

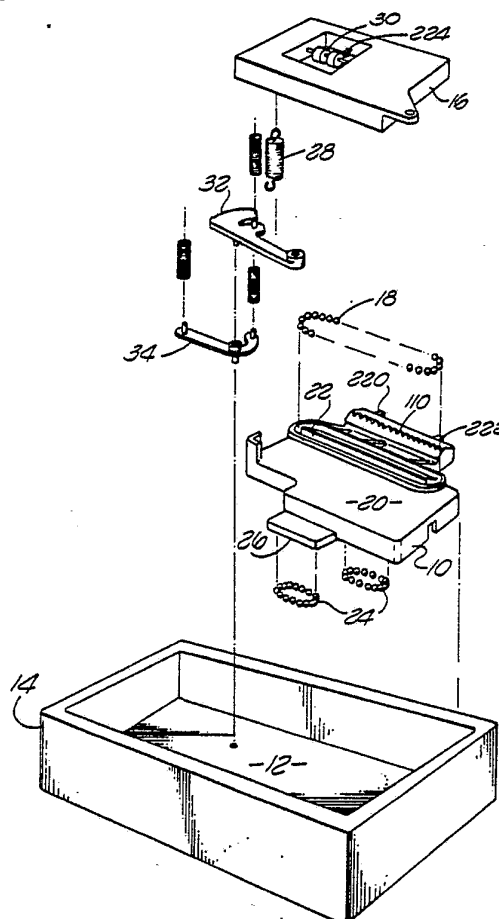
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(54) Title: MAGNETIC TRANSDUCER POSITIONING APPARATUS

(57) Abstract

A movable carriage (10) carrying magnetic transducer means is positioned at predetermined points adjacent a floppy disc surface for appropriate reading/writing of data thereon. The transducers are positioned by a permanent magnet stepping motor (30) which rotates a cam shaft (84). The cam surfaces (116) on the shaft drive a plurality of balls (114) into engagement with opposed racks (110, 112) defining 'V'-shaped notches (161-180) of different pitches formed on a surface of the carriage and on a fixed surface adjacent thereto. As the balls (114) sequentially enter the 'V'-shaped notches, (161-180) they force individual pairs thereof into alignment, the carriage (10) and thus the transducer means are sequentially and discretely positioned at predetermined data track positions on the floppy disc.



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MAGNETIC TRANSDUCER POSITIONING APPARATUS

5

Field Of The Invention

This invention relates to apparatus for positioning magnetic transducers in data storage systems and more particularly to such apparatus for precise location of magnetic transducers with respect to rotating flexible magnetic recording media (floppy discs) for reading/writing data thereon.

15

Background of the invention

The most common positioning system used in prior art expensive drives, is the moving coil system in which a multi-turn electromagnetic coil is mounted on the carriage and positioned in a flux gap of a fixed, permanent magnet structure. By controlling the polarity and amplitude of current flowing through the coil, a thrust of controlled magnitude and direction is generated. A second element of this system is a position encoder attached to the carriage which is usually optical, but occasionally inductive. Electronically controlled forces generated by the electromagnetic coil are used to guide the carriage to the



positions indicated by the position encoder.

5 Most of these systems also require a velocity
transducer to sense the velocity of the carriage.
Because of the many elements of this closed loop
system, it is quite expensive and therefore it
is not economically competitive for very low cost
10 positioners such as those used in the typical
floppy disc drive.

In low cost drives, open loop systems
are preferred, in which a positioner moves di-
15 rectly to inherent, predetermined discrete posi-
tions with no electronic verification that the
desired position has been reached. Such systems
cost little and work well provided their torque
and inertia limits have not been exceeded.

20 The simplest of these open loop systems
is a linear stepper. One version in commercial
production utilizes a long cylindrical armature
of soft iron attached to the carriage upon which
25 the transducers are mounted. Into this armature
are cut circumferential grooves, 16 to the inch.
Around this armature, and separated from it by
only a narrow air gap, is an electromagnet coil
structure with soft iron pole pieces with matching
30 circumferential grooves, 16 to the inch. When a
current is applied to this coil, the system becomes
a variable reluctance motor which seeks the posi-
tion of least reluctance and the armature and car-
riage are moved until the grooves are aligned.
35 A second coil structure, identical to the first,
is positioned 1/48 of an inch out of phase with
the first, and the third identical coil structure

5 is positioned 1/48 of an inch out of phase with
the first but in the opposite direction from the
second. By turning on the current to the second
coil and turning off the current to the first, the
armature will be induced to move 1/48 of an inch.
10 By turning on current to the third coil and turning
off the second, the armature will move an additional
1/48 of an inch and so forth. By properly sequen-
cing the current to the three coils, this linear
stepper system can be directed to any of the 48
15 positions per inch defined by the three coil
structure.

One drawback of linear steppers of the type
above described is that the armature material must
be chosen for its magnetic properties rather than
20 for the desired rates of thermal and hygroscopic
expansion. Consequently, the pitch of the linear
stepper varies with the materials expansion and con-
traction rates as the ambient temperature and humi-
dity changes.

25 Another problem is that the electromag-
netic coils must be positioned in close proximity
to the armature, and the substantial heat generated
by their operation expands and distorts both their
30 supporting structures and the length and pitch of
the armature. When one coil is turned on for any
substantial length of time, for example to hold the
transducers in a desired position, that portion of
the armature adjacent the active coil is being heat-
35 ed while other portions are cooling. When the motor
is sequenced to another position, another coil be-



5 gins heating a different portion of the armature,
and the motor changes its thermal and dimensional
nature over a period of time.

10 In a motor such as this, the air gap must
be wide enough to prevent even momentary physical
contact between armature and stator. In addition,
the carriage guidance system must have great lateral
stiffness, for as the armature deviates from a cen-
tral balanced magnetic position, magnetic attrac-
tions decline slightly on the wide gap side, but
15 increase greatly on the narrow gap side. This inc-
reasing imbalance pulls the armature even further
off center. To oppose these forces requires a stiff
armature, very stiff guidance means, and a suffi-
cient air gap.

20 To achieve reasonable efficiency, the size
of the teeth in the magnetic structure must be propor-
tionally bigger than the air gap. As the teeth be-
come very small relative to the air gap, their inter-
action across the air gap becomes negligible. Conse-
25 quently, linear stepper motors are not suitable for
fine pitch systems such as 96 increments per inch of
storage on the disc.

30 A further problem with linear stepper motors,
is the large mass of their armatures. Moving this
mass imparts substantial kinetic energy to the system,
which must be removed, primarily by mechanical friction,
before the system comes to rest. This requires a long
35 settling time while the carriage oscillates about the
desired position and comes to rest which causes rela-
tively long delays before the system is ready for data



transfer operations. Additionally, since the system
5 is not balanced, the heavy armature becomes a substantial burden to be driven "uphill", if the carriage travel is not exactly horizontal.

To overcome some of the foregoing problems, rotary stepping motors in conjunction with
10 some means to convert the rotary motion to linear motion have become widely used in the prior art as transducer positioners in low cost drives.

Because the magnetic structure is fairly
15 compact, narrow air gaps can be achieved. When the structure expands due to heat generation, the expansion is equal in all directions, therefore the output which is angular in nature, is not distorted.
20 Ball bearings provide a reasonably stiff support to resist unbalanced magnetic forces.

Rotary stepping motors have a number of limitations, among which are dead band and oscillation.
25 The static friction of the system defines a zone called the dead band, in which the motor has insufficient torque to initiate motion. Therefore, if the armature comes to a stop anywhere in the dead band, it will remain at that position.
30 To minimize this positional indeterminacy, the motor's torque must be maximized, and the static friction minimized. If the friction load is very low, then the energy stored in the moving armature causes it to oscillate about its terminal position for unacceptably long periods. To reduce
35 settling times to an acceptable level, a moderate amount of friction must be present, which in turn increases the dead band.



Another weakness in stepper motors used
5 as precision positioners is pole shift. Two kinds
of stepper motors are possible. The variable re-
luctance type utilizes magnetically soft iron struc-
tures in which magnetism is selectively induced by
electromagnetic coils. The permanent magnet type
10 utilizes magnetically hard material to permanently
induce magnetism in magnetically soft iron struc-
tures, while additional magnetomotive forces are
selectively generated by electromagnetic coils.
Ideally the permanent magnets in such devices
15 should be infinitely hard magnetically, and the
magnetically soft structures should be infinitely
soft magnetically. These ideals are not possible,
and in low cost motors, the choice of material for
magnetic structures is heavily influenced by cost,
20 and magnetic hardness is not the primary concern.
The magnetically soft structures, which should
ideally be manufactured of ingot iron or electro-
magnet steel, are more often, for cost reasons,
manufactured of low carbon steel. They are also
25 generally stamped and formed without subsequent
annealing. They are also frequently electroplat-
ed in baths that can add carbon to the steel.
For all of these reasons, the supposedly magne-
tically soft structures in low cost motors are
30 frequently moderately hard magnetically.

As a result, the magnitude and polarity
of magnetism in portions of the magnetic structures
and particularly, portions of the soft iron struc-
35 tures, are slightly and temporarily altered during
operation. Consequently, if a rotary stepping motor,
with holding current applied to one of its coils,



5 is mechanically deflected a few degrees clockwise from that null point, and the mechanical torque is then slowly released, the motor will return to a null position. During this deflection, magnetomotive forces from the coil in combination with magnetomotive forces in the magnetically hard and soft structures, impresses a pattern of magnetic flux on the structure which is constantly changing until the system comes to a stop indicating that the internal forces are balanced. If the motor is then mechanically deflected a few degrees counter-clockwise from that pole position and the mechanical torque again slowly released it will return to a new null position far removed from the original null position. A small portion of this change in null position or "pole shift" was due to the width of the dead band, but the majority of this shift was due to the creation of a different pattern of magnetism, which resulted in a new point of balanced forces. The positional indeterminacy in low cost stepper motors due to pole shift, is typically on the order of 3% of a full step which can radically affect data storage.

30 Another problem with the use of low cost stepper motors as precision positioners, is the loose tolerances applied to their fabrication and assembly. Positional errors of 5% of a step or more from this cause are not uncommon.

35 In order to achieve a small dead band, high torque is required, which means the highest possible ampere turns in the electromagnetic coils.



