A hydrostatic rotary bearing with angled surface self-compensation

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Abstract

A design for a low profile hydrostatic rotary bearing is presented that utilizes a new type of self-compensation that allows a large number of bearing pockets to be present with only a few precision round parts, thus allowing high hydrostatic averaging with relatively low design complexity. A key feature of this design is that it does not use capillaries or orifices or a complex network of passageways to route fluid. The design consists of five easily assembled round parts which can be manufactured by cylindrical grinding. A seal is incorporated which prevents fluid leakage and creates a closed system. The theory of operation is discussed along with manufacturing considerations. Tests show that the rotary bearing has high stiffness, and achieves a high degree of averaging, resulting in a radial error motion of 0.05 μm from components ground to an accuracy of only 2.5 μm.

Keywords: Hydrostatic bearing; Rotary bearing; Rotary table; Self-compensation; Hydrostatic averaging

1. Background

In order to exhibit static stiffness, hydrostatic bearings must regulate flow into the pockets with some kind of restrictor or feedback system. This enables the bearing to counteract externally applied loads by varying the fluid pressure in individual bearing pockets [1,2]. Many hydrostatic bearings in machine tool applications use fixed resistance restrictors such as orifices or capillaries whose resistances are nominally equal to the flow resistance out of the bearing pocket. However, in order to achieve accuracy, the restrictors’ flow resistance must all be equal or of a specific ratio. Since capillary resistance, for example, varies with the fourth power of the diameter, tuning all the restrictors can be time consuming. Since one restrictor is required for each bearing pocket, the desire of having as many bearing pockets as possible in order to enhance averaging and improve accuracy greatly increases cost. Thus, rolling element bearings are often used whenever possible in machine tools [3,4].

Nevertheless, hydrostatic bearings’ advantages and disadvantages were recognized early, and in the 1940s, Hoffer was apparently the first to propose using the opposed gap as a means to regulate flow to pockets on the opposite side of a bearing [5]. Hence, Hoffer’s bearings were self-compensating. In the 1960s, Arneson [6,7] introduced an atypical aerostatic bearing design that achieved compensation by using grooves of a precise depth on the surface of a shaft which acted as flow restrictors. This form of regulating the flow on the surface also eliminated the need for separate restrictors. However, since the grooves act as restrictors, they must be machined or etched to a very precise depth and width that is matched to the radial clearance. Professional Instruments Company developed a highly successful and refined process for making the grooves that few have been able to duplicate, thus allowing them to establish a substantial market for their BlockHead® aerostatic spindles.

But hydrostatics offer substantially greater load capacity than aerostatic bearings, and Hoffer’s self-compensation method was refined by Hedberg [8] and incorporated into many different types of precision grinding machines developed by Lidköping primarily for machines used for grinding bearing rings. Other variants of self-compensation designs were developed, such as by Zollern Vertriebs-GmbH. Self-compensation was also to be highly developed in the former USSR, where it was also used mainly for precision grinding and diamond turning machines [9]. Other refinements of self-compensation were also developed [10,11], but required either cross-drilled holes or external plumbing to route the fluid from the compensating structures on one side of the bearing to the pockets on the other side.
Self-compensated bearings are less prone to clogging and can have fewer parts; however, their primary advantage may be that their stiffness is not adversely affected by bearing gaps that are smaller or larger than intended [3]; however, their stiffness is still finite and generally lower than ball or roller bearings; hence servostatic bearings were developed, where the fluid flow to the pockets was actively regulated by measurement of bearing gaps and the use of servo valves to achieve “infinite” stiffness [12–15]. On the other hand, the rest of the machine structure is never infinitely stiff, and a valve on every pocket can become very expensive very quickly.

With the exception of Arneson’s design, previous self-compensation designs required cross-drilling or the use of external fluid lines to connect the compensator to the opposed pad. Other designs evolved this general principle to create, for example, a thrust bearing where the compensation for the thrust lands came from features on the shaft radius [16]. This was a forerunner of the present design; however, these designs still required the groove depths to be carefully tuned to the radial clearance. Wasson and Slocum [17] ultimately created the first true surface self-compensating bearing where the compensating features are located opposite the pockets, so compensation is gap-independent and the compensating features are then connected to the pockets via channels on the surface of the bearing. Kotilainen and Slocum developed the high-speed flow theory for this design concept, and showed that it was robust enough that it could even be cast, including all the pockets and compensation features [18]. Furthermore, Kane and Slocum evolved Wasson’s surface self-compensation design to create a modular profile rail hydrosstatic bearing [19]. These designs, however, do not lend themselves to low profile rotary tables, and hence angular surface self-compensated rotary bearings were developed.

