100 to 200 ma peak current. Since this circuit is quite acceptable for a micro-miniature computer, recording efficiency can be sacrificed if this results in a net savings in manufacturing cost. The practical geometry of Fig. 7 clearly looks inefficient magnetically, but economy and ease of manufacture are in its favor. The 0.01 inch long legs are highly desirable for mechanical structure since a clamp holds the permalloy against the silver shim. The silver shim is wide for the size of the recorded dipole, but head spacing is far less critical than if the silver shim were closer to a more reasonable appearing dimension. Laminating the legs of the magnetic structure will improve the performance since penetration of the magnetic field at 546 kc is about 10 percent into either side of the material (assuming non-saturation) for the half amplitude point. In practice, excess drive is used which causes the penetration to be greater than the 10 percent mentioned above. The penetration is greater because the permeability of the material is lowered as it becomes saturated, resulting in an increased speed of propagation in the saturated region. The final choice of magnetic head is likely to be a compromise between the schemes shown in Figs. 6 and 7. For practical reasons, the dimensions shown in Fig. 7 make a good starting point for the development of a useful system.

The magnetic head structure is made of 0.001-inch permalloy rather than ferrite which would be too hard to handle in sufficiently small sizes. Under less than ideal conditions for recording, there are very marked transients where recording starts and stops, since some recording on a minor hysteresis loop takes place under the full region of the magnetic head. As recording density is increased without scaling down the head gap and head-to-recording surface spacing, this problem becomes more marked. For a chosen minimum head-to-drum spacing, the useful recording density can be greatly increased if the magnetic structure of the recording head is reduced to the smallest dimensions possible so that its influence does not appreciably extend beyond the recorded dipole. Care must be taken in using legs of small cross sectional area because there is not a large excess of flux over the amount required to saturate the coating. Flux leakage may prevent recording altogether unless the over-all head structure is kept very small.

Design of Read Head

It was shown earlier that a 150-microinch thick recording surface with the B-H loop characteristic of Fig. 4b would have $7.7 \times 10^{-8}$ flux lines at the center of a 0.025 inch wide recorded dipole. An ideal head would intercept these lines (and even increase the available flux by reducing the demagnetization). If the flux change were sinusoidal (any other wave form would give greater peak-to-peak volts) the read signal would be

$$E = \phi \cos \omega t \times 10^{-8}$$

where $\phi$ is the total flux in the recorded magnetic dipole and $E = 2\phi \cos \omega t$ peak-to-peak volts per turn of the reading head. At 546 kc, which is the maximum frequency used, a signal approaching 5.2 mv per turn could be expected from an idealized structure. With this ideal structure, it would be easy to determine that resolving signals at a much higher density would be possible and thus it would most likely be used at a density where it would give much less than the theoretical maximum signal. The magnetic head tested with the memory system described falls far short in obtaining the maximum obtainable signal at maximum density. In fact the presently used heads develop a signal in the range of 12 mv peak-to-peak at 546 kc as against a possible 780 mv calculated for a 150-turn head. Reference to Fig. 7 indicates that unlike the situation in Fig. 6 where more than half the flux would couple the head windings, only a small part of the flux will be useful in generating a read signal. Calculation of the exact magnetic flux coupling in this situation is most difficult, but a glance at a scale drawing makes the finding of $1/66$ of the possible signal quite plausible. The fact that the low output is tolerated is a compromise between signal level, and economy and ease of manufacturing the heads. Since an excellent signal-to-noise ratio and margins in clock pulse timing are obtained in this situation, the compromise is quite tolerable.

At 273 kc, which represents the pattern 0 1 0 1 in Manchester recording, the read signal obtained is about 30 millivolts in comparison with a possible 390 millivolts if all the flux in the recorded poles interlinked the head winding. The loss of signal by a 13 to 1 ratio is explained by the presence of an air gap, which provides a substantial reluctance in series with the head structure, and also by the fact that the head

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structure itself does not have a zero reluctance. Fig. 8 shows the response of the read head versus recording density and indicates that the head shown in Fig. 7 is being used beyond its optimum density.

