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DISC STORAGE SYSTEM WITH HYDRODYNAMIC BEARING

The present invention relates to magnetic disc drive storage systems. More specifically, the present invention relates to a hydrodynamic bearing for use in a magnetic disc drive storage system

Magnetic disc drives are used for magnetically storing information. In a magnetic disc drive, a magnetic disc rotates at high speed and a transducing head "flies" over a surface of the disc. The transducing head records information on the disc surface by impressing a magnetic field on the disc. Information is read back using the head by detecting magnetization of the disc surface. The transducing head is moved radially across the surface of the disc so that different data tracks can be read back.

Over the years, storage density has tended to increase and the size of the storage system has tended to decrease. This trend has lead to greater precision and lower tolerance in the manufacturing and operating of magnetic storage discs. For example, to achieve increased storage densities the transducing head must be placed increasingly close to the surface of the storage disc. This proximity requires that the disc rotate substantially in a single plane. A slight wobble or run-out in disc rotation can cause the surface of the disc to contact the transducing head. This is known as a "crash" and can damage the transducing head and surface of the storage disc resulting in loss of data.

From the foregoing discussion, it can be seen that the bearing assembly which supports the storage disc is of critical importance. One typical bearing assembly comprises ball bearings supported between a pair of races which allow a hub of the

storage disc to rotate relative to a fixed member. However, ball bearing assemblies have many mechanical problems such as wear, run-out and manufacturing difficulties. Thus, there has been search for alternative bearing assemblies for use with high density magnetic storage discs.

One alternative bearing design which has been investigated is a hydrodynamic bearing. In a hydrodynamic bearing, a lubricating fluid such as air or liquid provides a bearing surface between a fixed member of the housing and a rotating member of the disc hub. In addition to air, typical lubricants include oil or ferromagnetic fluids. Hydrodynamic bearings spread the bearing interface over a large surface area in comparison with a ball bearing assembly which comprises a series of point interfaces. This is desirable because the increased bearing surface reduces wobble or run-out between the rotating and fixed members.

However, hydrodynamic bearings themselves suffer from a number of disadvantages. These disadvantages include a low stiffness-to-power ratio and a likelihood of damage due to gyroscopic precession. These problems lead to a high sensitivity of the bearing to external loads or shock.

Figure 2 is a cross-sectional view of a prior art drive spindle assembly 30 having a hydrodynamic bearing. Spindle assembly 30 includes a rotating hub 32 which rotates relative to a fixed shaft 34. Hub 32 includes sleeve 33 which receives a grooved journal 36 of fixed shaft 34. The hydrodynamic bearing includes counterplate 38 coupled to hub 32 and thrust plate 40 coupled to shaft 34. A hydrodynamic fluid 42 is carried in gaps between the fixed portion of the hydrodynamic bearing and the rotating portion. In operation, as hub 32 rotates the hydrodynamic fluid 42 is caused to circulate through the hydrodynamic bearing. The

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hydrodynamic fluid separates the rotating portions from the fixed portions and provides a bearing surface therebetween.

Figure 2 shows dimensions d_2 and e_2. Dimension d2 is the diameter of thrust plate 40 and e_2 is the thickness of thrust plate 40. Figure 2 also shows force P_{R2}.

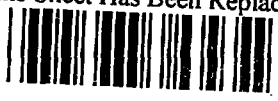
According to one aspect of the present invention there is provided apparatus for supporting at least one disc for rotation in a storage system, the apparatus comprising a rotatable part for connection to the discs and a fixed part, between which is formed hydrodynamic bearing, characterised in that, at the interface between the rotatable and fixed parts forming the hydrodynamic bearing, the width of the rotatable and fixed parts varies along the axis of rotation of the rotatable part .

One arrangement of the magnetic disc storage system includes a base and a rotatable magnetic storage disc having an axis of rotation. A transducing head proximate a surface of the disc is used for reading and writing information on the disc surface. A rotating member attached to the disc rotates about a fixed member attached to the base on an axis of rotation. The rotating and fixed members form a hydrodynamic bearing therebetween having a surface which forms an interface between the two members. Angles of the interface relative to the axis of rotation are acute. In one embodiment, a width of the inner face proximate a central portion of the bearing is less than a width of the inner face distal the central portion. Width is measured in a direction perpendicular to the axis of rotation.