2. Angled surface self-compensation

The design presented here incorporates many of the attributes of previous self-compensated and surface compensated designs into a low-profile geometry. The important innovative feature is a restrictor gap region that makes an acute angle relative to a bearing gap region with pockets whose depth is non-critical to the hydrostatic performance. An exploded view of a five piece rotary bearing that incorporates this concept is shown in Fig. 1.

The train of thought for many self-compensating designs is that the restrictor land must be either directly opposed to the bearing land it feeds, as is the case with the Hoffer design, or, as is the case with the Arneson design, the other extreme—a restricting slot that is on the same face. The design presented here lies somewhere between these two extremes—the restricting surface is not directly opposed, nor on the same face—but rather on a face that is at some angle in between, preferably an acute angle for better feedback efficiency. Compared to the more traditional opposed self-compensation schemes, the angled approach eliminates the need to connect each restrictor to each bearing pocket with a passageway—a very complex proposition if many pockets are present. Compared to the Arneson approach, the need to make slots that have a precise depth is eliminated, and the load bearing efficiency and stiffness is potentially better by a factor of two or more, for the following reasons. First, the feedback is more efficient in most directions, because the restrictors and bearing lands are close to being opposed and hence work synergistically, in contrast to the diminished feedback a slotted restrictor provides when it is co-planar with a bearing land. Second, the present design can have a small restrictor region and expansive pockets, resulting in an effective load bearing area that is around 80%, in contrast to the slotted restrictor approach, whose effective load bearing area inherently cannot be better than 50%.

The bearing in Fig. 1 comprises the stator, which is the largest part; the restrictor ring, which is shrunk fit into the stator; the rotor, which consists of two geometrically identical pieces that contain 20 bearing pockets, each 0.5 ± 0.2 mm deep; and the spacer which aligns and connects the two halves of the rotor. Fig. 2 shows a cross section of the assembly. Although it cannot be seen in the figure, the dimensions of the parts are chosen so that after assembly, a nominal restrictor and bearing land gap of 20 μm is present between the rotor and the stator.

It should be noted that for this design to function properly, the acute edge on each rotor half must be left sharp after grinding—otherwise the pockets will be shorted to one another and radial and tilt stiffness will be nearly zero. Therefore, the rotor halves must be handled with care after they are ground so that the acute edge remains sharp and unharmed.

Fig. 3 illustrates the fluid flow path through the bearing. The small arrows indicate the flow direction of the fluid during operation. After entering the bearing through the supply hole on the right side, the pressurized fluid fills the annular...
supply channel which is formed by the restrictor ring and the spacer. From there it flows through the restrictor gap, where its pressure drops a certain amount, then into window frame pockets, and then through the bearing gap, where its pressure drops to atmospheric in the drain channel. The depth of the pockets is at least 10 times larger than the bearing gap to ensure the pressure drop across them is negligible.

With this design, any small displacement or tilt of the rotor relative to the stator will cause changes in the bearing gaps and/or the restrictor gaps which will in turn cause an opposing force and moment to be generated on the rotor. Fig. 4a-c illustrate this feedback principle, with the bearing gaps exaggerated for clarity. Fig. 4a shows the case of radial displacement. On the right side, the bearing gaps close and the restrictor gaps open. Both of these changes act to increase the pocket pressures there. The opposite happens on the left side. As indicated in the figure, the resulting pressure difference creates a restoring force which opposes the perturbing force.

Self-compensation for vertical displacement and tilt is equivalent to fixed compensation because the restrictor gaps essentially remain constant while the bearing gaps change. In Fig. 4b, a vertical displacement is shown, and the effect is that the upper bearing gaps close and the lower ones open, resulting in a net restoring force. In Fig. 4c, a tilt displacement is shown, and the effect is a pressure change on diagonally opposite sides of the bearing, providing a net restoring moment.