Fig. 3 shows signals read by the head with clock times indicated. As can be seen, the signals can be interpreted with adequate reliability since there is no noise or mistiming in evidence.

There is, of course, much room for improvement of the magnetic head; however, each improvement increases the difficulty of making the head and the increased cost must be balanced against the economic benefits of the improvement.

Construction of the Magnetic Head

Fig. 9 shows the essential detail of the magnetic head. In assembling these heads the lower part is insulated and slipped into an aluminum tube. The tube is compressed forming a subassembly which can be tested. The subassembly heads are clamped into a holder (Fig. 10) and fixed in place with a suitable high temperature epoxy resin compound. Two such assemblies are made with the heads staggered so that with the assemblies mounted 15 to the inch a track density of 30 per inch is achieved. The assemblies are then mounted in the shoes.

Principle of Operation of Head Support Mechanism

A rotating drum moves a considerable volume of air in its close vicinity even though the drum surface is quite smooth by normal standards. This phenomenon is due to a boundary layer effect. That is, air molecules which are immediately in contact with the drum tend to adhere to that surface. Due to the viscosity of the air, the air molecules immediately about this initial layer are dragged along and as the distance from the drum surface increases, the velocity of the air molecules which are dragged along decreases. With this concept in mind, it is seen that if a stationary surface which is curved to match that of the drum is held near the rotating drum surface, the air will be dragged between the two surfaces. Since the air will also tend to adhere to the second surface, there will be a drag or friction force as shown in Fig. 11. If this stationary surface is inclined to the drum surface so that the space decreases in the direction of rotation, the air which is dragged in is squeezed into a progressively smaller space as is shown schematically in Fig. 12. This squeezing effect is of course a compression process, and pressure forces normal to the two surfaces develop. If this second surface is held in place by a spring force of proper magnitude, it will be held off the drum to a distance where the fluid

Fig. 8—Typical response for magnetic read-record head.

Fig. 9—Essential details of the magnetic head.

Fig. 10—Partly assembled magnetic heads.

MECHANICAL DESIGN DETAILS

Principle of Operation of Head Support Mechanism

A rotating drum moves a considerable volume of air in its close vicinity even though the drum surface is quite smooth by normal standards. This phenomenon is due to a boundary layer effect. That is, air molecules which are immediately in contact with the drum tend to adhere to that surface. Due to the viscosity of the air, the air molecules immediately about this initial layer are dragged along and as the distance from the drum surface increases, the velocity of the air molecules which are dragged along decreases. With this concept in mind, it is seen that if a stationary surface which is curved to match that of the drum is held near the rotating drum surface, the air will be dragged between the two surfaces. Since the air will also tend to adhere to the second surface, there will be a drag or friction force as shown in Fig. 11. If this stationary surface is inclined to the drum surface so that the space decreases in the direction of rotation, the air which is dragged in is squeezed into a progressively smaller space as is shown schematically in Fig. 12. This squeezing effect is of course a compression process, and pressure forces normal to the two surfaces develop. If this second surface is held in place by a spring force of proper magnitude, it will be held off the drum to a distance where the fluid

Fig. 11—Schematic view of drum with shoe in parallel position.

Fig. 12—Schematic view of drum with shoe at angle to develop wedge of lubricant.
pressure force equals the spring load force. When such a condition exists, the layer of fluid which separates the two surfaces is referred to as a hydrodynamic lubricating film and such surfaces which react in this manner are referred to as a self-acting bearing. In the example given, air is used as the lubricating fluid; however, any fluid, liquid, or gas which will adhere to the bearing surfaces without causing damage will perform in this manner.