For a given volume available for the copper magnet
5 wire, a trade off is made between ampere turns and
the greatest heat build up that can be tolerated.
Consequently, stepper motors are major heat sources,
and unlike rotating motors that assist in cooling
themselves by generating windage, the stationary
10 stepper conducting a holding current through one
of its windings becomes quite hot. To alleviate
this heat problem, some drive circuits incorporate
a time delay to reduce the normal drive current to
a lower holding current after the motor has stopped
15 for a specific time interval. This approach re-
duces the heat build up, but it increases the po-
tential dead band width, and assumes that the ar-
mature has already come to rest within the original
high current dead band, and that subsequent shock,
20 vibrations, etc., will not deflect the armature
into the new, wider, low current dead band. Al-
though permanent magnet type steppers possess a
low level of torque when no current is flowing
through the coils, this has not been sufficient
25 in prior art to permit turning off the holding
current entirely.

A number of means have been devised in
prior art for converting the rotary motion of the
30 rotary stepper motor into linear motion. One of
these is a spiral face cam mounted on the stepper
motor shaft. A spherical follower attached to
the carriage, is spring biased into the spiral
"V" groove of the face cam. As the stepper motor
35 rotates the spiral cam, the follower moves radi-
ally and the carriage thus moves lineally with it.
When the follower is close to the axis of the spi-
ral cam, the friction between the follower and



5 the cam, as well as the reaction to the inertia
and friction of the carriage, are all acting at
a small radius, (moment arm) and therefore, the
torque loads on the stepper motor are small.
Consequently, damping is minimal and settling
times are long. Conversely, when the follower
10 reaches the outer portions of the spiral cam,
the torque loads on the stepper are large, and
the dead band is excessive.

Another problem with the spiral cam, is
15 its close proximity to the stepper motor. Heat
from the stepper motor is conducted through the
motor shaft into the spiral cam. When the drive
is first turned on, the stepper and cam are cold.
Data written on the cold drive is located in the
20 position dictated by the cold spiral cam. After
the drive has been operating for a time, the spi-
ral cam has been expanded by heat from the motor.
The transducer directed to the same nominal data
position will instead be displaced by the amount
25 of thermal expansion.

Another problem with the spiral cam, is
that it is positionally referenced from the step-
per motor itself. Since the motor is a major
30 heat source, the structure to which the motor is
mounted becomes heated and dimensionally distorted
by that heat. This distortion changes the basic
reference point from which the carriage position
is derived. As a result, spiral cams are not used
35 in the more precise drives.

Another prior art design uses a miniature ball bearing cam follower which runs against a spiral face cam. A small extension spring biases the cam follower against one side of the face cam.

This design adds the inaccuracies of the cam follower to the other inaccuracies of the spiral cam approach. Also the extension spring acting against the slope of the cam applies a biasing torque against the motor. This torque works against the motor in one direction as the motor extends the spring and thus stores energy in the spring. In the other direction, the motor is augmented by the energy released by the relaxing spring. In an open loop stepper system such as this, such augmentation is particularly unwelcome, since the stepper must be operated at a much slower rate to avoid accidentally over accelerating and losing proper synchronism. This is particularly a problem in this design due to the small amount of damping friction provided by the ball bearing cam follower which also causes substantial settling times.

Another prior art means of coupling a rotary stepper to a linear carriage has been a lead screw. In this mechanism, the stepper motor shaft is elongated several inches, and a helical groove or thread is cut into the surface of the shaft. A follower attached to the carriage is spring biased into this groove.



5 To achieve sufficient accuracy in this
lead screw, the helical groove must be precision
ground. To keep the polar inertia of the system
low, the shaft diameter must be small. As a conse-
10 quence, the depth of the ground groove is a sub-
stantial percentage of the shaft diameter, and the
unbalanced stresses created by the asymmetrical
grinding on the periphery of the shaft distorts
it, and maintaining lead screw straightness be-
comes a problem. Some lead screws are ground
15 with a double pitch thread to create symmetrical,
balanced stresses to improve straightness.

The thread grinding process by which
these lead screws are manufactured is a sequential
process. The entire thread cannot be produced at
20 the same time. Instead, the grinding wheel is first
plunged into the material at one end of the thread,
and then follows the helical path of the thread
until reaching the opposite end of the thread, it
is retracted. Heat is generated during the grind-
25 ing process. Although an attempt is made to stan-
dardize the temperature of the blank shafts, the
grinding wheel, and the cutting oil, nevertheless
the shaft gains temperature during the grinding
operation. Consequently, since the pitch of the
30 ground thread is maintained as a constant during
the grinding operation, after the part is complet-
ed and its temperature becomes equal along its
length, the pitch at the beginning end of the ground
thread is greater than the pitch at the completion
35 end of the thread. This is because the pitch at
the completion end was ground into a warmer expand-



5 ed shaft which has since contracted and with it
the pitch of the thread ground into its surface
has contracted. Finishing the thread with a very
light second grind could largely eliminate this
problem, but is not economically practical in
low cost applications. Even though the crudest
10 form of thread grinding is used, these ground
lead screws contribute substantially
to the cost of the drives.

15 Since the lead screw is an extension
of the motor shaft, the substantial heat of the
stepper motor is conducted down the shaft, and
radiated from the shaft. The result is a heat
gradient down the length of the shaft. When the
drive is first turned on, the pitch of the lead
20 screw is as manufactured. After the drive heats
up, the pitch is expanded greatly near the motor,
and to a lesser degree at the end away from the
motor. This loss of positional accuracy due to
motor heat is independent of a similar loss of
25 accuracy due to changes in ambient temperature.
As the ambient temperature changes, the dimen-
sions of the data disc, which is fabricated from
biaxially oriented polyethylene terephthalate,
change also. Ideally the changes in pitch of
30 the lead screw from ambient temperature changes
should match the changes in pitch of the data
tracks from the same cause. This has not been
accomplished in the prior art, and for practical
manufacturing reasons, lead screws have been fab-
35 ricated from stainless steel with thermal expan-
sion rates different from that of the data discs.



5 In some designs the stepper motor be-
comes the point of reference from which the car-
riage is positioned by the lead screw. Since
the stepper is a major heat source, this heat
from the motor conducts into the structure to
which the motor is attached, and through thermal
10 expansion distorts that structure, thereby chang-
ing the reference point from which positioning
begins and shifting the carriage position.

To eliminate this problem, and to mini-
15 mize the effects of stepper motor heat on the
lead screw accuracy, some designs float the motor
on a flexible support, and reference the opposite
end end of the lead screw. In these designs, the
portion of the lead screw furthest away from the
20 motor and therefore the coolest, is used for posi-
tioning the carriage, and since it is fairly cool,
it creates only a small amount of distortion from
thermal expansion in the structure to which it is
attached. The drawbacks to this system are added
25 complexity, and in some cases troublesome, once-
around, elastic torque loads are generated. In
those systems, three bearings are mounted on a
slightly curved lead screw, and the motor which
contains two of those bearings is supported on an
30 elastically flexible mount. As the curved lead
screw rotates, the motor is forced to wobble with
it, and in the process alternately stores and re-
leases energy in the elastically flexible mount.
This alternating spring biasing of the torque out-
35 put has the same negative effects as the exten-
sion spring in the face cam drive.



5 In this system, since the motor is flexibly
mounted, rather than directly mounted to a solid
supporting structure such as a cast chassis, the heat
sink effect of such a chassis is lost, and since the
flexible mount usually has less thermal conductivity,
10 the stepper motor warms to a higher temperature, as
does the lead screw. Thus, the thermal expansion of
the lead screw pitch is even greater in this system.

When the biaxially oriented polyethylene
terephthalate disc changes size in response to a
15 change in ambient humidity, it is preferred, that
the pitch of the lead screw would change by the same
amount. Since the lead screws in prior art have been
manufactured from stainless steel which is dimen-
sionally unaffected by changes in humidity, no such
20 hygroscopic compensation in the pitch of lead screws
has been accomplished.

A further difficulty in lead screw designs
is the proper tracking of the lead screw by the fol-
25 lower. In order for the follower to exactly dupli-
cate the pitch of the lead screw, it must move ex-
actly parallel with the axis of the lead screw. If
the lead screw were exactly straight, this would
present a substantial alignment problem, and since
30 due to manufacturing limitations, the lead screws
are frequently slightly curved, the problem is
worse.



5 In the prior art, an attempt is usually
made to minimize this alignment problem by mounting
the carriage on the lead screw itself by means of a
pair of bearings in addition to the follower which
rides in the ground thread with these extra bearings
riding frictionally against the outside diameter of
10 the lead screw. As a result, the carriage moves
laterally and vertically in response to rotation of
the somewhat curved lead screw as a whole, while de-
partures from proper pitch by the follower are limited
to those caused by curvature of the lead screw only
15 in the length between the two supporting bearings.

The addition of these extra carriage support
bearings adds substantially to the frictional torque
load which must be driven by the stepper motor, and
20 that in turn increases the dead band.

Some of these lead screw designs utilize a
90 degree "V" thread profile, in which case when the
carriage reaches the end of its travel, the follower
is driven out of the thread without harm to the system.
25 Other designs use 60 degree Acme thread profiles, and
when the carriage in such a system is driven to the
end of its travel, destructive forces are generated by
the lead screw. Such systems are generally augmented
by mechanical stops which impact a rotating element on
30 the lead screw against some portion of the carriage
to provide a less destructive stopping means. These
stopping means still create high rates of deceller-
ation, and large forces in the structure, and are a
cause of failures and misalignments.
35

The third means of coupling the rotary
5 stepper to the linear carriage in the prior art has
been a flexible band. In this design, a cylindrical
drum or capstan is mounted on the shaft of the stepper
motor. A thin, flexible metal band encircles the
10 capstan and is attached to the capstan at the midpoint
of the band. By this means, when the stepper motor
rotates, the flexible band moves with it carrying the
carriage along as well.

The magnitude of movement of the carriage
15 is a function of the pitch radius of the flexible
band, which is the outside radius of the capstan plus
one half of the band thickness. A problem with this
design is that the capstan mounted directly onto the
stepper motor shaft is heated by the motor and ther-
20 mally expands, thus increasing the pitch.