3. Sealing

A major disadvantage of hydrostatic bearings is the potential risk of fluid leakage. This usually requires countermeasures such as seals and gutters to collect the fluid and prevent contamination. For the prototype, an integrated seal system was designed in order to make it a closed system so as to not expose any fluid to the environment. The seal consists of a narrow Buna-N rubber ring fitted into a groove in the stator and bonded in place with oil resistant adhesive, as shown in Fig. 5. The seal was made as small as practical to minimize intruding on valuable bearing land area. The seal is self-closing, i.e. increasing internal oil pressure in the drain channel increases the force of the sealing lip on the mating surface. Such an increase in drain channel pressure could occur if the drain line is very long, or if the hydraulic tank is located at a higher elevation than the bearing. Tests showed that the seal worked reliably in any spatial orientation of the bearing.

4. System model

Prior to the design of the prototype, a mathematical model was created to predict the following performance parameters of the bearing: stiffness, load capacity, flow rate, and pumping power. For this analysis, these parameters are computed for the case where the rotor is not moving relative to the stator. The results will also apply for slow rotor speeds for which hydrodynamic effects are negligible [18]. As a first order effort at predicting static performance, the bearing was modeled as a network of lumped regions [9], each with a
Due to the very small Reynolds number that occurs when oil is used, and the large width to length ratio of the major rectangular land regions, for the case where the rotor is fixed relative to the stator it is reasonable to treat flow over a land region as one dimensional Poiseuille flow between parallel plates. For one dimensional Poiseuille flow, the Navier–Stokes equation reduces to the well known form

$$\frac{\partial p}{\partial x} = \mu \frac{\partial^2 u}{\partial y^2} \quad (1)$$

where $\mu$ is the fluid viscosity, $p$ the fluid pressure, $x$ the coordinate in the direction of the flow, $u$ the fluid velocity in $x$, and $y$ is the coordinate direction perpendicular to $x$. By integrating twice and applying no slip boundary conditions at each of the stationary surfaces, the above equation can readily be solved to yield flow rate $Q$ through the gap in terms of the gap height $h$, land width $w$, land length $l$, fluid viscosity $\mu$ and pressure drop $(p_1 - p_2)$ that occurs in the direction of flow.

$$Q = \frac{w h^3}{12 \mu l} (p_1 - p_2) \quad (2)$$

In analogy to electrical networks, the flow rate $Q$ is equivalent to electrical current, the pressure difference $(p_1 - p_2)$ is equivalent to a potential difference, and the fluid resistance is

$$R = \frac{12h \mu l}{w h^2} \quad (3)$$

With this analogy in mind, the active bearing area is segmented into discrete rectangular land elements as shown in Fig. 6, and a resistor network is created. In Fig. 6, each resistor in the network represents a rectangular land element, nodes 1 to $n$ represent the $n$ bearing pockets, node ‘s’ represents the
supply channel, and node ‘d’ represents the drain channel. A gap \( h \) at the midpoint of each rectangular land element is used to compute a lumped resistance value for each land region.

The next step in the analysis is to find the gap \( h \) at the midpoint of each land element in terms of a small displacement and tilt of the rotor relative to the stator. Relative to an \( x, y, z \) coordinate system positioned with the \( z \)-axis along the stator axis, as shown in Fig. 7, a land element on the rotor can be located and oriented by the four parameters \( r, z, \alpha \), and \( \beta \) where \( r, z \) and \( \alpha \) locate the land midpoint and \( \beta \) orients the face in the radial cross section. A perturbing displacement of the rotor relative to the stator can be fully represented by linear displacements \( \delta_x, \delta_y, \delta_z \), and a tilt angle \( \delta \phi \) about an axis located by angle \( \gamma \). For small displacements, the gap \( h \) at the midpoint of a land element is given by

\[
h = h_0 - (\cos \alpha \cos \beta)\delta_x - (\sin \alpha \cos \beta)\delta_y - (\sin \beta)\delta_z - [r \sin(\alpha - \gamma) \sin \beta - z \sin(\alpha - \gamma) \cos \beta] \delta \phi
\]

(4)

where \( h_0 \) is the design gap present when the rotor is centered relative to the stator. Given the location parameters \( r, z, \alpha \), and \( \beta \) for each land and the rotor displacement components \( \delta_x, \delta_y, \delta_z, \delta \phi \), the midpoint gap \( h \) can readily be computed for each land element. Given the effective length and width of each land element and the fluid viscosity, \( R \) can be computed for each resistor in the Fig. 6 network for any small displacement of the rotor.

