INTRODUCTION

Pneumatic servo systems are used in many fields, such as pneumatic robot systems, aspherical glass molding machines, and vibration isolation systems [1]. The use of air power has a number of advantages, including compressibility, high power ratio, and low heat generation. Also air is non-magnetic and a clean energy source.

The performance of the pneumatic servo systems improved greatly as the pneumatic servo valves became commercially available in the late 1980s. At present, one of the best pneumatic servo valves has a dynamic response of 100 Hz and a spool position accuracy of 2% to the full stroke. By using these pneumatic servo valves, the precise position control by the pneumatic servo system has entered a practical stage.

There are many researches to improve the pneumatic position control system. Despite these efforts, the position controllability of the pneumatic system is inferior to the electric system. Therefore, high precise pneumatic servo positioning systems are often used electric actuators, like a piezoelectric actuator, in fine stroke [2]. However, these hybrid systems using electric actuators eliminate the advantages of pneumatic systems, which are non magnetic field generation and low heat generation.

The authors have developed a pneumatic servo table system for precise position control [3]. The system achieved position error within 0.2 µm using an air bearing to reduce friction force. We also made clear that the position controllability is dominated by the dynamic characteristic of the servo valve. Therefore, we have developed a novel pneumatic servo valve [4]. The natural frequency and the damping ratio of the servo valve could be set freely by gain tuning.

In this paper, the effect of the dynamic characteristic of the servo valve on the controllability of the pneumatic servo table system is investigated experimentally. Step response tests, frequency response tests and trajectory tracking tests of the pneumatic servo table are implemented in various natural frequencies of the servo valve.

PNEUMATIC SERVO TABLE SYSTEM

The main components of the pneumatic servo table system are a pneumatic actuator and two pneumatic servo valves. The system is shown in Fig. 1. The experimental setup including a schematic view of the pneumatic actuator is shown in Fig. 2.

Pneumatic actuator

The pneumatic actuator consists of a slider, which is 1-DOF moving part, and a fixed guide. The slider is mounted with externally pressurized air bearings. The air bearings act through holes in the surface of the guide, attaining a non-contact drive between the slider and the guide during movement. Due to the air bearing, the slider can move without stick-slip effect and can accelerate smoothly. The slider is driven by the pressure difference at both pressurized chambers. The full stroke of the slider is ±70 mm. The displacement of the slider is measured by a linear transducer of resolution 0.05 µm. Values of the displacement are passed to a PC through a counter. The PC acts as a controller and sends the control signals to the servo valves.

The open loop transfer function from the control input $u$ to the slider position $x$ is given by Eq. (1) [3]:
Servo valves are connected to a pneumatic actuator via a guide and slider. A position sensor is used to measure the position of the actuator. The pneumatic actuator is connected to a pressurized chamber and an air bering. A current sensor measures the current. A DA converter and a DIO interface are used to control the system.

Figure 1: Pneumatic servo table system.

Figure 2: Experimental setup of pneumatic servo table system including a schematic view of the pneumatic actuator.

The open loop transfer function from the control input $u$ to the spool position $x_{sp}$ is modeled as a second order delay system. Using the state feedback, the closed loop transfer function is as follows:

$$G_{sp} = \frac{x_{sp}(s)}{u(s)} = \frac{\omega_{sp}^2}{s^2 + 2\zeta\omega_{sp}s + \omega_{sp}^2}$$

(2)

The dynamic characteristic of the HPPSV can be determined to a desired value by setting the natural frequency $\omega_{sp}$ and the damping ratio $\zeta$. The sonic conductance $C$ of the HPPSV is proportional to the spool position $x_{sp}$.

Setting several natural frequencies $\omega_{sp}$, the dynamic characteristics of the HPPSV are changed. The damping ratio $\zeta$ is fixed at 0.7. Figure 3 shows the results of the frequency response test of the HPPSV with 10% of the full-stroke. From Fig. 3, the dynamic characteristic of the HPPSV agree well with the theoretical curve at any natural frequency. Furthermore, the band width of the HPPSV is up to 300 Hz.

POSITION CONTROL

To control the pneumatic servo table system, a PDD controller is used. The position, the velocity and the acceleration are used to calculate the controller input:

$$u = K_p(x - x_{ref}) - K_v \dot{x} - K_a \ddot{x}$$

(3)
Because the velocity and the acceleration are not measurable, these values are estimated by a Kalman filter. Figure 4 shows the block diagram of the positioning system.

