PNEUMATIC SERVO BEARING ACTUATOR FOR ULTRAPRECISE POSITIONING – PART 1: PRINCIPLE AND BASIC PERFORMANCE

Tomoko Hirayama1, Takakazu Kitagawa1, Masato Kadotani1, Hiroki Danjyo1, Takashi Matsuoka1, Katsumi Sasaki2 and Hiroshi Yabe3

1Department of Mechanical Engineering, Doshisha University
1-3 Miyakodani, Tatara, Kyotanabe, Kyoto, Japan
2Pneumatic Servo Controls Ltd.
2266-22 Anagahora, Shimo-Shidami, Moriyama-ku, Nagoya, Aichi, Japan
3Department of Mechanical Engineering, Kyoto University (former affiliation)
Yoshida-honmachi, Sakyo-ku, Kyoto, Japan

INTRODUCTION

Advances in various areas of precision mechanics, such as semi-conductor exposure apparatus, MEMS/NEMS, nano-imprint lithography devices, and other nano-technology applications, have required ultraprecise positioning techniques with nanometer-scale accuracy to be developed. To meet such specific needs, many different kinds of actuators capable of ultraprecise motions have been developed. However, the needs for ultraprecise positioning system continue to expand – for example, for higher load and for circumstances free from electromagnetic fields. Therefore, new actuators with higher accuracy and better reliability are strongly required.

In this paper, we have developed a ‘Pneumatic Servo Bearing Actuator (PSBA)’ as a new actuator that uses pneumatic servo technology for ultraprecise positioning. The actuator mainly consists of an aerostatic thrust bearing and a pneumatic servo valve. In this first report, we describe its operation principle, prototype construction, and basic performance of the prototype PSBA under open- and closed-loop controls.

PRINCIPLE OF OPERATION

A schematic diagram of the PSBA is shown in Figure 1. The actuated part is the output spool, as shown in the figure. Its position is set by balancing two forces; force \( f_s \) is generated in the aerostatic thrust bearing and acts to the left, and force \( f_b \) \((=a_h \cdot p_b)\) is a constant preload force and acts to the right. The inlet pressure of air, \( p_c \), from the compressor to the bearing clearance is controlled with a servo valve. After the air works as the fluid of the aerostatic bearing in the bearing clearance, it passes out into the atmosphere. Force \( f_s \) is the integrated value of the pressure distribution in the bearing clearance. Because aerostatic bearings have innately stiffness, a change in inlet pressure alters the bearing clearance, which directly corresponds to a change in the displacement of the actuated spool. The spool is capable of frictionless movement because it is supported by an aerostatic journal bearing.

The flow diagram of physical quantities in the PSBA is expressed using a four-terminal network, as shown in Figure 2. The figure illustrates the relationship between input and output, which can be expressed as a matrix:

\[
\begin{bmatrix}
q_e
f_s
\end{bmatrix} =
\begin{bmatrix}
Y_{11} & Y_{12} \\
Y_{21} & Y_{22}
\end{bmatrix} \begin{bmatrix}
p_c \\
h
\end{bmatrix} + \begin{bmatrix}
0 \\
-f_b
\end{bmatrix}
\] (1)

FIGURE 1. Schematic diagram of pneumatic servo bearing actuator (PSBA).

FIGURE 2. Four-terminal network flow diagram of physical quantities in PSBA.
In the PSBA, an aerostatic thrust bearing with groove compensation was used. General design of the bearing is illustrated in Figure 3 (left). According to some previous researches [1], the performance of a bearing with a groove compensation can be approximately estimated using a simple shallow-recessed bearing model with equivalent recess depth (Figure 3, right). Applying the lumped parameter model proposed by Mori [2] enables the admittances in Figure 2, from $Y_{11}$ to $Y_{22}$, to be expressed as follows:

\[
Y_{11} = \frac{\Delta q_{in}}{\Delta p_{c}} = \frac{\beta \eta}{\alpha + \beta} \frac{1 + \tau_{4}s}{1 + \tau_{2}s} \tag{2}
\]