Next, the resistor network is used to solve for the pressures \( P_i \) at nodes 1 to \( n \), given the pressure at node 1, drain node d, and the resistances. This is a straightforward process of applying conservation of flow to each node, deriving a set of linear equations, putting them in matrix form, and solving for the unknown pressures. For simplicity of notation it is convenient to invert all resistances to obtain admittances.

\[
A_{ji} = \frac{1}{R_{ji}} \quad (5)
\]

Next, continuity at each node \( i \) dictates the following:

\[
Q_{i-1} + Q_{i+1} + Q_i = 0 \quad (6)
\]

\[
A_{ji}(P_i - P_j) + A_{i-1j}(P_{i-1} - P_j) + A_{i+1j}(P_{i+1} - P_j) + A_{ij}(P_d - P_i) = 0 \quad (7)
\]

where \( Q_{ji} \) is the flow from node \( j \) to node \( i \), \( A_{ji} \) is the admittance between nodes \( j \) and \( i \), and \( P_i \) is the pressure at node \( i \).
The above indexes are cyclical, meaning that an index equal to 0 should be replaced with \( n \), an index equal to \( n+1 \) should be replaced with 1. Setting the drain node pressure \( P_d \) to 0 and rearranging yields

\[
A_{l(j-1)} P_{j-1} + (−A_u − A_{l(j-1)} − A_{l(j+1)} − A_b) P_l
+ A_{l(j+1)} P_{j+1} = −A_u P_d
\]

(8)

Eq. (8) can be put in matrix form where \( \vec{P} \) is a vector of the unknown node pressures \( P_j \), and \( \vec{B} \) is a constant vector.

\[
[Z] \vec{P} = \vec{B}
\]

(9)

The non-zero terms of \( [Z] \) are the coefficients of the \( P_j \) values in Eq. (8):

\[
Z_{l(j-1)} = A_{l(j-1)}
\]

(11)

\[
Z_u = −A_u − A_{l(j-1)} − A_{l(j+1)} − A_b
\]

(12)

\[
Z_{l(j+1)} = A_{l(j+1)}
\]

(13)

The vector \( \vec{P} \) is readily found from

\[
\vec{P} = [Z]^{-1} \vec{B}
\]

(14)

The next step is to use the pocket pressures to compute the net force and moment on the rotor. For each rectangular land element, the normal force \( F_l \) is given by

\[
F_l = \frac{1}{2} w P_n + \frac{1}{2} l P_b
\]

(17)

where \( w \) and \( l \) are the land width and length, respectively, and \( P_n \) and \( P_b \) are the pressures present on opposite edges of the land. The normal force \( F_l \) on each pocket is given by

\[
F_p = w_p P_p
\]

(18)

where \( w_p \) is the pocket width, \( l_p \) is the pocket length, and \( P_p \) is the pocket pressure. The force and moment exerted on the rotor by a land element or a pocket element is given by

\[
F_x = −F \cos \alpha \cos \beta
\]

(19)

\[
F_y = −F \sin \alpha \cos \beta
\]

(20)

\[
F_z = −F \sin \beta
\]

(21)

\[
M_x = F_z \sin \alpha \cos \beta + r \sin \alpha \sin \beta
\]

(22)

\[
M_y = F_z \sin \alpha \cos \beta − r \sin \alpha \sin \beta
\]

(23)

\[
M_z = 0
\]

(24)

where \( F \) is the normal force on the element at hand, and \( r, z, \alpha \) and \( \beta \) locate the midpoint and orient the element face, as before. Given these equations, the net force and moment on the rotor can be computed by adding the contribution of each land and pocket element.

The model presented here can be used to iteratively compute the five displacements that result from an applied force and moment. This is useful for comparison with experimental data, where it is natural to apply a force and/or a moment to the rotor and to measure the resulting displacement.

The hydrostatic stiffness can be computed numerically at a given displacement by perturbing the displacement and computing the change in opposing force that results, and then dividing the opposing force change by the displacement change. The load capacity in a given direction can be computed by inputting a displacement in the direction of interest which closes the gap to some minimum allowed value, and then computing the net force in the direction of interest.

