Magnetic fluid seal for linear motion system with gravity compensator

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Abstract

Demands of ultra-precision machining systems have recently increased for machining three dimensional nano-geometries. Such systems require not only horizontal positioning systems but also vertical positioning systems with a nanometer positioning accuracy. However, the gravity load of moving mass negatively affects the positioning accuracy especially in a vertical positioning system. Therefore, it is necessary to compensate the influence of the gravity load for vertical ultra-precision positioning. Although the gravity compensation method using a non-contact vacuum cylinder has been proposed, it is difficult to improve the sealing performance of the non-contact seal. Thus, this study proposes the magnetic fluid seal for linear motion system with a gravity compensator. In general, magnetic fluid seals have been applied to rotary motion systems, however there are few studies on linear motion systems so far. When a magnetic fluid seal is applied to linear motion systems, the magnetic fluid leaks out of the seal clearance due to the movement, and it cannot keep the performance as a seal. In this study, it is possible to prevent leaking the magnetic fluid from the seal clearance and also to maintain the performance of the seal by holding a magnetic fluid by the two pole pieces shaped tapered tip and putting it near the piston made of non-magnetic material and coated with oil repellent layer. The proposed magnetic fluid seal performance has been evaluated. As a result, the result confirmed that the proposed magnetic fluid seal is useful as a linear motion seal, and the oil repellent layer on the piston surface can largely improve the sealing performance.

Keywords: Magnetic fluid seal, Gravity compensator, Linear motion system, Vacuum cylinder, Oleophobic coating.

1. Introduction

Demands for precise positioning have recently increased in the various industrial sectors. Generally, three dimensional ultra-precision positioning has been required for machining complicated nano-geometries such as aspheric micro lenses and optical reflection devices. In order to meet such demands, it is necessary to develop not only horizontal positioning systems, but also vertical positioning systems with a nanometer positioning accuracy. In such vertical positioning systems, however, a gravity load of moving mass negatively affects the positioning accuracy because a moving and a gravity directions are same. For this reason, it is necessary to add a mechanism which can support the gravity load of a moving part and minimize the positioning error caused by the gravity load. In order to compensate the gravity load, counterweights and counterbalances have been widely used so far \cite{1,2}. Those typical gravity compensators have positioning error factors such as propagation of vibrations and frictions due to contact mechanism. Therefore, ultra-precision positioning systems are constructed with totally non-contact elements. Besides, a new non-contact gravity compensator has been required.

A non-contact gravity compensator using a vacuum cylinder has been developed \cite{3}, as shown in Fig. 1. It can flexibly change output because it supports the gravity load with the attraction force resulting from vacuum. However, it is difficult to improve the sealing problems. In particular, pumping pulsation and output fluctuation because of the narrow seal clearance as Fig. 2. This study, therefore, newly proposes the magnetic fluid seal for the linear motion mechanism, then, develops a vacuum cylinder using it as a new non-contact gravity compensator.
2. Structural design of magnetic fluid seal for linear motion mechanism

In general, magnetic fluid seals have been applied to rotary motion systems. Fig. 3 shows a typical magnetic fluid seal. It is composed of a permanent magnet, pole pieces, a magnetic fluid and a rotary shaft, and all of components are model of magnetic material. When it is applied to linear motion systems, there are the following problems:

- Magnetic fluid can leak out of the seal clearance in accordance with moving of a shaft.
- Eddy current which causes resistive losses is generated in accordance with moving of a shaft.
- Magnetic attraction force can tilt a moving shaft due to unbalanced magnetic field.

Therefore, a magnetic fluid seal for linear motion system is required to meet the following requirements:

- The material of the piston should be non-magnetic material in order to eliminate magnetic attraction force between the pole piece and the piston. Furthermore, it can reduce magnetic flux passing through the piston and suppress eddy current generated in accordance with moving of the piston.
- Magnetic field should be formed to prevent the leakage of the magnetic fluid. Therefore, two pole pieces shaped tapered tip. These pole pieces act as magnetic lenses and can apply a string magnetic field to the magnetic fluid. In addition, it forms magnetic gradient toward the center of the magnetic fluid.
- The piston should be coated with oil repellent layer. It can reduce the share force applied to the magnetic fluid, and prevent the leakage of the magnetic fluid.

Figure 4 shows that the structure of the proposed magnetic fluid seal meets those requirements. A gap between a magnet and pole pieces is filled with a resin to prevent the magnetic fluid’s moving into the gap.