In one embodiment, the interface forms a portion of a sphere which rotates in a sphere-shaped sleeve.

In one embodiment, the hydrodynamic bearing is manufactured using a process involving metal injection moulding and may include grooves therein for interacting with a fluid. The axial position of the bearing is determined using a self-positioning

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technique in which the bearing is automatically positioned within the disc storage system such that tolerances are less critical. In one embodiment, this includes injecting air into the hydrodynamic bearing such that a gap of the interface is set within predetermined limits.

The fixed member of the storage system may be a shaft. An upper seal may be provided, and the shaft may extend therethrough for coupling for to a cover of the storage system.

Alternatively, the rotating member of the storage system may be a shaft.

The shaft may have a first spherical member attached at one end and a second spherical member attached at the other end. The member which co-operates with the shaft may comprise a sleeve which is shaped for receiving the first and second members. The spherical members may be attached to the shaft by an adhesive. Also, a coefficient of thermal expansion of the sleeve may be greater than a coefficient of thermal expansion of the shaft. The sleeve may include an expansion sleeve extending between portions for receiving the first and second spherical members, and which has a coefficient of expansion greater than a coefficient of expansion of the portions receiving the first and second spherical members. In this instance, the first and second spherical members and the shaft may have expansion coefficient substantially equal to the expansion coefficient of the expansion sleeve.

The interface may have a width at a proximal end reletive to the rotating member which is less than a width at a distal end relative to the rotating member.

The hydrodynamic bearing may include a curved portion received in a curved sleeve forming a gap therebetween for circulation of hydrodynamic fluid.

The storage system may be one using either magnetic discs or optical discs.

The hydrodynamic bearing assembly may provide an increased restoring force in response to wobble, run-out or gyroscopic precession of the bearing.

For a better understanding of the present invention, embodiments it will now be described by way of example, with reference to the accompanying drawings, in which:

Figure 1 is an exploded perspective view of a magnetic disc storage system including a hydrodynamic bearing in accordance with the invention.

Figure 2 is a cross-sectional view of a prior art spindle assembly.

Figure 3 is a cross-sectional view of the spindle assembly of Figure 1.

Figure 4 is a cross-sectional view of a hydrodynamic bearing cartridge in a rotating shaft design.

Figure 5 is a cross-sectional view of a hydrodynamic bearing cartridge in a fixed shaft design.

Figure 6 is a cross-sectional view depicting assembly of a portion of a hydrodynamic bearing.

Figure 7 is a cross-sectional view showing assembly of a hydrodynamic bearing in accordance with the invention.

Figure 8 is a cross-sectional view of a spindle assembly in accordance with another embodiment.

Figures 9A and 9B are cross-sectional views of another embodiment illustrative of a technique for thermo-compensation.

Figure 1 is an exploded perspective view of a magnetic disc storage system 10 in accordance with the present invention. Storage system 10 includes base 12 having drive spindle 14 which rotatably carries storage discs 16. An armature assembly 18 moves transducing heads 20 across surfaces of discs 16. The environment of discs 16

is sealed by seal 22 and cover 24. In operation, discs 16 rotate at high speed while transducing heads 20 are moved radially across the surfaces of discs 16. This allows transducing heads 20 to read and write magnetically encoded information on surfaces of discs 16 at selected locations.

Figure 3 is a cross-sectional view of spindle 50 for disc drive storage system 10, shown in Figure 1. Spindle 50 includes rotor 14 rotatably coupled to shaft 52. Shaft 52 includes upper hemisphere portion 54 and lower hemisphere portion 56 received in sleeve 58. Shaft 52 is fixedly attached to base 60. Sleeve 58 receives journal 62 of shaft 52 and has upper hemisphere-shaped receptacle 64 and lower hemisphere-shaped receptacle 66. A fill hole 68 is also provided to fixed member 52 and rotor 14 includes counterplate 70.