4.1. Compliance tests

Compliance tests were performed on a prototype bearing which had an overall diameter of 254 mm, a height of 86 mm, and a bore diameter of 56 mm. Referring to Figs. 8 and 9, the rotor was clamped down on a 50-mm thick cast iron base plate using a threaded clamping rod. A spacer disk prevented the stator from touching down on the base plate so it is free to move. All the loads for axial, radial, and tilt stiffness tests were applied through the stator of the bearing using appropriate fixturing. In each case, the force on the bearing was determined by using an S-shaped load cell with a maximum load capacity of 22,000 N and a sensitivity of 0.2 V/μN. The bearing’s compliance was determined at 10 and 40 bar supply pressures.

The measurement data showed significant deviation from the hydrostatic theory of up to 42%, depending on the loading case. The measured displacements were larger than predicted and it was suspected that the elastic compliance of the test set-up (which includes the elasticity of the external fixtures as well as the bearing parts themselves) was responsible. Examining the structural loop indicated that the extra compliance was likely due to spreading of the rotor components caused by the fluid pressure.

A general source of error, independent of the loading case, is the initial spread of the rotor halves when the supply pressure is turned on. The consequence of this deformation is that the bearing land gaps become effectively larger than the design calls for. As a result, the pocket pressures drop to lower levels than desired, which causes the actual bearing stiffness to decrease. The amount of initial spread was characterized using Mechanica™ finite element analysis software. An FEA
A simulation was run with a uniform pressure of 10 bar on each pocket, and the x and y deflection at the midpoints of upper and lower rotor bearing lands were reported. To compute the increase in the bearing land gaps, the displacements in x and y were decomposed into components perpendicular to the bearing land surface. If one takes the average of each, the top and the bottom half, the initial spread deformation translates into a bearing land gap opening of about 0.059 m/bar. This correction was implemented into the spreadsheet, although it did not account for the large discrepancy observed.

To predict to a first order the parasitic elastic compliance of the parts, lumped elastic compliance values were computed for the rotor by using deformations computed from several FEA simulations. The approach is described here step by step.

First, four FEA runs were performed on the rotor, each with a pocket pressure load set that corresponded to the four external load cases—no applied load, a radial external load, an axial external load, and an off center vertical force load. From the FEA results, a net reaction force was obtained, and the x and y displacements of the bearing lands that were in plane with the applied forces were obtained. The predicted elastic displacements are shown in Table 1, and the points at which the displacements occurred are shown in Fig. 10.

Given the predicted rotor deformation data in Table 1, the amount the stator would displace in response is now discussed. Conceptually, in order for the pocket pressures to change, the gap nearest the force must close by an amount sufficient to produce an equal opposed pressure force. If elastic deformation begins to open up the gap, the stator will continue to move to close it until the gap is sufficiently small again to produce the needed pressure to oppose the force. So to a first order at least, when the parts deform, the stator will move to close the gaps to what they would be had the parts been perfectly rigid. Applying this reasoning, we can compute the parasitic elastic deflection by computing how much the stator should move in order to close the gaps by the amount
Fig. 9. Diagram (a) and photo (b) of axial and tilt compliance test apparatus.
Table 1
FEA predicted elastic deflections at bearing land points on rotor (deflections in μm)

<table>
<thead>
<tr>
<th>Loading case</th>
<th>X(1)</th>
<th>Y(1)</th>
<th>X(1)</th>
<th>Y(1)</th>
<th>X(2)</th>
<th>Y(2)</th>
<th>X(2)</th>
<th>Y(2)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Zero load uniform pocket pressure (10 bar)</td>
<td>0.268</td>
<td>2.088</td>
<td>0.068</td>
<td>0.213</td>
<td>0.269</td>
<td>2.089</td>
<td>0.068</td>
<td>0.211</td>
</tr>
<tr>
<td>Radial load: 11,139 N</td>
<td>2.649</td>
<td>7.532</td>
<td>0.453</td>
<td>0.190</td>
<td>N/A</td>
<td>N/A</td>
<td>N/A</td>
<td>N/A</td>
</tr>
<tr>
<td>Axial load: 22,099 N</td>
<td>0.150</td>
<td>1.146</td>
<td>0.146</td>
<td>5.394</td>
<td>N/A</td>
<td>N/A</td>
<td>N/A</td>
<td>N/A</td>
</tr>
<tr>
<td>Off center load: 7356 N at r = 114.3 mm</td>
<td>1.485</td>
<td>1.099</td>
<td>0.012</td>
<td>0.875</td>
<td>2.265</td>
<td>6.694</td>
<td>0.257</td>
<td>0.052</td>
</tr>
</tbody>
</table>

Fig. 10. Points on rotor where FEA elastic deflections were obtained.