The theoretical aspects of this phenomenon were first proposed by O. Reynolds about 75 years ago, and solutions of his equation for the incompressible lubricating films have been well accepted in the literature on bearing lubrication. In recent years considerable attention has been directed toward the case of the compressible or gas lubricating film for many promising advantages such as chemical stability, extremely low friction, the maintenance of close clearance between moving parts, and the use of the ambient gas as a lubricant. The technology of the analysis of the bearing using a compressible fluid as a lubricant as in the example (Fig. 12) is quite involved and beyond the scope or purpose of this paper. Work on this phase for use in the design of such bearings and shoe surfaces as the attack angle on this phase for use in the design of such bearings and shoe surfaces as the attack angle on this phase for use in the design of such bearings will be described below. For the purpose here, let us denote the angle between the drum and shoe surfaces as the attack angle \( \alpha \), the edge farthest from the drum surface as leading edge, and the edge nearest the drum as the trailing edge. Drum rotation is in the direction from the leading edge towards the trailing edge.

The gas lubricating film is not only unique for its thinness but also its high spring rate. The gas lubricating film supported shoe possesses certain unique properties which make it a useful device for the support of a recording head. The most important properties will be described below. For the purpose here, let us denote the angle between the drum and shoe surfaces as the attack angle \( \alpha \), the edge farthest from the drum surface as leading edge, and the edge nearest the drum as the trailing edge. Drum rotation is in the direction from the leading edge to the trailing edge.

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Properties of Lubricating Film Supported Shoe

The lubricating film supported shoe possesses certain unique properties which make it a useful device for the support of a recording head. The most important properties will be described below. For the purpose here, let us denote the angle between the drum and shoe surfaces as the attack angle \( \alpha \), the edge farthest from the drum surface as leading edge, and the edge nearest the drum as the trailing edge. Drum rotation is in the direction from the leading edge to the trailing edge.

The gas lubricating film is not only unique for its thinness but also its high spring rate.

![Fig. 13—Mean pressure vs. trailing edge to drum space.](image)

Fig. 13—Mean pressure vs. trailing edge to drum space.

Fig. 13 shows the typical relationship of the pressure force which can be developed under typical operating conditions. It will be noted that at the operating conditions shown in the figure, a mean pressure of 1.5 psi gage at a trailing edge spacing of 400 micro-inches is obtained. As the trailing edge spacing is decreased, the mean pressure increases at a rapid rate so that at a spacing of 200 micro-inches the mean pressure has increased virtually four fold or inversely with the square of the spacing. This characteristic is most desirable from electrical and mechanical points of view for recording drum applications. For any fixed design as the load is increased the shoe and hence recording head to drum spacing is decreased. This is, of course, helpful to the electrical performance as far as output signal is concerned. As a greater load is applied to the shoe, the ratio of the applied load to the weight or inertia of the shoe and its associated mechanism is increased. When this ratio is increased, the ability of the shoe to withstand accelerations and run-out-irregularities of the drum is also increased.

In the case of the drum which is the subject of this paper, the effective area of the shoe is 1.6 square inches and its normal operating load is 10 pounds, which gives a mean pressure of about 6 psi. The shoe with recording heads in place has an effective weight of about 1.5 ounces, and so the load-to-weight ratio is slightly over 100. Since at this operating condition, slight changes in the spacing result in a considerable change in the lift force, there is available a large force to restore the proper head to drum spacing. Let us consider an example at the conditions cited above.

In a broad sense, since the curve shown in Fig. 13 is one of a force vs. displacement, the lubricating film may be regarded as a spring of variable rate. If the displacements are left small, the lubricating film may be approximated by a linear spring and the slope of the curve may be taken as the spring constant. For the conditions cited above, this slope or linear spring constant is about 100,000 pounds per inch for a show of the given effective area. The spring rate of the spring used to produce the load force would have to be added to this rate; however, since this spring would have a rate of about 100 pounds per inch, it is virtually insignificant in its effect on the natural frequency of the system of forces acting on the show. The spring rates of 100,000 pounds per inch acting on the effective mass of the shoe give a resonant frequency of more than 3000 cycles per second. Thus it follows that such a mechanism is quite capable of withstanding accelerations of 10 g's up to 2000 cps without seriously affecting the output electrical signal.