Another problem is that the portion of
the band that is wound around the capstan assumes
nearly the same temperature as the capstan, while
25 that portion of the band lying away from the capstan
does not. As a result, when a hot capstan turns, it
unwinds hot and therefore thermally expanded metal
band, and winds in cool and therefore contracted metal
band. After the capstan stops at the newly selected
30 position, the hot newly unwound band begins to cool
and contract. At the same time the cool, newly wound
band draws heat from the capstan and begins to warm



and expand. The result is that initial positioning
5 by the stepper is exaggerated, followed by a slow
correction as thermal equilibrium is again estab-
lished.

Another weakness in this system is that
10 the stepper motor is the basic reference point from
which positioning is established, and the heat from
the stepper motor is conducted into the supporting
structure, thermally expanding and distorting it,
thus changing the reference point and therefore the
15 carriage position.

The capstan in such a system is a preci-
sion part, the diameter and concentricity of which
must be very accurately established to closely coin-
20 cide with the nominal pitch value. Consequently,
existing designs have fabricated this part by preci-
sion grinding a metallic part. Ideally this part
should expand thermally at the same rate as the bi-
axially oriented polyethylene terephthalate data disc
25 to achieve exact compensation for changes in ambient
temperature. The materials used for capstans in the
prior art do not achieve this goal.

It is also desirable, that the capstan ex-
30 pand hygroscopically at the same rate as the disc,
but this too has not been achieved in the prior art
mechanisms. The metallic capstans are totally un-
affected dimensionally by changes in humidity, and
so the changes in data ring pitch are completely
35 uncompensated.



Another problem with this design, is the short life of the flexible band. The maximum practical number of steps that can be provided in a stepper motor is approximately 200 per revolution. In order to couple such a 200 step motor to a 96 track per inch carriage, the capstan must have a pitch radius of .3316 inches. In order to flex the band around such a tight radius, even for a small number of cycles, the band must be very thin, stresses are high, and service life is limited. Because of these limitations, further pitch reductions are not practical with this system. A further limitation is the requirement, that the capstan must turn somewhat less than a full turn, otherwise the flexible band would start to wind on top of itself.

If the carriage in such a system is run to the end of its travel, and the rotating motor is stopped suddenly by the flexible band, it is overstressed and breaks the band. Consequently, stops must be added to the motor to limit its rotation. When the motor is run at full speed into one of these stops, rapid deceleration occurs. This sudden deceleration is conveyed to the carriage by means of the flexible band, and as the momentum of the carriage is suddenly applied to the band, high stresses are generated. The elements of this system are precise, fragile, and easily damaged during assembly.



Objects And Advantages Of The Invention

5

It is an object of the invention to produce a low cost positioner that can be assembled from inexpensive injection molded plastic parts.

10

It is a further object, to provide for easy assembly without skilled labor, or expensive tooling.

15

It is a further object, to provide precise positioning and eliminate the positional indeterminacy of dead bands.

It is a further object, to provide movement to a selected position without overshoot.

20

It is a further object, to eliminate oscillations around the selected position, and by so doing enable the drive to transfer data immediately upon arrival of the transducer at the selected data track.

25

It is a further object, to provide positional accuracy which is not degraded by pole shift phenomena.

30

It is a further object, to reduce power consumption and heat generated in the drive, by providing a means for precisely maintaining a selected position without a holding current.



5 It is a further object, to provide substantial thermal isolation between the heat generating components of the positioning system, and the precision elements that determine the geometric positions.

10 It is a further object, to provide an incremental positioner whose pitch changes with changes in ambient temperature, and which approximates the thermal expansion of biaxially oriented polyethylene terephthalate.

15 It is a further object, to provide an incremental positioner whose pitch changes with changes in ambient humidity, and which approximates the hygroscopic expansion rate of biaxially oriented polyethylene terephthalate.

20 It is a further object, to provide a positioner which can be quickly and easily aligned to a standard position without special skills.

25 It is a further object, to eliminate tightly toleranced motor parts and narrow air gaps.

30 It is a further object, to eliminate movement of motor coils as a means of adjustment, and thereby permit installing those coils as fixed, permanent elements on a printed circuit board.



5 It is a further object, to eliminate mechanical stops on rotating elements, and permit unlimited rotation with automatic decoupling of the carriage should the carriage reach the end of its travel, such uncoupling to be accomplished without damage to components and subject to automatic re-
10 coupling when excessive rotation has ceased.

 It is a further object, to provide an incremental positioner capable of moving in very small increments.

15

 It is a further object, to provide a positioner with a long service life.

20

Summary Of The Invention

 A transducer positioning device which includes a movable member affixed to the magnetic transducer and having a plurality of force receiving surfaces of a first predetermined spacing. Positioned adjacent the movable member is a fixed member having a second plurality of force receiving surfaces of a second predetermined spacing, the second predetermined spacing being different from the first predetermined spacing. Driving means is disposed adjacent the fixed and movable members and includes apparatus for moving the driving means into simultaneous engage-
25
30



ment with the first and second surfaces for aligning
at least one of the movable surfaces with one of the
fixed surfaces for aligning the same.

Brief Description Of The Drawings

Figure 1 is an exploded schematic illustration
of a transducer positioning apparatus constructed in
accordance with the present invention;

Figure 2 is a cross-sectional view illustrating
the positioning apparatus in assembled form;

Figure 3 is a cross-sectional view taken
about the lines 3-3 of Figure 2;

Figure 4 is a fragmented exploded schematic
illustration of a portion of the apparatus illustrated
in Figure 1;

Figure 5 is a plan view illustrating the drive
mechanism used in the transducer positioning apparatus
of the present invention;

Figure 6 is an elevational view of the structure
shown in Figure 5;

Figure 7 is a cross-sectional view illustrating
one of the cam details;



Figure 8 is a fragmented view partly in cross-section illustrating the relationship between the moving rack and the fixed rack constructed in accordance with the present invention;

Figure 9 is a cross-sectional view taken about the lines 9-9 of Figure 8;

Figure 10 is a view taken about the lines 10-10 of Figure 9.

Figure 11 is a view taken about the line 11-11 of Figure 2 illustrating the drive motor in further detail.

Detailed Description

The transducer positioning apparatus of the present invention is particularly designed for the precise positioning of the magnetic transducer heads of a floppy disc for reading/recording thereon as is well known in the art. However, the structure may have applications other than the positioning of transducer heads such, for example, as wherever it is desired to accurately position a structure utilizing a minimum amount of force in doing so. Generally, the apparatus to be positioned is effectively floated between a plurality of balls which are held on opposite sides thereof within continuous track means. Pressure is applied to keep at least some of the balls in contact with the opposite surfaces of the device to be positioned while force is applied through appropriate motive means to the device to cause it to move to the

desired predetermined position. In the preferred embodiment, a plurality of balls are discretely forced, one at a time into opposed "V" shaped notches formed by a pair of spaced apart racks, one fixed and the other movable. The balls ride upon a plurality of cams mounted on a shaft turned by a motor responsive to electrical signals applied thereto.

As is illustrated in FIGURE 1 there is provided a carriage assembly 10 which is sandwiched between the bottom surface 12 of a chassis 14 and the inner surface (not shown) of a support assembly 16. A first group of balls 18 is positioned upon one surface 20 of the carriage assembly 10 and follow a channel or groove 22 formed therein. A second plurality of balls 24 are positioned on the opposite surface 26 of the carriage 10 and are retained within a groove or channel (not shown) formed therein. A biasing means such as the spring 28 is interconnected between the support member 16 and the base 12 of the chassis 14, thus completing a continuous track within which the balls 18 and 24 travel. A means for applying force to move the carriage 10 may take the form of a motor 30 which is held within the support assembly 16 and is interconnected by means (not shown) to the carriage 10. Appropriate adjusting mechanisms 32 and 34 are positioned intermediate the support member 16 and the chassis 14 to properly adjust the positioning of the various elements contained therein.



As is shown in FIGURE 2, the carriage 10 is sandwiched between the chassis 14 and the support member 16. the balls 18 are held in position between the carriage 10 and the support member 16 while the balls 24 are held in position between the opposite surface of the carriage 10 and the chassis 14. The motor is shown generally at 30 and is held in place by the support member 16 with appropriate power and control signals being applied to it through circuits positioned upon printed circuit board 36. The details of construction of the motor and the printed circuit board are set forth below.

As is illustrated in FIGURE 3, the carriage 10 forms on the surface 20 thereof, the continuous channel 22 within which the balls 18 are positioned. Preferably the carriage 10 is made from a glass-filled, polycarbonate plastic which has been appropriately molded. It should be understood, however, that the carriage 10 may be constructed from any material which is desired. As is illustrated, the channel 22 contains one section which functions as a return section for the balls and is formed as a generally U-shaped section 38 which is depressed further into the surface 20 of the carriage 10. The U-shaped section or portion of the channel 22 is formed in such a manner that the balls 18 are freely movable within the channel but, at the same time, are restrained so that they remain within the tract. It should be noted that the arms of the U-shaped channel are effectively closed by the end section 40 of the support member 16,

5 thereby trapping the balls internally within this portion of the track so that they cannot be dislodged therefrom. A second section of the track is formed effectively by a V-shaped channel 42 within which the balls 18 are positioned.

10 The balls are firmly retained in an opposed V-shaped groove 52 formed in the surface 54 of the support member 16 by force generated by the spring 28 interconnected between the support members 16 and the chassis 14. As will be noted, the support member 16
15 is fixed relative to the carriage 10 and the groove 52 formed therein thus also remains fixed. As a result, the groove 52 provides the defined path along which the carriage 10 moves responsive to the force being applied thereto. As the carriage moves the balls 18
20 contained within the track 22 are thus circulated.

On the opposite surface 40 of the carriage 10, there is provided a pair of curved channels 62 and 64 within which the balls 24 are permitted to continuously and freely circulate. As is noted, the channel
25 62 is generally U-shaped in cross-sectional configuration with one surface thereof 66 being depressed slightly less than the other surface thereof 68 into the surface 60 of the carriage 10. As a result, the balls
30 24 which are within the section 66 are in contact with the chassis 14 while the balls within the section 66 are restrained within the channel by the chassis 14 but are not contacted by it. As a result only the balls which contact the chassis 14 have a force applied



5 thereto. A similar interaction is provided for channel
64 as shown at 70 and 72 and functions in the same
manner.