Figure 3 also shows dimensions e_1 and d_1. Dimension e_1 is the thickness of hemispheres 54 and 56, and d_1 is the distance of a line which is tangent to a hemisphere 54,56 and which passes through the center of fixed member 52. Figure 3 also shows a force P_{R1}. As shown in Figure 3, the hydrodynamic interface 55 formed by portions 54 and 56 forms an acute angle with the axis of rotation 57 of spindle 50.

In operation, the bearings shown in both Figure 2 and Figure 3 are hydrodynamic bearings in which a lubricating fluid such as oil circulates through gaps between the fixed member and the rotating member. Rotation of the rotating member causes the lubricating fluid to circulate. As shown in Figure 3, oil is trapped in the hydrodynamic bearing by a diverging capillary seal 72.

The invention resulted from a desire to improve the performance of hydrodynamic bearings with regard to external loads and shock. One aspect of the invention is directed to reducing the effects of gyroscopic precession on the motor.

Another aspect of the invention is directed to strengthening the motor in terms of its ability to withstand external loads and shock without sustaining damage.

The embodiments described accomplish these two primary goals through a number of aspects which include:

1. Increasing the stiffness to power ratio (rocking mode) of the bearing.
2. Increasing the bearing contact area and ensuring no line contact.
3. Increasing the bending strength and stiffness of the shaft/bearing structure.

The angular stiffness to power ratio of a hydrodynamic bearing is proportional to $(\text{length})^2 \div (\text{gap})^2$, where the length is the moment arm (d) of the restoring force and the gap refers to oil thickness within the bearing. As further reduction in gap size is difficult, the length term is altered in the embodiments. As the height of the motor is generally fixed, the motor topology is changed. This was done by incorporating a bi-spherical configuration, as shown in Figure 3.

As shown in Figure 3, a "bi-spherical" hydrodynamic bearing is provided. The stiffness of the bi-spherical bearing is increased over that of a bearing with a flat thrust plate, as shown in Figure 2. In the bearing with a flat thrust plate, the angular restoring moment ($dM/d\theta$) is a function of the thrust plate diameter d_2 but not the overall length of the bearing or the distance from the thrust plate to the center of the bearing journal. This is not the case for the bi-spherical bearing. As can be seen from Figure 3, the moment arm d_1 is a line drawn tangent to the hemisphere which passes through the journal's center. As the overall bearing length increases, the stiffness to power ratio will go up by $(\text{length})^2$. As compared to the bearing shown in Figure 2, the bi-spherical bearing in Figure 3 will have an angular stiffness increase of 20% with no increase in power consumption.

The threshold for shock, or contact related damage, is increased as the contact surfaces are essentially ball and socket joints. In other words, contact is made between two spherical surfaces with almost identical radii, separated by a film of oil.

The increased structural strength of the bearing may be provided by three factors:

1. The narrowest portion of the shaft (the journal center) is shorter in the bi-spherical design.

2. The stress concentrating geometry at the bottom of the shaft has been removed.

3. The hemispheres are approximately twice as thick as the existing thrust plates for the same size motor, thus the retaining force due to a press fit, or adhesive, will be twice as great. Also, the thicker cross section should allow a greater press fit interference with less distortion of the hemisphere.

The following equations show an advantage of the embodiment as shown in Figure 3 in comparison to the prior art shown in Figure 2. The restoring moments for the bearings shown in Figures 3 and 2 are, respectively:

$$M_{R1} = 2d_1P_{R1} \qquad \text{Eq. 1}$$

$$M_{R2} = 2d_2P_{R2} \qquad \text{Eq. 2}$$

For the examples shown in Figures 3 and 2, $d_1 = 2d_2$ such that if the forces P_{R1} and P_{R2} are the same, $M_{R1} = 2M_{R2}$. As shown in the Figures, this advantage becomes greater as the overall height of the journal increases. In this example, the stiffness of the bearing of Figure 3 is twice that of Figure 2. However, the power loss is slightly greater using the bearing of Figure 3 than that of a typical prior art bearing. This power loss is much less than twice the power loss of the conventional bearing of Figure 2. The increased power loss is due to the increased surface area of

the bi-spherical bearing. This, for example, assumes the spherical bearing has a diameter of 6.6 mm versus an 8 mm diameter thrust plate of Figure 2. Using these diameters, increasing the bearing length by 4.3 mm increases the angular stiffness approximately 30%. If an 8 mm diameter hemisphere is used and the bearing spacing is not increased, the angular stiffness will increase over 200% (as will the power consumption).