By using a normalized Laplace operator $s_0$ and dimensionless parameter $\alpha$ and $\beta$,

$$G_i(s) = \frac{x(s)}{x_{ref}(s)} = \frac{1}{s_0^3 + \alpha s_0^2 + \beta s_0 + 1} \quad (4)$$

The velocity gain $K_v$ and the acceleration gain $K_a$ can be expressed as functions of $\alpha$, $\beta$ and the proportional gain $K_p$.

**EXPERIMENTAL RESULTS**

**Step response experiment**

To improve the dynamic characteristic and the position controllability, a larger value of the proportional gain $K_p$ is required. In the actual system, the proportional gain is limited by the nonlinearity of the system.

Table 1 shows the proportional gain of the system at the limit of stability condition with several natural frequencies of the HPPSV. The natural frequency of the HPPSV is set as 100 Hz, 200 Hz and 300 Hz. The step amplitude of the system is set as 0.1 mm and 0.5 mm. From Table 1, the proportional gain at the limit stability condition becomes larger for the higher natural frequency of the servo HPPSV, clearly.

**Table 1: Comparison of stability limit proportional gain chaining valve natural frequency.**

<table>
<thead>
<tr>
<th>Stroke</th>
<th>$\omega_{sp}$ [rad/s × 2π]</th>
<th>$\omega_{sp}$ = 100 × 2π [rad/sec]</th>
<th>$\omega_{sp}$ = 300 × 2π [rad/sec]</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.1 mm</td>
<td>100</td>
<td>0.17</td>
<td>0.49</td>
</tr>
<tr>
<td>0.5 mm</td>
<td>200</td>
<td>0.37</td>
<td>0.23</td>
</tr>
<tr>
<td></td>
<td>300</td>
<td>0.49</td>
<td>0.23</td>
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**EXPERIMENTAL RESULTS**

**Figure 5: Experimental results of transient response.**

**Figure 6: Experimental results of steady state error.**

Step response experiments are implemented with no load at the amplitude of 0.1 mm. The proportional gain is set to the 90% of the limit of stability. Figure 5 shows the transient responses and Fig. 6 shows the steady state error. From Fig. 5, the rising time is improved by using a higher natural frequency of the HPPSV. Furthermore, it is clear from Fig. 6, that the steady state error is improved from $\pm 0.25 \mu$m to $\pm 0.05 \mu$m by changing the natural frequency of the HPPSV from 100 Hz to 300 Hz.

**Frequency response experiment**

Figure 6 shows the Bode diagram at the amplitude of 0.1 mm. The dashed lines in Fig. 6 show the numerical calculation results of the system based on Eq. (4). The experimental results agree well with the numerical calculation one at both gain response and phase response.
Using a higher natural frequency, the bandwidth becomes wider. It means the velocity tracking ability is improved by using a higher natural frequency servo valve.

Trajectory tracking experiment
The maximum velocity of the system is restricted by the characteristic of the position encoder (300 mm/s in this system). The maximum jerk, or time variation of acceleration, of the system is restricted by the maximum sonic conductance of the HPPSV (1300 m/s² in this system).

From these restrictions, the selection of the reference trajectory is important to realize precise positioning. Hence, a reference trajectory of the 5.0 mm amplitude sinusoidal curve is designed with feedforward compensation. The frequency of the reference trajectory is set to 0.1 Hz. This reference trajectory is not affected by the restrictions mentioned above.

Figure 7 shows the experimental results of the tracking control test of the system. From Fig. 7, the tracking error is improved from ±3 μm to ±0.5 μm changing the natural frequency of the HPPSV from 100 Hz to 300 Hz.

CONCLUSION
Using a novel pneumatic actuator and novel pneumatic servo valves, a pneumatic precise positioning system is well constructed. The pneumatic actuator has low friction. The pneumatic servo valve has high performance. It is confirmed that the performance of the pneumatic servo system is affected by the dynamic characteristic of the servo valve.

The natural frequency of the servo valve is set three times higher than the best commercial one. We conducted several experiments and made clear that the steady state error of the system becomes lower than ±0.1 μm for step response. Moreover the tracking error of less than ±0.5 μm was realized during sinusoidal motion.

REFERENCES