10 As will be clearly recognized by those skilled
in the art, through the application of force to only a
portion of the balls 18 held within the continuous
tracks on opposite surfaces of the carriage 10, only a
minimal amount of friction is generated. Through such
controlled application of force and by balancing the
15 channels on opposite surfaces of the carriage so that
a force is applied effectively to the center of the
carriage on one side and at opposite ends on the other,
the carriage effectively "floats" between the support
member and the chassis upon a minimal number of the
balls and can be thus easily moved through the appli-
20 cation of a minimal amount of force. Through such
capability the motor 30 can be a relatively low power
motor which generates relatively little heat.

25 The carriage 10 is moved through forces
applied thereto in a manner that will be described
fully hereinbelow. The forces are generated through
the utilization of the motor 30 as shown in Figure 2.
The motor 30 includes a pair of magnets 80 and 82
which are mounted upon a shaft 84. The shaft 84 is
30 mounted for rotation within a recess 86 of the support
member 16. The recess 86 receives a ball 88 which
is held in place within a conical depression 90
formed in the end of the shaft 84.



5 The magnets 80 and 82 are ceramic magnets
which have been cemented to the cam shaft 84 to
hold them in place. A rotary orientation flat 92
is provided in the shaft 84. The flat 92 is uti-
lized to assist in clamping the shaft 84 into an
appropriate angular position in a magnetizing tool
10 where a desired number of flux zones are induced
into each of the two magnets 80 and 82. In accord-
ance with the preferred embodiment of the present
invention, twelve such flux zones are induced into
each of the magnets 80 and 82.

15 The printed circuit board assembly 36
(Figure 20) includes two bifilar wound coil bobbin
assemblies 94 and 96. Each of the bobbin assemblies
94 and 96 encircles two soft iron positioner stators
20 98 through 104. These stators are held in place by
a stator clip 106. When thus assembled, the driver
motor 30 comprises a twenty-four steps per revolution
permanent magnet stepping motor. The magnetic flux
zones in the magnets 80 and 82 are oriented such that
25 for each of the twenty-four step positions of the
motor the carriage 10 is moved a discrete amount in
a manner as more fully described hereinbelow.

30 As is shown in Figure 1, the carriage 10 is
formed on the surface 20 thereof a rack 110. The rack
110 is disposed adjacent the support member 16 as more
clearly seen in Figure 3. As illustrated in Figure 4,
the support member 16 also has a rack 112 formed thereon.
A plurality of balls shown generally at 114 are posi-



tioned so as to be retained loosely within the rack
5 112. The shaft 84 has formed thereon a plurality of
cams shown generally at 116 and is disposed to rotate
upon bearings in the form of balls 88 and 118 which are
appropriately supported at one end in the recess 86 and
at the other upon the armature pivot support 120 which
10 in turn is supported upon the projection 122 of the
support member 16. The armature pivot support 120 in
turn is secured by a spring 124 within openings 126
provided on the support member 16. As the cams 116 are
rotated by the motor 30, the balls 114 are each discrete-
15 ly moved into contact with the racks 112 and 110. The
racks 110 and 112 define the opposed notches with dif-
ferent pitches in each rack and are aligned in such
a manner that one ball at a time will position the rack
formed on the carriage 10 with respect to the support
20 member 16 thereby causing the entire carriage 10 to
move a discrete amount. As the shaft 84 rotates, a
different cam forces a different ball discretely into
joint engagement with the two racks thus again moving
the carriage relative to the support 16.

25

Additional details of the cam assembly and
its structure are shown in Figures 5 and 6. As is
therein illustrated, the balls 114 are eight in number
131 through 138 and each is disposed in contact with
30 one of the cams 141 through 148, also eight in number.
Preferably, the shaft 84 with the cams 116 formed
thereon is constructed from an integral unitary member
of molded polyamide plastic. Each of the cams 141
through 148 has a three lobe profile as shown in Figure
35 7. Each of the high spots on the cam has a ten degree



5 area of constant radius as shown at 150. Each of
the cams is also rotated 15 degrees relative to an
adjacent cam on the shaft 84. The shaft 84 is pro-
vided with a pair of conical sockets 90 and 152 at
each end thereof. Into these conical sockets 90 and
152 the balls 88 and 118 respectively fit. The ball
10 88 is held in position by the conical socket 154 in
the support member 16 (Figure 2). The ball 118 is
held in position by the conical socket 156 provided
in one end 158 of the armature pivot support 120.
An additional socket which is generally conical in
15 configuration and shown at 160 is provided inter-
mediate the ends of the armature pivot support 120
and receives the spherical projection 122 on the
support member 16. The hook 162 of spring 124 en-
gages an opening 164 provided in the opposite end
20 166 of the armature pivot support 120. The oppo-
site hook 166 of the spring 124 engages the opening
126 molded into the support member 16. The exten-
sion spring 124 is positioned so as to simultane-
ously bias the armature pivot support 120 against
25 the spherical projection 122 and against the ball
118. This biasing force against the ball 118 is
resisted by two reaction forces, the first acting
axially through the shaft 84 to the ball 88 and the
second acting at right angles to the shaft through
30 one of the eight balls 131 through 138.

It can now be seen that the shaft 84 biased
by the spring 124 acting through the armature pivot
support 120 and rotated by changes in magnetic flux
35 in stators 98 through 94 becomes a means for sequen-
tially applying a positioning force against the eight
balls 131 through 138. By thus sequentially switching



currents in the two bifilar wound coils 94 and 96,
positioning forces are sequentially applied to the
balls 131 through 138 and the carriage 10 is thus
moved from any data track position on the floppy disc
to any other data track position in accordance with
the signals which are applied.

Reference is now made to Figures 8 through
10 wherein the details of the rack 110 on the car-
riage 10 (the movable rack) and the rack 112 on the
support member 16 (the fixed rack) are shown along
with an illustration of the manner in which forces
are applied to move the movable rack relative to the
fixed rack. In the preferred embodiment as illus-
trated in Figures 8 through 10, a series of twenty
"V"-shaped notches 161 through 180 are molded into
the carriage 10. Each of these "V"-shaped notches
include first and second walls 190 and 192 as shown
in conjunction with the "V"-shaped notch 180. The
walls 190 and 192 slope downwardly toward the sur-
face 194 and are bridged by an additional flat mold-
ing surface 196. The eight notches 181 through 188
molded into the fixed rack 112 are constructed in a
similar fashion as is shown more clearly in Figure
10. Preferably, the notches 161 through 180 have a
110 degree dihedral angle and a notch to notch pitch
of .0833 inches. The eight "V"-shaped notches 181
through 188 on the fixed rack are molded into the
support member 16 and also have a dihedral angle of
110 degrees but a notch to notch pitch of .09375.
Also molded as part of the support member 16 in con-
junction with each of the eight notches 181 through



188 is a flange or plate 200 defining retaining
5 pockets 201 through 208. Each of the pockets 201
through 208 is .078 inches in diameter such that
the pockets will receive a 1/16 inch diameter stain-
less steel ball therein. Obviously, the dimensions
10 above set forth can vary depending upon the parti-
cular application and the diameter of the balls
which are to be used to drive the movable carriage.

As illustrated more clearly in Figures 9
and 10, the ball 131 is driven upward by a force
15 applying means 210 (for example one of the cams
141 through 148 on the shaft 84) until the ball
wedges solidly against four points of contact.
Two of those points of contact are on the movable
rack 110 and two are on the fixed rack 112. For
20 example, the two flat surfaces of the "V"-shaped
notch 181 are opposed the two flat surfaces in
"V"-shaped notch 161 (see notch 180 of Figure 8),
and thus provides the four surfaces against which
the ball will wedge to provide the four points of
25 contact above-referred to. The action of the force
supplying means 210 causes the movable rack 110 to
move so that the "V"-shaped notches 161 and 181
align with each other. With this alignment accom-
plished, the magnetic transducer attached to the
30 carriage 10 is properly aligned with a nominal
position of the outermost data track on the rotat-
ing disc commonly called in the art, track zero.
During this alignment, balls 132 through 138 are
allowed to move freely within the pockets 202
35 through 208 formed in the flange 200.



To position the carriage 10 from track zero to track 1, the force supplying means 210 is withdrawn and at the same time a similar means (not illustrated) applies a similar force against ball 138 in socket 208 thus causing contact between ball 138 and flat surfaces on the "V"-shaped notches 169 and 188. As this is done, the ball 138 first contacts the "V"-shaped notch 169 at point 214 and drives the movable rack 110 and the carriage 10 in the direction shown by the arrow 212. As the ball 138 rotates and moves more deeply into the opposed "V"-shaped notches 188 and 169, contact is finally made at the four points thereon when the ball is fully wedged in place. At this time, carriage 10 motion is stopped almost instantly and the two opposed "V"-shaped notches are aligned. The carriage has been moved .0104 inches, when utilizing the various dimensions set forth in accordance with the presently preferred embodiment of the present invention, and the magnetic transducer means attached to the carriage 10 is aligned with data track number 1. It is in this manner that all of the eighty-nine data tracks normally available upon a typical floppy disc are accessed in either direction. That is, by sequentially applying a driving force against the balls 131 through 138 and such may be done in the sequence indicated in the following table:



	FORCING	NOTCH	DATA	MOTOR
	BALL	ALIGNMENT	TRACK	COIL
5	131	131 161	0	1
	138	138 169	1	2
	137	137 168	2	3
10	136	136 167	3	4
	135	135 166	4	1
	134	134 165	5	2
	133	133 164	6	3
	132	132 163	7	4
15	131	131 162	8	1
	138	138 170	9	2
	137	137 169	10	3
	136	136 168	11	4
	135	135 167	12	1
20	134	134 166	13	2
	133	133 165	14	3
	132	132 164	15	4
	131	131 163	16	1
	138	138 171	17	2
25	137	137 170	18	3
	136	136 169	19	4
	135	135 168	20	1
	134	134 167	21	2
	133	133 166	22	3
30	132	132 165	23	4
	131	131 164	24	1
	138	138 172	25	2
	137	137 171	26	3
	136	136 170	27	4
35	135	135 169	28	1
	134	134 168	29	2
	133	133 167	30	3
	132	132 166	31	4