Another unique property of the floating shoe is its inherent stability. Fig. 14 shows a typical pressure distribution between the trailing and leading edge of the shoe. As the attack angle \( \alpha \) is increased, the center of pressure shifts towards the trailing edge, and similarly as the angle \( \alpha \) is decreased the center of pressure shifts to the leading edge. Let us fix a certain shoe...
geometry and allow the shoe to pivot about an axis at the center of pressure and parallel to the drum surfaces. Now if the shoe is tipped so that the angle $\alpha$ is increased, the center of pressure moves toward the trailing edge. This action develops a turning moment on the shoe. The turning moment is in the direction required to return the shoe to the original position. Similarly, when the shoe is tipped to an angle less than the stable angle, a turning moment of opposite sign develops to return the shoe to the original stable position. From experience it has been found that the system has sufficient damping to make it stable. Thus it follows that the location of the pivot axis is not critical, for the shoe will tend to seek a value of the angle $\alpha$ so that the action line of the center of pressure will pass through the pivot axis.

**Design Requirements**

The design of a mechanism to make use of the lubricating film supported shoe or for keeping a recording head in proper location with respect to the drum recording surface requires careful attention to the precision requirements of the mechanism. The development of a design framework which requires a minimum of very precise parts which are amenable to precision manufacturing techniques is necessary to the successful execution of the task. It is not only necessary to have surfaces which are geometrically true, but it is also required that the proper geometric relationship between the various parts be accurately maintained. The most important of these relationships is the alignment between the shoe and the drum. It is essential that the center of curvature of the shoe be maintained parallel to the axis of rotation of the drum. The limits of accuracy required are dependent upon the particular design and the performance required. For the design the out-of-parallellism is kept to less than 3 parts in 10,000. The other important requirement is that the load on the shoe be uniformly distributed so that tipping does not occur. As will be shown later, the load on the shoe of the subject drum is applied at two points. The difference between these forces is kept to a value less than 7 percent. The tolerances given above are those used in the design of the drum with due allowance for possible manufacturing tolerance and also the expected deflections of the mechanical parts.

Figs. 2, 15, and 16 show the drum in various stages of assembly. It will be noted that the rotating portion of the drum is set into a very rigid frame, and access to the drum-recording surface is through appropriately located cutouts in this frame. A V-groove is machined into the sides of this frame so that it is accurately parallel to the axis of the drum. Guide slots for radius arms are machined at precise right angles to the V-groove. Each of the radius arms are provided with polished sapphire pivot pins which are cemented in place in an assembly fixture. The center-line distance between the pins is accurately maintained so that it is virtually the same for a given pair of arms associated with a given shoe. One pin of each arm operates in the V-groove of the frame, while the other pin operates in a V-groove in the shoe. The V-groove in the shoe is located in the line of action of the center of pressure, and it is made accurately parallel to the axis of the cylindrical surface of the shoe. To prevent smearing of the pole pieces of the recording heads, the curvature of the shoe is ground by means of a contoured abrasive wheel so that the lay of the grinder marks is parallel to the head gaps. Final finishing is done on a cylindrical lapping tool which has a diameter 0.1 percent greater than the drum. The load for the shoe is supplied by the spring which is adjusted by a single centrally located screw. By this means, equal forces are applied to each side of the shoe. The load forces the pins to seat in the V-grooves of the shoe and frame and precisely locate the shoe with respect to the drum so that the axis of the drum and shoe are parallel within the extremely close limits previously cited.

Special consideration must be given to start and stop conditions, for without sufficient drum speed the lubricating wedge or film cannot develop and a high-
friction condition will exist. To prevent this, it is necessary to unload the shoe and lift it slightly off the drum surface until sufficient speed for normal operation is attained. For stopping the drum, the procedure is reversed. There are basically two methods by which this may be accomplished. One method involves removing the spring load until operating speed is reached. The second method involves introducing lubricant under pressure through a very small hole in the shoe into the space between the shoe and drum. If sufficient lubricant (in the subject drum it is air) is supplied, the shoe will be lifted off the drum surface. After operating speed is reached, this supply of air may be shut off and normal operation resumed. This latter method requires the use of an air compressor, a fact which makes it somewhat unattractive for airborne use. The first method is used in this drum design. It will be noted that in Fig. 16 the radius arms extend from the side of the frame which has the V-groove to the opposite side.