\[
Y_{12} = \frac{\Delta q_{in}}{\Delta h} = \frac{\alpha}{\alpha + \beta} \frac{\theta - \psi}{1 + \tau_{2}s} + \psi \tag{3}
\]

\[
Y_{21} = \frac{\Delta f_{s}}{\Delta p_{c}} = A_{e} \cdot \frac{\eta}{\alpha + \beta} \frac{1}{1 + \tau_{2}s} + A_{c} = A_{p} \cdot \frac{1 + \tau_{3}s}{1 + \tau_{2}s} + A_{c} \tag{4}
\]

\[
Y_{22} = -\frac{\Delta f_{s}}{\Delta h} = A_{e} \cdot \frac{\theta - \psi}{\alpha + \beta} \frac{1 + \tau_{3}s}{1 + \tau_{2}s} \tag{5}
\]

where the linearized parameters are:

\[
\alpha = (\frac{\partial q_{in}}{\partial p_{c}})_{0}, \ \beta = 1 - (\frac{\partial q_{in}}{\partial h})_{0}, \ \psi = (\frac{\partial q_{in}}{\partial p_{c}})_{0}, \ \theta = (\frac{\partial q_{out}}{\partial h})_{0}, \ \chi = (\frac{\partial q_{in}}{\partial p_{c}})_{0}, \ \varphi = (\frac{\partial q_{out}}{\partial h})_{0} \tag{6}
\]

$q_{in}$: inlet mass rate of flow to bearing
$q_{out}$: outlet mass rate of flow from bearing
$q$: mass rate of flow in bearing clearance
$p$: pressure at recess edge in bearing clearance

0: operating point

and

\[
A_{e} = \text{effective bearing area} = \frac{\partial f_{s}}{\partial p_{c}, \varphi} \tag{7}
\]

\[
A_{c} = \text{equivalent bearing area} = \frac{\partial f_{s}}{\partial p_{c}, \varphi} \tag{8}
\]

Therefore, the bearing stiffness and damping coefficient are given by:

\[
k = A_{e} \cdot \frac{\theta - \psi}{\alpha + \beta} \tag{7}
\]

\[
b = k (\tau_{3} - \tau_{2}) \tag{8}
\]

The damping coefficient estimated using Eq. (8) is innately positive, so the operation of the bearing should be stable, free from the inherent pneumatic self-excited oscillation.

The dimensions of the prototype PSBA are listed in Table 1. The servo valve is a nozzle-flapper type valve with an axial torque motor (Pneumatic Servo Controls Ltd.). The displacement of the spool was measured with a high-resolution capacitance-type non-contact displacement sensor (Microsense, ADE Ltd.). Its minimum resolution is 0.3 nm.

**TABLE 1. Dimensions of prototype PSBA.**

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Outer radius of bearing</td>
<td>13.5 mm</td>
</tr>
<tr>
<td>Radius of inlet port</td>
<td>2.5 mm</td>
</tr>
<tr>
<td>Number of grooves</td>
<td>16</td>
</tr>
<tr>
<td>Average depth of grooves</td>
<td>19 µm</td>
</tr>
<tr>
<td>Spool mass</td>
<td>260 g</td>
</tr>
<tr>
<td>Volume $V$</td>
<td>1574 mm$^3$</td>
</tr>
</tbody>
</table>

**BASIC PERFORMANCES OF PSBA UNDER OPEN-LOOP CONTROLS**

The repetitiveness of the actuator’s positioning was tested by continuously changing the signal input to the servo valve. The pressure of the air supplied to the servo valve, $p_{c}$, was a constant 0.3 MPa and the zero position (= base clearance) was 7 µm. As shown by the typical results in Figure 4, the repetitiveness of the prototype was less than 10 nm, meaning that the actuator exhibited almost no hysteresis even under open-loop control. The level of fluctuation of the actuated spool depended on the bearing clearance and inlet air pressure. When the bearing clearance was less than 20 µm and the inlet air pressure $p_{c}$ was less than 0.1 MPa, the
fluctuation was less than 1 nm under any condition. The step response of the prototype PSBA under open-loop control with a 5 nm-high step is shown in Figure 5. The zero position was set to 7 µm. Even though there were some small fluctuations (including electric noise), the actuator performed sharp switching. This means that the developed actuator has a high potential for ultraprecise positioning with nanometer-scale accuracy.