	FORCING	NOTCH	DATA	MOTOR
5	BALL	ALIGNMENT	TRACK	COIL
	131	131 165	32	1
	138	138 173	33	2
	137	137 172	34	3
10	136	136 171	35	4
	135	135 170	36	1
	134	134 169	37	2
	133	133 168	38	3
	132	132 167	39	4
15	131	131 166	40	1
	138	138 174	41	2
	137	137 173	42	3
	136	136 172	43	4
	135	135 171	44	1
20	134	134 170	45	2
	133	133 169	46	3
	132	132 168	47	4
	131	131 167	48	1
	138	138 175	49	2
25	137	137 174	50	3
	136	136 173	51	4
	135	135 172	52	1
	134	134 171	53	2
	133	133 170	54	3
30	132	132 169	55	4
	131	131 168	56	1
	138	138 176	57	2
	137	137 175	58	3
	136	136 174	59	4
35	135	135 173	60	1
	134	134 172	61	2
	133	133 171	62	3
	132	132 170	63	4



	FORCING	NOTCH	DATA	MOTOR
	BALL	ALIGNMENT	TRACK	COIL
5	131	131 169	64	1
	138	138 177	65	2
	137	137 176	66	3
10	136	136 175	67	4
	135	135 174	68	1
	134	134 173	69	2
	133	133 172	70	3
	132	132 171	71	4
15	131	131 170	72	1
	138	138 178	73	2
	137	137 177	74	3
	136	136 176	75	4
	135	135 175	76	1
20	134	134 174	77	2
	133	133 173	78	3
	132	132 172	79	4
	131	131 171	80	1
	138	138 179	81	2
25	137	137 178	82	3
	136	136 177	83	4
	135	135 176	84	1
	134	134 175	85	2
	133	133 174	86	3
30	132	132 173	87	4
	131	131 172	88	1



5 It will be recognized that the force apply-
ing means 210 is one of the cams 114 on the shaft 84.
That is, the camshaft 84 which is biased by the ex-
tension spring 124 through the armature pivot support
120 and rotated by changes in magnetic flux in stators
98 through 104 becomes a means for sequentially ap-
10 plying the positioning force against the eight balls
131 through 138. By sequentially switching currents
in the two bifilar wound coils 94 and 96, positioning
forces are sequentially applied to balls 131 through
138 and the carriage 10 is moved from any data track
15 position to any other data track position in accord-
ance with the energization of the coils.

After the carriage 10 has been positioned
to a desired data track in response to a sequence
20 of current pulses applied to the bifilar coils 94
and 96, a holding current is temporarily sustained
in the last coil energized during the sequence.
If this coil is one of the bifilar windings in coil
assembly 94 for example, then the electromagnetic
25 flux will interact with permanent magnet 82.

Camshaft 84 initially rotates beyond the
selected null position, and then oscillates about
that null position until kinetic energy is damped
30 from the system. Since each cam lobe has a 10
degree dwell 150, the effects of this overshoot and
oscillation are not imparted to the carriage. Any
overshoot or oscillation 10 degrees or less in am-



plitude, has no effect on the biasing force being
5 applied to the selected ball. Consequently,
the carriage reaches the selected position with
negligible overshoot, the ball becomes fully seated
in the matching "V"-shaped notches with negligible
oscillation, and the carriage is at rest at the
10 selected position long before the kinetic energy
in the camshaft assembly has been dissipated.

Since the camshaft 84 can stop anywhere
on the 10 degree dwell of a given lobe without
15 effecting carriage position, the angular magnitude
of the dead band of the stepping motor need not be
minimized. Consequently, driving currents can be
low, minimizing power requirements and heat genera-
tion, air gaps can be wide minimizing expensive
20 precision fits and difficult alignments, and fric-
tion levels can be raised to improve reliability
during high speed operation.

Similarly, pole shift does not effect
25 carriage position, therefore, there is no need to
minimize pole shift, and the materials and processes
used in fabricating the magnetic structure of the
stepping motor can be selected for maximum economy.

30 Any shock applied to the carriage 10 might
momentarily move the carriage a small distance from
the selected position, and in the process force the
selected positioning ball and the camshaft 84 to also
move a small distance. This shock induced motion



5 by the camshaft would extend the biasing spring 124
which would then return the system to its normal con-
dition as soon as the shock had passed. The accurate
retention of a selected position is a function of
the force of the extension spring 124 and does not
require a steep torque curve from the stepper motor
10 30. Since linear shock forces applied to the car-
riage 10 cannot be transmitted to the camshaft 84 as
rotary forces, and since the camshaft assembly is
inherently balanced, a very low torque is sufficient
to prevent unintentional rotation. The permanent
15 magnets 80 and 82 generate a cogging torque when no
current is flowing through coils 94 and 96. This
cogging torque is sufficient to prevent unintention-
al rotation, therefore the temporary holding current
in the last switched coil is sustained only enough
20 for the oscillations to damp out. It is then switch-
ed off, and the permanent magnet provides the holding
torque until carriage repositioning is initiated.
By turning off this holding current, energy is saved
and heat build up in the drive is minimized.

25 Although the coil assemblies 94 and 96
generate heat only during positioning movements and
a few milliseconds thereafter, this heat is still
a significant consideration in positioning accuracy.
30 Heat from the coils is conducted through the stators
98 through 104 and radiates across the air gaps into
the permanent magnets 80 and 82. From the magnets,
the heat is conducted into the camshaft. This heat
expands the length of the camshaft, which has no



effect on the interaction of the cams and the positioning balls. This heat is also conducted down the length of the camshaft 84 to the cam area 116, but due to the low thermal conductivity of the polyamide and the large surface area of the shaft, the temperature rise at the cam end of the camshaft 84 is quite low, and therefore the distortion of the positioning racks 110 and 112 from heat radiated from the camshaft and conducted from the camshaft by the balls 131 through 138 is negligible.

Since racks 110 and 112 are substantially isolated from heat sources within the drive, they will assume the temperature and humidity of the ambient atmosphere. Since the glass filled polycarbonate from which the racks are manufactured approximates the thermal and hygroscopic expansion of the biaxially oriented polyethylene terephthalate data disc, the pitch of the positioning system expands and contracts with changes in ambient temperature and humidity to match the expansion and contraction of the pitch of the data tracks written on the disc.

If the camshaft 84 is rotated in either direction in excess of the normal travel range, projections 220 or 222 on the carriage 10 will strike projection 224 on armature support 103. Further movement of the carriage will be obstructed, and as a result, the next positioning ball will be unable to fully seat into its pair of "V"-shaped notches, and as a result, the camshaft will be deflected



5 slightly from its normal position by a small extension of spring 124.

Referring to Figure 8, this action can be illustrated by assuming that carriage 10 has been driven from data track 3 to 2 to 1 to 0 and then beyond 0 in the same direction. The corresponding notch alignments would be then 186-167, 187-168, 188-169 and 181-161. With notches 181 and 161 in alignment as shown in Figure 8, the carriage is positioned at data track 0. If the camshaft continues to rotate in the same direction, the next drive ball to be activated is 132 and the grooves to be aligned are 182 and 162. However, in this case, we are assuming that the carriage has reached the end of its travel, and as a result, notches 182 and 162 are unable to fully align and therefore drive ball 132 is unable to fully seat in the 182-162 notches and, therefore, the camshaft is radially deflected slightly by the 132 ball acting on the 142 cam. As the camshaft continues to rotate, the 143 cam is brought into contact with the 133 ball and since the 183 and 163 "V"-shaped notches are even further out of alignment, the 133 ball is prevented from seating to an even greater amount than the 132 ball had been. Consequently the camshaft is deflected radially a still greater amount by the 133 ball acting against the 143 cam. Still further rotation brings the 144 cam into contact with the 134 ball, and since the 184 and 164 "V"-shaped notches are in still greater misalignment, the 134 ball is still further away



5 from complete seating, and the 134 ball acting against the 144 cam, deflects the camshaft radially still further. Still further rotation brings the 145 cam into contact with the 135 ball, and since the 185 "V"-shaped notch is evenly positioned between the 165 and 166 "V"-shaped notches, the 135
10 ball is unable to enter either notch, but bears instead on the shoulder between the two "V" notches. As a result, the camshaft is deflected radially still further by the 135 ball acting on the 145 cam. Still further rotation brings the 146 cam into contact with the 136 ball, and since the 186 notch is
15 somewhat aligned with the 167 notch, the 136 ball can penetrate to some extent into the 136-167 notch combination. As a consequence, the spring 124 can advance the camshaft 84 radially back somewhat
20 toward its normal position. Still further rotation brings the 147 cam into contact with the 137 ball and since the 187 notch is in even better alignment with the 168 notch, the 137 ball is able to penetrate to a still greater amount the 187-168 notch
25 combination. As a consequence, the camshaft advances radially even further back toward its normal position. Still further rotation brings cam 148 into contact with ball 138, and since the 188 notch is nearly aligned with the 169 notch, the 138 ball
30 is able to penetrate into the 188-169 notch combination to an extent where the surface of the 138 ball is acting against the sloping walls of the "V"-shaped notches rather than against the various mold lines of the "V"-shaped notches as is the case dur-



ing large degrees of mismatch. Because the ball 138
5 is acting against the side walls of the 188-169
notch combination, the resulting linear force against
the carriage is greater than the resisting friction
forces, and therefore the carriage is driven in a
reverse direction until ball 138 is fully seated in
10 notch combination 188-169, at which point the car-
riage 10 is positioned again at data track 1. Still
further rotation of the camshaft 84 in this direc-
tion will then position the carriage 10 in the normal
manner at data track 0 with notch combination 181-161
15 aligned, and still further rotation will again repeat
the above cycle. Unlimited rotation of the camshaft
is thus possible without damage to the mechanism.
Automatic uncoupling of the rotating and sliding ele-
ments is provided. The camshaft can be reversed from
20 its direction of excess rotation at any point, and
it will quickly and automatically reestablish the
normal drive coupling in the opposite direction with-
in a maximum of seven steps.

25 For illustrative purposes, this description
has assumed that the limit of carriage travel is
located just beyond the data track zero position.
In actual practice, additional travel is normally pro-
vided beyond the range of useful data tracks by pro-
30 viding additional "V"-shaped notches in the carriage
moving rack 110 such as "V"-shaped notch 180.

During carriage positioning, the camshaft
84 will rotate at more than 800 rpm. It is an advan-



5 tage of this system that there are no rotational,
mechanical stops against which the kinetic energy
stored in the rotating camshaft assembly could im-
pact destructively.