Fig. 16—Shoe and radius arms.

At this side of the frame, the ends of the arm can ride on a simple eccentric cam which is operated by the small gears. During normal operation, these ends of the arms are free of the cam. For offspeed operation the cam is rotated to a position where the ends of the arms are lifted. Since the mechanism is extremely rigid, a movement of less than one mil of the end of the arm is sufficient to transfer the spring load from the shoe to the cam. In this condition, the lubricating film between the drum and shoe must support the weight of the shoe. Since the weight of the shoe is very much less than the operating load, the resulting friction is negligible. If the magnetic coating is very durable, the slight contact between the shoe and drum under these conditions is not serious and may be eliminated completely by operating the drum with the axis in a vertical position. When the shoe is in this free condition, a state of instability may develop if the cam is inadvertently set to lift the ends of the arm too high. Should this condition develop, serious damage to the drum and shoe surface will occur. To eliminate this possibility, one arm of each pair for a given shoe is provided with a spring-loaded pin as shown in Fig. 17. This pin is allowed to act upon the side of a shoe to cause a small amount of friction damping. Since the load at which the shoe is operated is much higher than the weight of the shoe, this damping friction does not affect the operation any noticeable amount.

The main frame, as almost all other parts of the drum assembly, is made of a precipitation-hardening stainless steel. For the sake of rigidity and precision, it is fabricated from one piece of stock and provided with generous ribs.

Fig. 17—Typical pair of radius arms.

REFERENCES


ADDITIONUM

In the body of this paper, it has been stated that as recording density is increased without scaling down the head and drum coating...
geometry, recording on a minor hysteresis loop takes place under the full range of the head. Since present conventional memory drum design does not follow the practice of scaling down the entire recording geometry with increases of recording density, it would appear appropriate to describe here two series of experiments which yield data supporting the above statement.

In the first series of experiments, a recording geometry shown in Fig. 7 in the body of this paper was used. It was found that for a given set of operating conditions there was an optimum head drive current for maximum output signal. This phenomenon can be explained by the demagnetizing effect of the fringe flux upon the adjacent dipoles, which is increased in strength by the increase in drive. Saturation of the head pole pieces and drum coating can not be the case because saturation would yield a limiting effect and not a maximum point of operation.

In the second series of experiments, a typical small ferrite head with pole pieces about 25 mls square was used which gave a head length of slightly more than 20 mls. The head to drum spacing and all other conditions were the same as in the first series of experiments described above. At a recording density of 109 bits per inch, the output signal was about 40 millivolts and there was little or no tendency to pattern sensitivity. When the recording frequency was increased to give 350 bits per inch, the output fell to 6 millivolts and the pattern was distorted to the extent that errors would be caused in reading it. It is believed that this pattern distortion and signal attenuation is due to an anticipation effect. That is to say that as each dipole starts under the leading edge of the pole piece, some of its flux passes into the pole piece and gives a read out signal. In these experiments, it was possible to correlate exactly the output signal waveform with the recorded bit length and the size of the pole piece.

In one case the drum was demagnetized and then a single 8 bit pattern was written on it. Upon reading back, the entire pattern was read three times: first as the leading edge of the head pole point passed over the dipoles, secondly with larger amplitude as the dipoles passed under the gap, and once again as they left the trailing edge of the pole piece. In an actual computer application, if this anticipation effect occurs, the true output signal may be so distorted that serious errors may be introduced or the effectiveness of the system considerably reduced.

It appears that from a practical point of view for an adequate design compromise for freedom from pattern sensitivity and signal level output the head dimension measured in the direction of drum rotation should not exceed the length of the recorded dipole by an appreciable amount.