The embodiment as shown in Figure 3 also has improved alignment and strength over a conventional design as shown in Figure 2. This is because the engagement thickness e_1 of spindle 50 is twice the engagement thickness e_2 of conventional spindle 30. The increased thickness e_1, where a hemisphere attaches to the shaft, provides a stronger connection between the hemisphere and the shaft in comparison to the prior art thrust plate and shaft shown in Figure 2. This increased area can be used for bonding the hemisphere to the shaft. Another advantage of the embodiment is that during tilt (gyroscopic precession), contact between a hemisphere portion and the sleeve will not be an edge contact as in a conventional design. Instead, the contact will be one smooth surface rubbing against another smooth surface, for example the curved surfaces of the hemispheres. This causes the load to be spread out over a much greater area than in typical prior art designs.

The present invention also includes methods for assembling the hydrodynamic bearing assembly shown in Figure 3. In one embodiment, sleeve 58 is made of aluminum which is heated to a temperature above ambient temperature, such as 90° C above ambient temperature, so that it expands. For example, if the expansion were 0.8 μm/mm and the sleeve had a height of approximately 10 mm, the overall expansion would be approximately 8 μm, or 4 μm for the top side and 4 μm for the bottom side. While the sleeve is heated, the hydrodynamic bearing could be

assembled such that hemisphere portions 54 and 56 are allowed to contact sleeve 58. As the sleeve cools, it will leave a 4 μm gap between each hemisphere and the sleeve. This concept can be extended by using different temperatures or other materials to achieve the desired interface gap. Similar materials can be used, but a greater temperature difference would be required. Example techniques for attaching a hemisphere to a shaft include: bonding, a press fitting, swaging, threading (i.e. the two parts are screwed together) or laser welding.

Another embodiment includes making the hydrodynamic bearing as a discrete cartridge. This is attractive in a rotating shaft design because the hub is attached after the bearing is assembled. Such a design is shown at Figure 4. Figure 4 is a cross-sectional view of drive spindle assembly 100 for a rotating shaft design. Spindle assembly 100 includes hub 102 attached to shaft 104. Hemisphere portions 106 and 108 are connected to the top and bottom of shaft 104, respectively. Portions 106 and 108 and shaft 104 are received in fixed sleeve 110 which includes hemisphere receiving portions 112 and 114 for receiving portions 106 and 108, respectively, and journal receiving portion 116 for receiving journal 118 of shaft 104. In the embodiment of Figure 4, shaft 104 and hemisphere portions 106 and 108 rotate in sleeve 110 along a hydrodynamic interface thereby allowing hub 102 to rotate. In this embodiment, hub 102 can be bonded, screwed or otherwise attached to shaft 104 of the bearing cartridge following assembly of the bearing cartridge.

Figure 5 is a cross-sectional view of a cartridge-type design hydrodynamic bearing having a fixed shaft 120 which is attached to base 121. Shaft 120 is attached to hemispherical portions 122 and 123 and is received in rotating sleeve 124. Oil circulates through passageway 126. Oil within the hydrodynamic bearing is retained using diverging capillary seals 125.

One technique for forming a hemisphere in accordance with the invention is metal injection molding. Figure 6 is a cross-sectional view of a hemispherical portion formed by metal injection molding. In Figure 6, hemisphere portion 130 is shown prior to mounting on shaft 132. Radial oil grooves 134 are formed in the top of portion 130 during molding. Portion 130 includes relief area 136 for oil backflow. Additionally, relief area 136 distributes forces on the outer diameter of portion 130 when portion 130 is used as a support surface. Figure 6 shows a shoulder mount configuration where portion 130 rests on a shoulder of shaft 132. Attachment of portion 130 to shaft 132 may be through, for example, adhesive bonding.