3. Theoretical analysis of the proposed magnetic fluid seal

When the piston is almost kept stationary and the magnetic fluid moves slowly, the dissipation energy due to viscosity of the magnetic fluid is negligibly small. As pointed out by Rosensweig [4], therefore, the Bernoulli equation can be extended to Eq.(1) in order to describe magnetic fluid flow:

$$\frac{1}{2} \rho v^2 + \rho gh + p - \int M dH = \text{const.}$$

where, \( \rho \) is the density of the magnetic fluid, \( v \) is the flow velocity, \( g \) is the gravity acceleration, \( h \) is the height from the reference point, \( p \) is the pressure, \( M \) is the magnetization of the magnetic fluid and \( H \) is the magnetic field intensity, respectively. From Eq.(1), if the piston moves in the axial direction, changes in kinetic and potential energies are also negligibly small. Thus, it can be derived the approximation maximum seal pressure of the magnetic fluid seal:

$$\Delta P = N M_s (H_{\text{max}} - H_{\text{min}})$$

where, \( N \) is the number of seal stages, \( M_s \) is the saturation magnetization and \( H_{\text{max}} \) and \( H_{\text{min}} \) are the maximum and minimum value of the magnetic field intensity in the magnetic fluid, respectively.

From the above equations, this study evaluated the maximum seal pressure measured from the actually developed magnetic fluid seal, and compared it with derived value from Eq.(2) to confirm whether the obtained theoretical values are appropriate. In addition, the piston of the proposed magnetic fluid seal was coated with oil repellency layer, but the effects of piston coating on the seal performance have not been studied so far.
4. Evaluation of proposed magnetic fluid seal performance

The performance of the proposed magnetic fluid seal was experimentally evaluated. Fig. 5 shows the structure of the evaluation system and Fig. 6 shows the appearance. One or two magnetic fluid seals could be mounted on the vacuum cylinder and were fixed on the base plate. The vacuum pump was connected to the cylinder through the vacuum regulator and it could arbitrarily regulate vacuum pressure in the cylinder. The pressure sensor was also connected to the cylinder for measuring the differential pressure between the cylinder and the atmosphere. The ball screw connected the stepper motor could position the piston in the axial direction. All of the components are made of non-magnetic material except magnets and pole pieces in order to eliminate disturbing the magnetic flux around the magnetic fluid seals. The specifications of the evaluation system are shown in Table 1.

First, the relationship between the maximum seal pressure and oil repellency was investigated. Three pistons made of duralumin were used for comparison. The first one was coated with ceramic, the second one was coated with fluororesin and the third one was not coated. Those pistons have different oil repellence respectively and it was evaluated by measuring the contact angle. As shown in Fig. 7, the duralumin surface hardly has oil repellence, however ceramics and fluororesin coating can vastly improve oil repellence. Moreover, those pistons were pulled out from the magnetic fluid seal at a speed of 1mm/s in order to observe the leakage of magnetic fluid. Fig. 8 shows that oil repellent layer can prevent the leakage in accordance with moving of a piston. Then, changes in the maximum seal pressure due to piston coating was measured using the evaluation device. Only one magnetic fluid seal was mounted on the cylinder, the piston was kept stationary and the seal clearance was 100μm. The maximum seal pressure was measured with following procedure, 1) filling magnetic fluid between the seal and the piston to seal the cylinder, 2) reducing the cylinder pressure gradually and measuring the differential pressure between atmosphere and the regulator and between atmosphere and the cylinder respectively. 3) When the magnetic fluid burst, the pressure in the cylinder rapidly increases because the air flows into the cylinder. 4) The differential pressure between atmosphere and the cylinder at this moment is determined as a maximum seal pressure. Fig. 9 shows the result of measurement. It indicates that the oil repellent layer on the piston surface can improve the sealing performance as its oil repellence becomes higher. It can be conjectured as a cause that interfacial tension of the piston is reduced by oil repellent layer, the shape of the magnetic fluid changes and $\mu$ decreases, as shown in Fig. 10. This result indicates a reduction of the maximum seal pressure predicted by Eq.(2).

In order to confirm the applicability of Eq.(2) to the proposed magnetic fluid seal, changes in the maximum seal pressure were measured when three parameters $(N, M, H$ in Eq.(2)) are changed respectively. The relationship between the maximum seal pressure and the seal clearance was investigated. In general, the seal clearance becomes larger, the maximum seal pressure reduces. It is speculated that the magnetic field intensity applied to the magnetic fluid $H$ decreases as the seal clearance becomes large [5]. Therefore, the maximum seal pressure can be predicted by analyzing the magnetic field distribution when the seal clearance is changed. Fig. 11 shows the model of magnetic field analysis which drafted actual

<table>
<thead>
<tr>
<th>Overall system</th>
<th>Stroke</th>
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<tr>
<td>Piston diameter</td>
<td>40mm</td>
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<tr>
<td>Seal clearance</td>
<td>100 to 200μm</td>
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<th>Pressure sensor</th>
<th>Measuring pressure</th>
</tr>
</thead>
<tbody>
<tr>
<td>Resolution</td>
<td>0.1kPa</td>
</tr>
<tr>
<td>Repeatability</td>
<td>0.2kPa</td>
</tr>
</tbody>
</table>