**DISCUSSION**

W. N. Papian: I wonder if you could tell us what the status of the project is right now?

Mr. Howard: This was a research and development effort to determine the feasibility and the basic design requirements of a drum of this type. We have built several test models and the model shown, which is in the form factor suitable for use in a micro-miniaturized computer. At the present time, it is running and undergoing environmental tests. We are still working on it, and we believe it to be a practical device. If our hopes materialize we shall wrap a computer around it.

M. J. Haima (IBM): How was the head to drum spacing and angle of attack measured?

Mr. Howard: It is very difficult to measure. The curve shown is for calculated values and has been verified by laboratory measurements. We have verified the general shape of the curve by at least two methods. One method made use of electrical capacity between the drum and small pads set into the shoe. The other method measured the lift distance of the shoe off the drum surface by means of sensitive dial gauges and special electrical capacity probes.

D. Kullen (Oliver Shepherd Industries): What is the peripheral speed of the drum, and what is the head inductance?

Mr. Howard: The drum is 2.5 inches in diameter; the synchronous speed is 12,000; and actual speed, allowing for motor slippage, is about 11,600. This gives a peripheral speed of about 120 feet per second. The head inductance is 60 microhenries.

W. G. Doose (MIT): What magnetic shielding do you have between motor windings and drum and readers, or is stray motor flux not a problem?

Mr. Howard: It would be a problem if you didn't take care of it. The motor is surrounded by a magnetic shield, and the magnetic recording material is plated on a separate sleeve which is shrunk on the motor drum assembly.

K. Enslin (Brooks Research): The storage density in your system is approximately 2,500 bits per cubic inch. Could you discuss the relative advantages of magnetic cores and your drum for the application at hand?

Mr. Howard: I am in no condition to do any mental computing up here. But, as for comparing, you can get a cheaper bit stored on a drum system than you can in a core. Of course, the disadvantage of the drum is the access time. By suitably programming a fixed program and using circulating registers or revolvers to bring the words up as required, you can cut this down. One of the main advantages is the possibility of extending this concept to a system with extremely large capacity. I don't think that cores could take the punishment that this thing will.

D. Roberts (Librascope): What air pressure range will the drum operate under? Is it pressurized?

Mr. Miller: A lot depends on the loading and hence spacing at which the shoe is going to be operated. We have operated the unit up to a pressure altitude of 30,000 ft. Since equipment like this should be operated in a dust free atmosphere, we do not feel it is a great disadvantage to the design if it is kept in a pressurized container. When it is finally installed, it will be in a pressurized box filled with some inert and dry gas such as nitrogen.

P. Smith (Gen'l. Transistor): What keeps the shoe from the drum before the drum gets up to speed.

Mr. Miller: The four small gears you saw in the slide are attached to ends of eccentric cams. During the start operation, the cams are rotated into a position in which the spring load is carried by the cam. There is also a spring loaded pin in the arms to help keep the shoe off the drum during the start operation. When the operating speed is achieved, the cams are rotated and the spring load is transferred to the shoe. During the first moment of starting there may be some tendency for the shoe to contact the drum; however, sufficient lift is developed at 500 rpm. To prevent damage during this early part of the starting cycle, we have used a very durable hard electrolestic plating of a nickel cobalt alloy.

J. Russell (University of Calif.): What is the peak to peak read signal at your bit density?

Mr. Howard: 12 millivolts.

G. G. C. Randa (IBM): What is resonant frequency of shoe and its spring mechanism?

Mr. Miller: If you will recall what the curve of pressure vs. spacing looks like, it is, of course, not linear and so does not really yield a resonant frequency as such. If we consider only small displacements and operate in the 200 microinch range, we find that the spring constant is about 100,000 pounds per inch. Since the shoe and arms have an effective weight of about one tenth of a pound, the resonant frequency would appear to be about 3000 cycles per second. With induced vibrations up to 500 cycles per second no malfunction of the shoe support mechanism seems to occur. However, at the higher frequencies, say at 2000 cycles per second, malfunction is impending and so some small amount of vibration isolation may be required in the final installation if the supporting structure does not attenuate these frequencies sufficiently.

P. Skelly (RCA Service): Are there any temperature controls used?

Mr. Howard: No, None at all.