The controllability of the axial force of PSBA with this simple position feedback was examined. The results are shown in Figure 7. The black circles show the axial force of the PSBA when the servo valve was kept full-open under constant air supply pressure into the valve (p = 0.3 MPa), and the white circles in the figure show the axial force controlled by the feedback for maximum stiffness. The actuator’s stiffness was increased about 12-fold by applying the feedback control. Reducing the loop gain reduced the stiffness. This means that the axial force can be perfectly controlled for the area under the curve (dotted area), so the stiffness appropriate for each device can be obtained.

Bode diagrams for open- and closed-loop control of the PSBA are shown in Figure 8. The black circles show the gain and phase for the PSBA under open-loop control, while the white ones show them for under closed-loop control where the loop gain was set for maximum stiffness.

**FIGURE 4. Repetition accuracy of prototype PSBA under open-loop control.**

**FIGURE 5. 5 nm-high step response of prototype PSBA under open-loop control.**

**IMPROVED PERFORMANCES OF PBSA UNDER CLOSED-LOOP CONTROLS**

The performance of the PBSA was improved by applying closed-loop control to the system. Simple position feedback was added, as illustrated by the block diagram in Figure 6. The only control parameter was loop gain, 

\[ K_p \]

**FIGURE 6. Block diagram of PSBA system with position feedback.**

**FIGURE 7. Axial force of the PSBA with a closed-loop control for maximum stiffness.**

**FIGURE 8. Bode diagrams of PSBA under open- and closed-loop control.**
These diagrams demonstrate that the PSBA is an inherently stable system. The closed-loop control expanded its bandwidth from 10 Hz to 115 Hz. Note that the break frequency, at about 10 Hz, in the gain diagram under the non-controlled condition is determined by the volume $V$ of air arranged prior to the inlet port to the aerostatic bearing (see Figure 1).

The step responses of the PSBA under the non-controlled and the position feedback control are shown in Figure 9. The zero position was 5 $\mu$m, and the step height was 40 nm. The controlled PSBA performed sharper switching without any drift or hysteresis than the non-controlled one. Position feedback is thus effective for achieving stable ultraprecise motion of the PSBA. In the experiments with feedback control, the gain was set for maximum stiffness, resulting in a damping ratio of about 0.2. By selecting the appropriate loop gain, we can obtain the optimal damping ratio for each device.

The step responses of the PSBA with a 10 nm-, 5 nm- and 2 nm-high steps without noise filtering are shown in Figure 10 for a sampling frequency of 1 kHz and with a loop gain for maximum stiffness. The zero position was 5 $\mu$m. The PSBA performed sharp switching and the minimum positioning resolution of the prototype PSBA was about 2 nm. Application of another feedback control such as velocity feedback, acceleration feedback, or disturbance observer, or use of another displacement sensor with higher resolution would result in even more precise motion of the PSBA. In addition, a new approach to ultraprecise positioning that uses PSBAs with multiple bearing pads in a tandem arrangement has been examined; the results are described in the following Part 2 report.

**CONCLUSION**

A pneumatic servo bearing actuator (PSBA) has been developed as a potential new actuator to meet the needs for ultraprecise positioning brought about by advances in ultraprecision mechanics. The specifications of the prototype PSBA are enumerated in Table 2. Even under open-loop control, the PSBA had good repetition accuracy and a sharp step response. Applying simple closed-loop control increased the stiffness 12-fold and sharpened the step response without any drift or hysteresis. The PBSA will thus succeed in breaking new technology for the development of ultraprecision positioning system in the immediate future.

**REFERENCES**