They elastically opened, or in other words, how much the stator needs to move to negate the effect of the rotor deformation. This deformation negating displacement, divided by the reaction force on the rotor produced by the FEA (which by the third law and by equilibrium must be equal to the applied force on the stator), provides a reasonable approximation of the parasitic elastic compliance present.

Using the data in Table 1, a deformation negating displacement of the stator was computed for each load case. For the zero external load case, its was computed the stator would move upward 1.2 μm to make the upper and lower bearing gaps equal, which they would be with no external load applied. This initial deflection is subtracted from the axial load case to obtain a change in displacement from idle. For the radial load case, on the loaded side of the stator, it was computed that the stator would move 4.35 μm toward the rotor to reduce the bearing gaps by the amount they were predicted to spread open. For the axial load case, it was computed that the stator would move 3.32 μm in order to negate the gap changes caused by elastic deformation. For the tilt load case, it was computed that at the left bearing lands on the stator, the stator would move down 2.59 μm, and at the right bearing lands the stator would move up 3.19 μm, in order to negate the gap changes caused by the elastic spreading of the rotor. Together these displacements produced a tilt angle of 6.7 arc secs, and a downward displacement at the location of the off center force (located 114.3 mm from the bearing axis) of 3.43 μm, due to elastic deformation.

Fig. 11. Radial compliance of prototype.
Dividing these values by the corresponding reaction forces shown in Table 1, the radial, axial, tilt, and off center elastic compliances are computed to be 0.00039 μm/N, 0.00015 μm/N, 0.00801 arc s/N m, and 0.00047 μm/N, respectively. These elastic compliances were used to compute a parasitic elastic displacement versus a given load type.

Figs. 11–14 show the measured bearing compliances at 10 and 40 bar supply pressures, for radial loading, axial loading, and off center loading. Also shown is the theoretical hydrostatic displacement before and after a parasitic elastic deflection is added. The plots show that with the elastic deflection added, there is reasonable agreement between the theoretical and experimental results. The results indicate that at 40 bar supply pressure, the elasticity of the parts contributes significantly to the total compliance, and therefore the part compliance must be taken into account if one wants to optimize
bearing stiffness given that level of supply pressure. The results also indicate that beyond a 40-bar supply pressure, for this structure there will be diminishing returns to increasing the supply pressure in an effort to reduce the compliance.

5. Flow rate measurements

The flow rate was measured at different supply pressures using a graduated cylinder and a stop watch. The oil used was Mobil Velocite No. 10 with a density of 865 kg/m³, and a kinematic viscosity of 36 cSt at 27.8°C. This viscosity was measured and is consistent with the viscosity versus temperature plot from the data sheet provided by Mobile. The bearing was connected to a variable displacement hydraulic pump with adjustable output pressure.

The theoretical flow rate through the bearing can be calculated using Eq. (15), or more expediently using the Poiseuille flow equations on the entire annular restrictor gap and bearing gap regions. The total resistance to flow through the bearing is the sum of the restrictor and bearing land resistances. Using Eqs. (2) and (3), one can write

\[ Q = \frac{12\pi}{2\eta} \left( \frac{L_1}{w_1 h_1} + \frac{L_2}{w_2 h_2} \right) \]