10 Shutter 232 as illustrated in Figure 2 is
a molded extension of carriage 10. Photoelectric
sensor assembly 234 is positioned so that shutter
232 interrupts the optical path of sensor 234 when
the carriage 10 is positioned at the unused data
track position of -2 (i.e. when "V"-shaped notch 183
15 is aligned with "V"-shaped notch 163). When the sys-
tem is first turned on, it is necessary to initialize
it in order to establish a known carriage position.
To accomplish this, when power is first supplied to
the unit, a series of current pulses are switched to
20 the bifilar coils in the sequence 4, 3, 2, 1, 4, 3,
2, 1, etc. As shown in Table 9, this sequence causes
the carriage to move in the direction from track 88
toward track 0 and beyond. When the carriage passes
the -2 track, the signal from the photosensor 234
25 halts the switching sequence. Due to dimensional
tolerances in locating the sensor, this signal may
be displaced as much as one and one-half data tracks
away from the nominal -2 track. Wherever the switch-
ing sequence is when the sensor signal is received,
30 the switching sequence stops at that point, and
after a pause of a few milliseconds for partial damp-
ing of camshaft oscillations, the coil switching se-
quence is reversed until coil No. 1 is again ener-
gized, at which point the initializing sequence is



ended, the "V"-shaped notches 181 and 161 are aligned, and the carriage is positioned at data track zero. After another pause of a few milliseconds to permit complete damping of camshaft oscillations, the temporary holding current is switched off, and the cogging torque of the stepping motor retains the system at data track 0 until commanded to move to another data track. The system is an open loop system, and no further positional information is required as long as current is supplied to the electronics which retains a continuous record of carriage position by means of an up/down counter. An advantage of the invention is the relatively loose tolerance on operation of the photosensor of plus or minus one and one-half tracks ($\pm 0.016''$) rather than the plus or minus one-half track ($\pm 0.005''$) of some prior art systems which must discriminate between adjacent tracks rather than every fourth track.



WHAT IS CLAIMED IS:

1 1. Magnetic transducer positioning appara-
2 tus comprising:

3
4 (a) a carriage carrying magnetic transducer
5 means.

6
7 (b) means for applying a force to said
8 carriage to move said carriage to a predetermined
9 position including:

10
11 (1) a first member having a first
12 plurality of force receiving surfaces of a first pre-
13 determined spacing;

14
15 (2) a second member disposed adja-
16 cent said first member and having a second plurality
17 of force receiving surfaces of a second predetermin-
18 ed spacing, said second predetermined spacing being
19 different from said first predetermined spacing;

20
21 (3) a driving means disposed adja-
22 cent said first and second members; and



23 (4) means for moving said driving
24 means into simultaneous engagement with said first
25 and second surfaces for aligning at least one of
26 said second surfaces with one of said first
27 surfaces.

1 2. Magnetic transducer positioning
2 apparatus as defined in Claim 1 wherein said driving
3 means includes a separate driving member for each of
4 said surfaces on said first member.

1 3. Magnetic transducer apparatus as
2 defined in Claim 2 wherein each of said driving mem-
3 bers defines a curved surface and said force receiv-
4 ing surfaces on said first and second members are
5 depressions defined by angular walls.

1 4. Magnetic transducer positioning
2 apparatus as defined in Claim 3 wherein said driving
3 members are balls.

1 5. Magnetic transducer positioning
2 apparatus as defined in Claim 4 wherein said walls
3 define flat surfaces.

1 6. Magnetic transducer positioning
2 apparatus as defined in Claim 5 wherein each of said
3 force receiving surfaces is a "V"-shaped notch.

1 7. Magnetic transducer positioning
2 apparatus as defined in Claim 2 wherein said means
3 for moving said driving members is a cam means.



1 8. Magnetic transducer positioning
2 apparatus as defined in Claim 7 wherein said cam means
3 has at least a single lobe for actuating one of said
4 driving members at a time.

1 9. Magnetic transducer positioning
2 apparatus as defined in Claim 7 wherein said cam means
3 has a plurality of cam surfaces only one of which at
4 any one time actuates a driving member to align one of
5 said surfaces on said first member with one of said
6 surfaces on said second member.

1 10. Magnetic transducer positioning
2 apparatus as defined in Claim 9 wherein said cam sur-
3 faces are carried by an elongated shaft.

1 11. Magnetic transducer positioning
2 apparatus as defined in Claim 10 wherein said cam sur-
3 faces are formed as an integral part of said elongated
4 shaft.

1 12. Magnetic transducer positioning
2 apparatus as defined in Claim 2 which further includes
3 means for retaining said driving members aligned with
4 said surface on said first member.

1 13. Magnetic transducer positioning
2 apparatus as defined in Claim 12 wherein said driving
3 members are balls and said surfaces are recesses and
4 said retaining means is a plate defining openings
5 therein receiving said balls, said openings being
6 aligned with the recesses in said first member.



1 14. Magnetic transducer positioning
2 apparatus as defined in Claim 1 wherein said second
3 predetermined spacing is greater than said first
4 predetermined spacing.

1 15. Magnetic transducer positioning
2 apparatus as defined in Claim 1 wherein said second
3 predetermined spacing is less than said first pre-
4 determined spacing.

1 16. Magnetic transducer positioning
2 apparatus as defined in Claim 7 wherein said cam
3 means is an elongated molded plastic shaft having
4 a plurality of individual cams formed integrally
5 thereon.

1 17. Magnetic transducer positioning
2 apparatus as defined in Claim 16 wherein said means
3 for moving said driving members further includes a
4 stepper motor for rotating said shaft.

1 18. Magnetic transducer positioning
2 apparatus as defined in Claim 17 wherein said shaft
3 is the shaft of said stepper motor.

1 19. Magnetic transducer positioning
2 apparatus as defined in Claim 18 wherein said shaft
3 is positioned for rotation on a single ball disposed
4 at each end thereof and received within a socket
5 formed in said shaft.



1 20. Magnetic transducer positioning
2 apparatus as defined in Claim 19 wherein said step-
3 per motor further includes a pair of permanent
4 magnets securely affixed to said shaft.

1 21. Magnetic transducer positioning
2 apparatus as defined in Claim 19 which further in-
3 cludes a spring loaded arm for receiving one end
4 of said shaft.

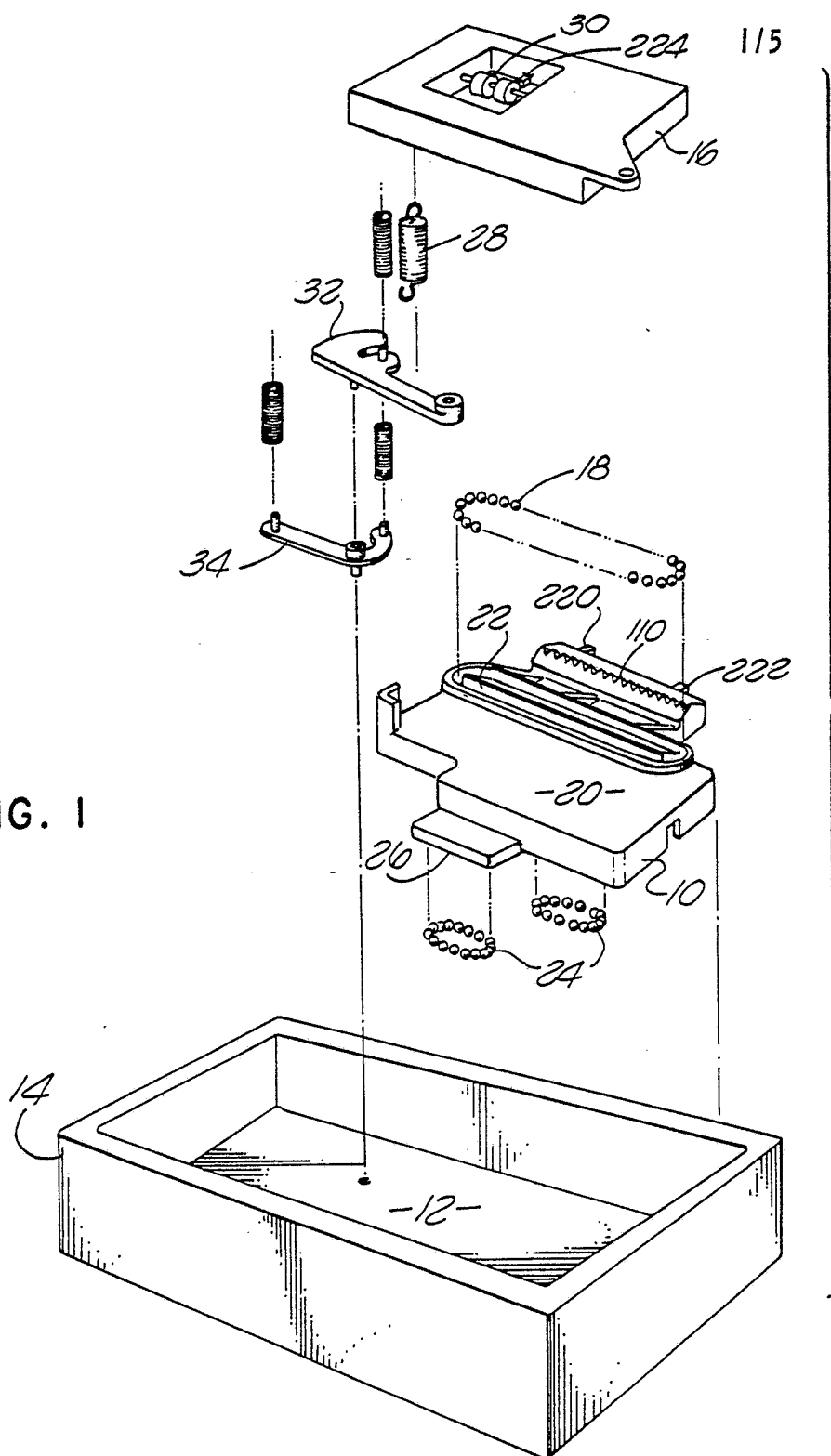
1 22. Magnetic transducer positioning
2 apparatus as defined in Claim 21 wherein said arm
3 receives said shaft at one end thereof is supported
4 upon a pivot point intermediate the ends thereof and
5 receives a spring means at the other end thereof
6 biasing said one end toward said shaft.