Another technique for manufacturing a hydrodynamic bearing is shown in Figure 7. In this design, pressurized air is applied to inlet 140 of shaft 142 which is received in sleeve 144. Shaft 142 is coupled to spherical portions 146 and 148. Adhesive is applied between portions 146 and 148 and shaft 142. O-ring seals 150 and 152 retain portions 146 and 148 relative to shaft 142. A force F_S is applied to portions 146 and 148. This force balances the air pressure applied at 140 and is set to a predetermined limit to determine the size of gap 154. The flow rate of the applied air may be monitored to determine the size of gap 154. As an alternative to applying a known force F_S to the bearing, members 146 and 148 can be held by a clamp which allows a sufficient gap 154 to form when pressurized air is applied. The size of the gap may be measured by monitoring the flow rate of the applied air. The adhesive used for bonding should be a low viscosity anaerobic adhesive. It should be placed along the inner diameter of the spherical members 146 and 148 prior to inserting the members onto shaft 142. The positive air pressure prevents adhesive from flowing into the hydrodynamic bearing area while O-rings 150 and 152 prevent the adhesive from seeping out of the adhesion area.

Figure 8 is a cross-sectional view of spindle assembly 160 in accordance with another embodiment. Spindle 160 includes rotor hub 162 which carries disc 164, and is rotatably coupled to shaft 166. Shaft 166 includes upper hemisphere 168 and lower hemisphere 170 received in cartridge sleeve 172. Sleeve 172 receives journal 174 of shaft 166. The top portion of shaft 166 is sealed with seal 176, and journal 174 extends therethrough. Hub spacer 178 is coupled to rotor hub 162, and carries permanent magnets 180. Shaft 166 is received in base 182, and includes fill hole 184. Fill hole 184 connects to passageway 186, which extends through the center of shaft 166. Passageway 186 connects to center passageway 188 and distal passageways 190. In operation, oil flows into passageway 186 through center passageway 188 and out distal passageways 190. This is indicated by arrows shown in passageway 186.

Figures 9A and 9B are cross-sectional views of spindle 200 in accordance with another embodiment, and are positioned to illustrate a technique for thermo-compensation. It has been noted that the viscosity of the lubricant changes over the operating range of a hydrodynamic bearing. This change has a pronounced effect on the load capacity, stiffness, and power consumption of the bearing. Figures 9A and 9B illustrate a technique for compensating for the change in lubricant viscosity by changing the gap size in the hydrodynamic bearing. In general, the gap size should be reduced as the temperature of the bearing is increased. In Figures 9A and 9B, spindle 200 includes expansion sleeve 202, sleeve 204 and shaft assembly 206. Thermo-compensation is achieved by utilizing a material having a low expansion coefficient for sleeve 204. Similarly, expansion sleeve 202 and shaft assembly 206 (including the hemispheres) are formed of a material having a higher expansion coefficient. Figure 9A shows spindle 200 prior to expansion, while Figure 9B shows

spindle 200 at a greater temperature than in Figure 9A following thermo-expansion. Line 208 shows the relevant change in position due to expansion between the two Figures. During the process of heating, the bearing seats expanded at a much slower rate than the rest of the components. Expansion sleeve 202 spreads the bearing seats apart at the same rate that the shaft and bearing hemispheres expand. This causes the bearing seats to effectively shrink towards the hemispheres. This causes a substantially uniform closure of the gap. The technique is independent of shaft and sleeve length because they are made of the same material. If the difference in thermo-expansion coefficients between the two different materials is about 14×10^{-6} $°C^{-1}$, a change in temperature from 25° C to 70° C will result in a 3.5 mm radius hemispherical gap closing approximately 2.2 μm. These coefficients of expansion are provided by aluminum and 400 series stainless steel, respectively. Note that the expansion sleeve 202 should be flexible in the radial direction, where it attaches to the bearing seat, in order to prevent the bearing seat from distorting as the assembly changes temperature.