where \( Q \) is the flow, \( P_s \) the supply pressure, \( \eta \) the oil’s dynamic viscosity, \( w_1 \) the restrictor width (which is equal to the restrictor’s circumference), \( L_1 \) the restrictor length, \( h_1 \) the nominal restrictor gap, and \( w_2 \), \( L_2 \) and \( h_2 \) correspond similarly to the bearing land. The factor of 2 next to \( P_s \) accounts for the flow that occurs in both the upper and lower halves of the bearing. Substituting the values for the dynamic viscosity and the dimensions of the gaps yields a theoretical flow rate of 0.157 l/min at 13.79 bar. The measured flow rate at this pressure, however, was 0.488 l/min. If an error of 10% is assumed between the pressure gauge and the value for the viscosity, the gaps would have to be about 40% larger than measured in order to allow the measured flow rate. Since the restrictor gaps and the bearing land gaps were measured with much higher confidence than ±40%—the gaps were determined by measuring the parts with a CMM—internal damages to the parts must be responsible for the flow rate to be that much higher. At the time when the flow rate was measured, the prototype bearing had been taken apart many times for various purposes such as part inspection and dimensional measurements, and it is possible that local damage occurred that was not measured by the CMM, which touched only at six points on the restricting surfaces.

6. Error motion measurements

The error motion tests were done with the assistance of Moore Tool Company in Bridgeport, CT. The bearing was mounted on a three-axis universal coordinate measuring machine as shown in Fig. 15. The spacer blocks provide room underneath the bearing to accommodate the indicator. The indicator was connected to an analog readout system with a resolution of 0.025 μm. A gear motor was used to spin the rotor at 4 rpm with a compliant fork-shaped coupling to isolate possible error motions of the gear motor spindle from the bearing.

The measurements were made using a high precision ceramic spherical artifact with a sphericity of better than 0.03 μm. The artifact and the indicator were set-up in three different arrangements to find the radial, axial and tilt error.
Fig. 15. Diagram (a) and photo (b) of error motion set-up.

Fig. 16. Polar plot of the (a) radial error motion and (b) axial error motion (1 division = 0.127 μm, the radial and axial error motions are approximately 0.05 μm).

motions. The first measurement was done to determine the pure radial error motion with the artifact located in the radial and axial center of the bearing and an indicator pointing to the equator of the precision sphere (see position A in Fig. 15a). Note that since the artifact was so spherical and the bearing was not expected to be so good, a Donaldson Reversal Test [20] was not done when the measurements were carried out, and hence strictly speaking, runout measurements were made; however, again, because the artifact was so accurate, the error motion is at least as good as the runout, and hence, here we will use the term “error motion”.

The axial error motion was measured by positioning the indicator tip on the very bottom center point of the precision sphere (see position B in Fig. 15a). Radial error motions during the measurement of the axial error motion will have a cosine error effect and therefore have only a negligible influence on the axial error motion.
Table 2

<table>
<thead>
<tr>
<th>Error motion summary table</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Radial runout (μm)</td>
<td>&lt;0.05</td>
</tr>
<tr>
<td>Axial runout (μm)</td>
<td>&lt;0.05</td>
</tr>
<tr>
<td>Tilt error motion (arc s)</td>
<td>0.09</td>
</tr>
<tr>
<td>Part accuracy (μm)</td>
<td>2.5</td>
</tr>
<tr>
<td>Averaging factor</td>
<td>50</td>
</tr>
</tbody>
</table>

The test showed very good results as shown in the polar plot in Fig. 16a and b. The scale on both plots is 0.127 μm per division, which means that each one of the circles for axial and radial error motion lies within approximately 50 nm.

The tilt error motion was estimated by computing the maximum difference that occurred between the error motion measured at positions A and C, and then dividing by the distance between A and C. This yields a maximum tilt error of about 0.09 arc s. Table 2 summarizes the error motion measurement results.

7. Part accuracy versus error motion

In order to characterize performance it is necessary not only to determine the error motions but also to relate them to the accuracy of the active surfaces, since part accuracy is a key indication of the bearing cost. Fig. 17 shows representative how the parts (here the rotor) were measured. The inspection was done on the same measuring machine as used to determine the bearing error motions. In Fig. 18, the lines in the polar chart are numbered according to the indicator positions in Fig. 17. One division in the polar chart equals 0.254 μm, which yields a maximum range of 2.5 μm.