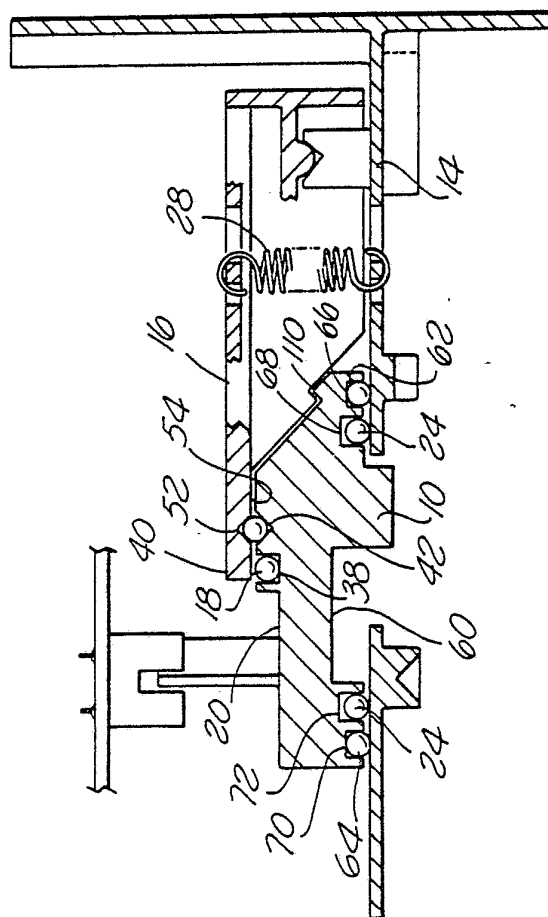
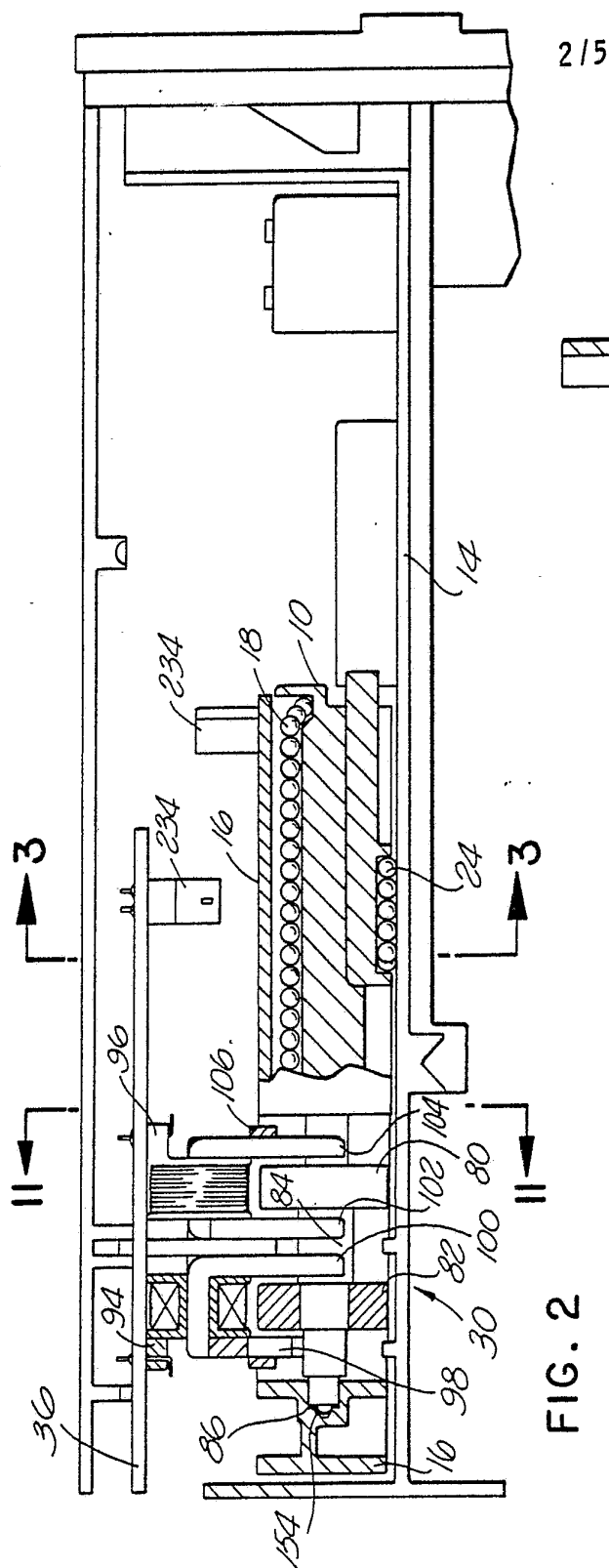
1 23. Magnetic transducer positioning
2 apparatus as defined in Claim 22 which further in-
3 cludes a plurality of balls contacting each of said
4 first and second surfaces of said carriage and which
5 further includes continuous track defining means
6 restraining said balls for movement along predeter-
7 mined paths to effectively float said carriage be-
8 tween said balls.

1 24. Magnetic transducer positioning
2 apparatus as defined in Claim 19 which further in-
3 cludes stop means for limiting the travel of said
4 carriage beyond first and second limit positions,
5 said spring loaded arm absorbing displacement of
6 said shaft as said shaft urges said ball against
7 said force receiving surfaces.



FIG. 1





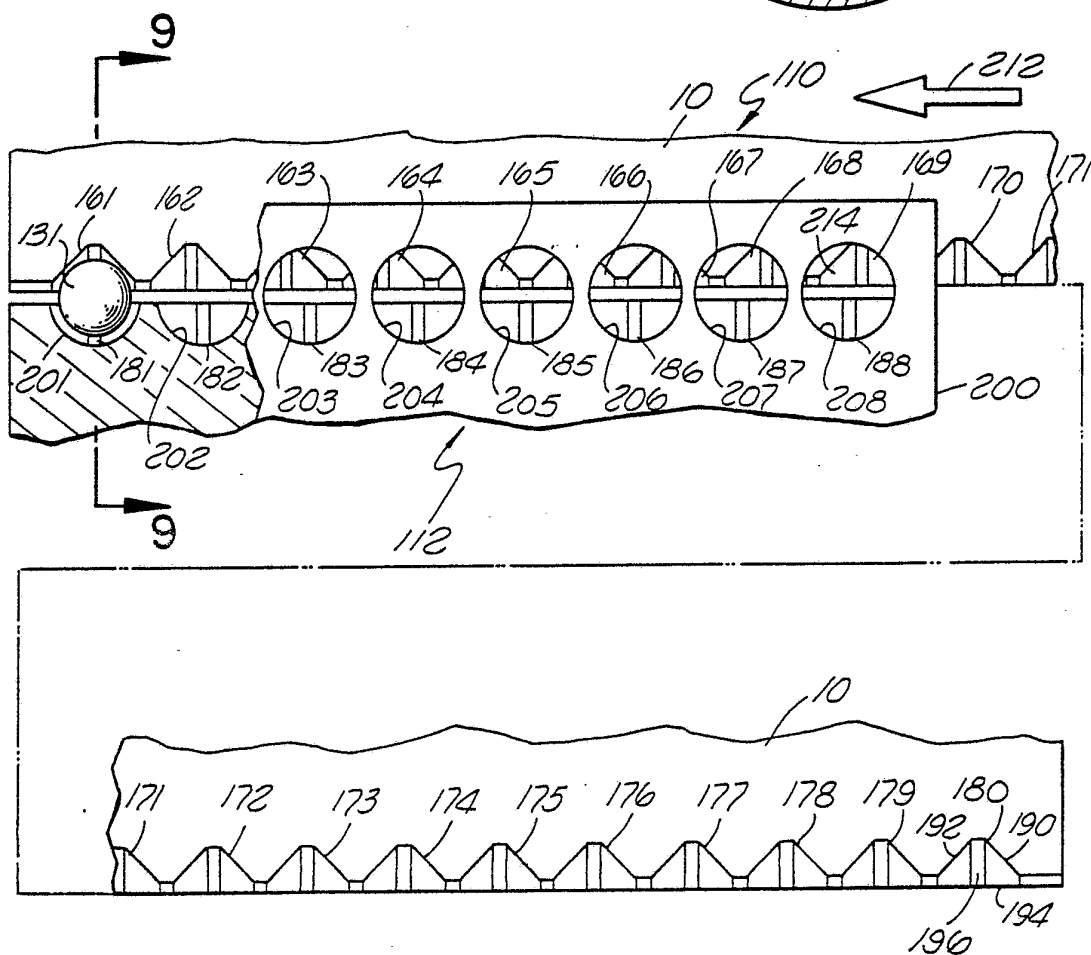
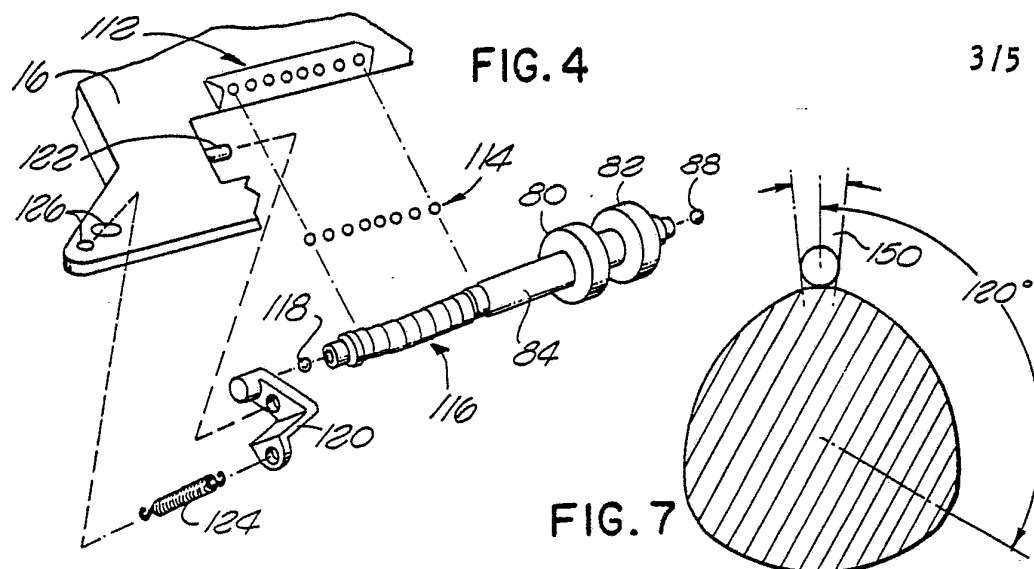