![Fig. 5. Structure of the evaluation system.](image)

![Fig. 6. Appearance of the evaluation system.](image)

![Fig. 7. Contact angle on each surface.](image)

![Fig. 8. Magnetic fluid adhering to the piston surface.](image)
dimensions in the cylindrical coordinate system, and the magnetic field intensity was analyzed with Finite Element Analysis. Fig. 12 shows a result of the magnetic field intensity analysis when the seal clearance is 200 μm. As shown in this figure, it is assumed that the x-axis is set in the axial direction, the magnetic fluid is filled within a tip of pole pieces and the dashed line A-A indicates where the magnetic field intensity analyzed to obtain $H_{\text{max}}$ and $H_{\text{min}}$. Fig. 13 shows the distribution of the magnetic field intensity when the seal clearance changes from 100 μm to 200 μm. As shown in Fig. 13, there are no changes in $H_{\text{max}}$, however $H_{\text{min}}$ decreases as the seal clearance becomes large. Changes in the maximum seal pressure due to the seal clearance was measured in a similar way to Fig. 9, however the piston coated with fluororesin was used, and compared with the analyzed value as shown in Fig. 14. From these results, experimental results were in good agreement with the analyzed values. Then, this study also investigated the influence of the saturation magnetization of magnetic fluid and the number of the seal stages on the maximum seal pressure. This measurement used three types of magnetic fluid A, B and C, as shown in Table 2. Fig. 15 shows the measurement results of changes in the maximum seal pressure due to the saturation magnetization of magnetic fluid and Fig. 16 shows the measurement results of changes in the maximum seal pressure due to the number of seal stages. As those experimental results show, the saturation magnetization of magnetic and the number of seal stages affect the maximum seal pressure proportionally as indicated by Eq.(2). Therefore, the applicability of Eq.(2) to the proposed magnetic fluid seal was confirmed.
Table 2. Specifications of the magnetic fluids

<table>
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<tr>
<th>Magnetic fluid</th>
<th>Saturation magnetization [mT]</th>
<th>Viscosity (20°C) [mPa·s]</th>
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<tr>
<td>A</td>
<td>51.4</td>
<td>850</td>
</tr>
<tr>
<td>B</td>
<td>45.0</td>
<td>410</td>
</tr>
<tr>
<td>C</td>
<td>35.6</td>
<td>360</td>
</tr>
</tbody>
</table>

Fig. 15. Changes in the maximum seal pressure due to the saturation magnetization of magnetic fluid.

Fig. 16. Changes in the maximum seal pressure due to the number of seal stages.

5. Evaluation of proposed non-contact vacuum cylinder output

The output of the proposed non-contact vacuum cylinder was experimentally evaluated in order to confirm that it is capable of supporting mass against the gravity load. First, the attraction force from the cylinder was measured by force sensor put on the piston. Fig. 17 shows the relationship between pressure in the cylinder $P$ and attraction force of the cylinder $F$. Theoretical value was obtained:

$$F = P \frac{\pi d^2}{4}$$

where, $d$ is the piston diameter. This result shows that the cylinder enables to generate linear attraction force. Thus, it indicates that the attraction force is easily estimated with the cylinder design and can be flexibly changed depending on the changes in the supporting mass. Second, the output fluctuation from the cylinder was evaluated. The piston was pushed or pulled in 10 mm/s and the pressure in the cylinder was measured concurrently. As Fig. 18 shown, the output fluctuation was less than 10 N and the cylinder can provide a stable output. Difference in the rate of variation of the output fluctuation rate when the piston was pushed and pulled, however, is caused by the response characteristic of the vacuum regulator. From these results, one cylinder can support the mass up to 14.8 kg and is suitable for gravity compensator because of its stability and linearity of the output.

Fig. 17. Output from proposed non-contact vacuum cylinder.

Fig. 18. Pressure fluctuation when the piston moves in 10 mm/s.

6. Conclusions

This study presented a newly magnetic fluid seal for linear motion system with the gravity compensator developed for achieving vertical ultra-precision positioning. As a result, the following conclusions could be obtained.

- This study formulated the design concepts of a magnetic fluid seal for linear motion system.
- The maximum seal pressure of the proposed magnetic fluid seal was evaluated and showed good agreement with analyzed values.
- It is revealed that the oil repellent layer on the piston surface could be largely improve the sealing performance.

In the future, it is necessary to be mounted the newly proposed non-contact gravity compensator on a vertical positioning system and perform a measurement experiment of positioning accuracy in order to confirm that it enables to cancel the negative effect of the gravity load and improve a positioning accuracy in a vertical direction.
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