If one compares the polar plots of Figs. 16 and 18, a high degree of hydrostatic averaging can be observed. The low radial error motion of 50 nm was achieved with parts that are only good to within 2.5 μm. In past work, the use of a large number of pockets has been shown to create excellent averaging effects in externally pressurized bearings. Therefore, the numerous bearing pockets, which this new bearing design economically permits, is most likely responsible for this considerably enhanced hydrostatic averaging effect, which is on the order of 50 (part accuracy divided by error motion). This averaging factor represents quasi static error motion behavior in accordance with the characteristics of the mechanical contact probe’s low pass filter. This is in accordance with hydrostatic and aerostatic bearings’ well known characteristic of being nearly free of high frequency error motion.

Tests or calculations to find the temperature or speed limits of the bearing were not carried out, because the intended application was for low speed rotary tables, and hence the seal was not designed for high speeds. Based on the work of Kotilainen and Slocum [18], the authors of this article see no fundamental reason why the design presented here could not be stable at high speeds. For high speeds, a smaller outer diameter and a larger height would be chosen to minimize the shear power while maintaining tilt stiffness. Investigating stability issues, and optimizing the geometry for high speeds are both interesting areas of future work for this bearing.

8. Manufacturing considerations

Only after several design iterations was a relatively easy to manufacture shape discovered. Figs. 19 and 20 show representatively the final design compared with the initially envisioned profile shapes. The prototype bearing was manufactured by Elk Rapids Engineering, a builder of cutter grinders. The manufacturing challenge arises from the requirement of having a 20 μm nominal no-load bearing gap between the stator and the rotor at both the bearing and restricting surfaces. In order to prevent the part errors from influencing bearing performance, it was assumed that the accuracy of the bearing parts should be on order of 2 μm, or one...
order of magnitude less than the nominal bearing gap. From a machinist’s point of view, the design changes in Figs. 19 and 20 represent major improvements. In Fig. 19, the stator is initially cut out of one piece with a radius at the neck of the hammerhead shape. This requires a profiled grinding wheel which causes many problems in terms of placing the radius within 2 μm and matching it to its counterpart on the rotor.

In the new design for the stator, the “hammer head” shape is added as a separate ring—the restrictor ring as shown in Fig. 1. Thus, a radius is avoided and a profiled grinding wheel is no longer needed. All the precision surfaces are straight and can be ground with straight grinding wheels. Additionally, the upper and lower restrictor surfaces will be very precisely concentric because they are part of a single ring surface.

Similar considerations apply to the rotor. The problem with the initial design is that some precision surfaces are hidden in a groove and therefore they are difficult to reach with grinding or measuring equipment. Fig. 20 shows the improved final design next to the initial design. Straight grinding wheels can be used here as well and the precision surfaces are easily accessible. Furthermore, in order to match the angular surfaces of the stator and the rotor, the parts are ground in the same set-up. Also, the bearing gap can be accurately set with minimal machining by grinding down or adding a shim to the spacer.

A minor potential drawback to this design is the sharp acute knife edges, which need to be sharp to block a potential short circuit between bearing pockets. A short circuit channel would not allow pocket pressures to be different from one another, which would virtually eliminate the radial and tilt stiffness of the bearing. Therefore, the knife edges require careful attention during the manufacturing and assembly process. However, once the bearing is assembled, the knife edges are completely protected from damage and hence pose no problems during operation.

9. Conclusions
A novel hydrostatic bearing has been presented which is potentially useful for applications that require very high rotational precision and stiffness in a low profile package. The relatively high averaging factor allows bearing part tolerances on the order of 50 times more generous than the intended bearing accuracy. This makes it possible to attain high accuracy of motion with relatively low cost parts. Additional cost reduction is achieved by the fact that the bearing...
does not need capillaries or orifices to be assembled or tuned for operation, nor does it require special grooves or slots that must have a precise depth. Further work may investigate casting the pockets into the rotors to lower the manufacturing cost.

A disadvantage that the presented bearing shares with every other hydrostatic bearing is the need for an external hydraulic power unit and return plumbing, which adds heat to the system and also adds extra cost. However, many machining centers and other machine tools are already equipped with a hydraulic unit to actuate work piece clamps or tool changers. Since the oil consumption of the bearing is only about 0.5 l/min at an operating pressure of 40 bar, only about 35 W of power is dissipated. For applications that cannot tolerate even the relatively small heat generation, a chiller can be added to the hydraulic unit to remove heat from the oil. Future work will focus on the integration of this design into existing or new machine tool structures and to investigate the benefits of high accuracy, stiffness, and damping during actual machining operations.

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References