FIG. 8

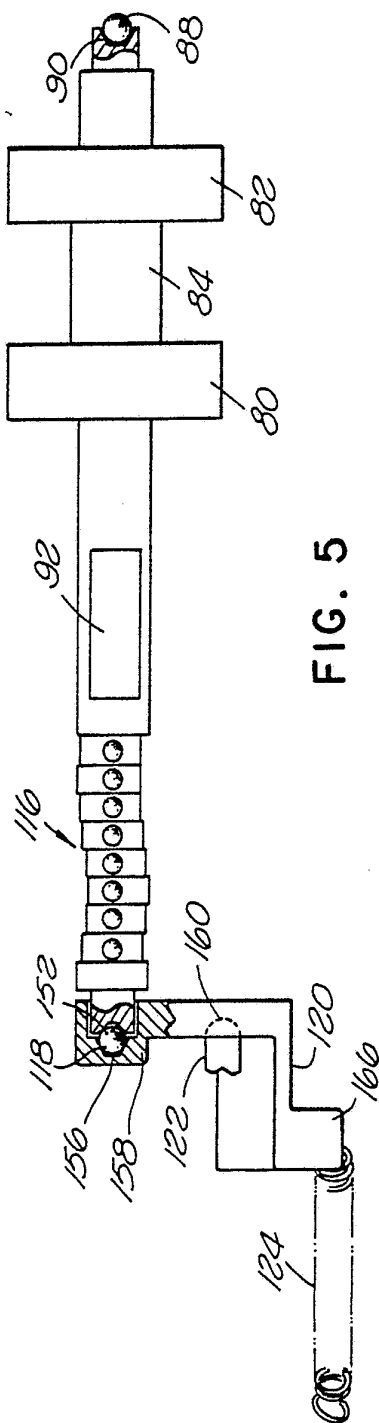


FIG. 5

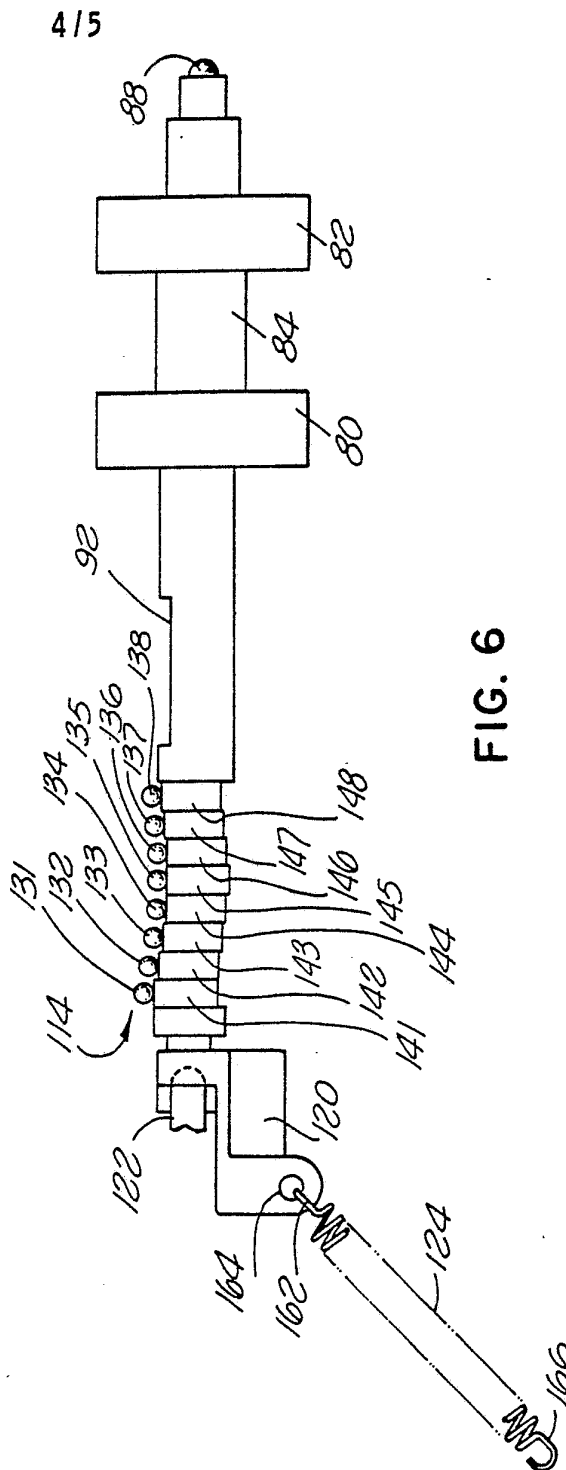


FIG. 6

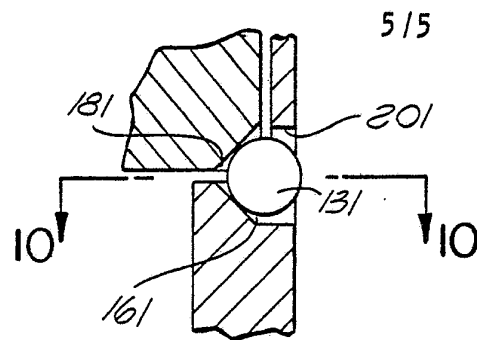


FIG. 9

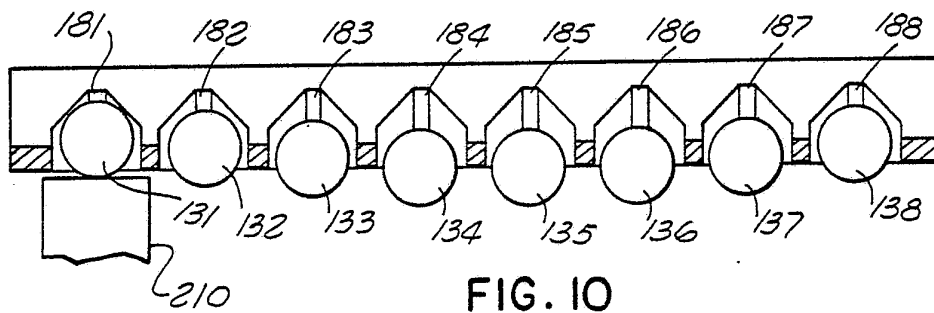


FIG. 10

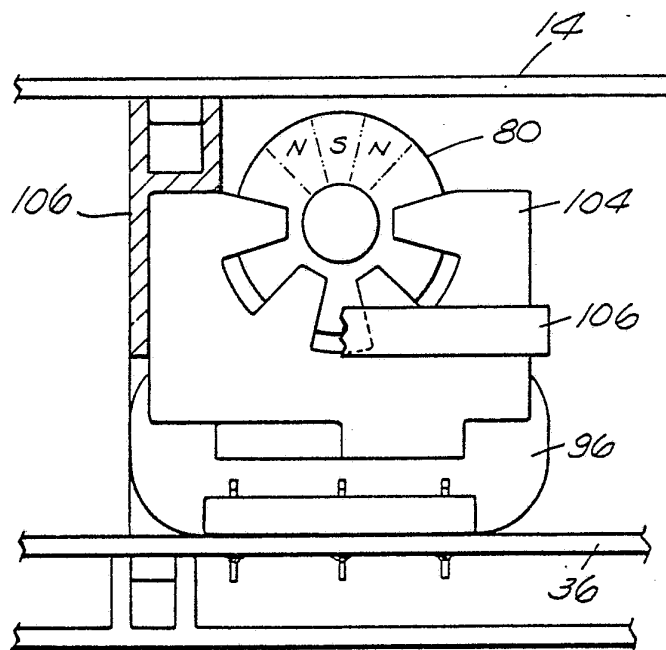


FIG. 11

INTERNATIONAL SEARCH REPORT

International Application No **PCT/US84/00147**

I. CLASSIFICATION OF SUBJECT MATTER (if several classification symbols apply, indicate all) ³		
According to International Patent Classification (IPC) or to both National Classification and IPC INT. CL. G11B 5/54, 21/08 U.S. CL. 360/106, 109		
II. FIELDS SEARCHED		
Minimum Documentation Searched ⁴		
Classification System	Classification Symbols	
U.S.	360/75, 78, 105-106 & 109. 74/567-569, 89.2	
Documentation Searched other than Minimum Documentation to the Extent that such Documents are Included in the Fields Searched ⁵		
III. DOCUMENTS CONSIDERED TO BE RELEVANT ¹⁴		
Category *	Citation of Document, ¹⁶ with indication, where appropriate, of the relevant passages ¹⁷	Relevant to Claim No. ¹⁸
X, P	US, A, 4,428,012 (Applequist et al) January 24, 1984	1
A	US, A, 4,270,155 (Bejerano) May 26, 1981	-
A	US, A, 4,261,222 (Datillo) April 14, 1981	-
A	US, A, 4,131,923 (Wachs et al) December 26, 1978	-
A	US, A, 3,859,662 (Habich et al) January 7, 1975	-
A	US, A, 3,818,500 (Lemelson et al) June 18, 1974	-
* Special categories of cited documents: ¹⁵ "A" document defining the general state of the art which is not considered to be of particular relevance "E" earlier document but published on or after the international filing date "L" document which may throw doubts on priority claim(s) or which is cited to establish the publication date of another citation or other special reason (as specified) "O" document referring to an oral disclosure, use, exhibition or other means "P" document published prior to the international filing date but later than the priority date claimed "T" later document published after the international filing date or priority date and not in conflict with the application but cited to understand the principle or theory underlying the invention "X" document of particular relevance; the claimed invention cannot be considered novel or cannot be considered to involve an inventive step "Y" document of particular relevance; the claimed invention cannot be considered to involve an inventive step when the document is combined with one or more other such documents, such combination being obvious to a person skilled in the art. "&" document member of the same patent family		
IV. CERTIFICATION		
Date of the Actual Completion of the International Search ²	Date of Mailing of this International Search Report ²	
February 29, 1984	23 MAR 1984	
International Searching Authority ¹	Signature of Authorized Officer ²⁰	
ISA/US	John H. Wolff	