In the embodiments described, the interface of the hydrodynamic bearing is extended over a greater area than in prior art designs. This provides increased bearing stiffness and stability. Further, the restoring moment is determined based upon the length of a line which extends from the center of the journal to an outer edge of the bearing member. In typical prior art designs, the hydrodynamic bearing interface is at a perpendicular angle with the axis of rotation. However, in the present invention, an angle between the interface and the axis of rotation is acute, i.e. it is not parallel. This is achieved by making the bearing have a narrow width at an inner side and widen towards an outer edge of the bearing. The curved surface of the bearing also facilitates smooth operation and reduces the effects of external loads or

shock or precession. Generally, forces on the bearing cause the hemisphere to rotate within the receiving portion about its center. It will be apparent that this design is less likely to be damaged than prior art flat thrust plate designs in which a small tilt of the drive shaft can cause the tip of the long level arm of the thrust plate to travel a relatively large distance.

Although the present invention has been described with reference to preferred embodiments, workers skilled in the art will recognize that changes may be made in form and detail without departing from the spirit and scope of the invention. For example, the hydrodynamic bearing of the invention is applicable to other typical spindle assemblies used in magnetic disc storage systems.

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CLAIMS:

1. Apparatus for supporting at least one disc for rotation in a storage system, the apparatus comprising a rotatable part for connection to the discs and a fixed part, between which is formed hydrodynamic bearing, characterised in that, at the interface between the rotatable and fixed parts forming the hydrodynamic bearing, the width of the rotatable and fixed parts varies along the axis of rotation of the rotatable part .

2. Apparatus according to claim 1 wherein, the interface is curved.

3. Apparatus according to claim 2, wherein the interface forms a portion of a sphere.

4. Apparatus according to claim 1, 2 or 3, wherein the interface forms an acute angle with the axis of rotation.

5. Apparatus according to any one of claims 1, 2, 3 or 4, wherein one of the rotatable part and the fixed part comprises an elongated shaft having a dome-shaped member at one end thereof, and the other of the rotatable part and the fixed part comprises a sleeve shaped for receiving the dome-shaped member.

6. Apparatus according to claims 5, wherein a fluid path is formed through the shaft and around the dome-shaped member between the shaft and the sleeve.

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7. A method of making a device for supporting at least one disc for rotation in a storage system, the method including providing a first part for connection to the discs and a second part, assembling the first and second parts to form a hydrodynamic bearing therebetween to allow relative rotation of the parts, chartecterised in that, at the interface between the first and second parts forming the hydrodynamic bearing, the width of the first and second parts varies along the axis of rotation of the rotatable part .

8. A method according to claim 7, including injecting a fluid under pressure into the hydrodynamic bearing to provide a predetermined gap size between the parts and the interface, and applying a known force to one of the parts to counter the force of the fluid and/or adjusting the rate of fluid flow to determine the gap size.

9. A disc storage system comprising apparatus according to any one of claims 1 to 7, or apparatus manufactured according to the method of claim 8 or 9.

10. A magnetic disc storage system, comprising:

a base;

a rotatable magnetic storage disc having an axis of rotation;

a transducing head for reading and writing information on the disc;

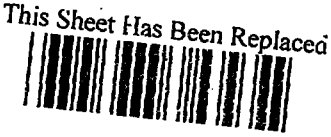
a fixed member attached to the base;

a rotating member attached to the disc which rotates relative to the fixed member on the asis of rotation, the rotating member forcing a hydrodynamic bearing over a bearing surface which forms an interface with the fixed member: and

wherein an angle between the interface and the axis of rotation is acute.



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11. A magnetic disc storage system, comprising:

a base;

a rotatable storage disc having an axis of rotation;

an transducing head proximate a surface of the disc for reading and writing information on the disc;

a fixed member attached to the base;

a rotating member attached to the disc which rotates relative to the fixed member on the axis of rotation, the rotating member and the fixed member forming an elongated interface therebetween having a center portion and first and second edge portions aligned with and spaced apart on the axis of rotation; and

a hydrodynamic bearing at the first edge of the interface, the bearing having an inner width proximate the center portion in a direction perpendicular with the axis of rotation which is less than an outer width distal the center portion.

