Motion Control of an Ultra Precision Positioning Stage

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Abstract - Motion control system of an ultra precision positioning stage with application to IC production is presented. Configuration and master/slave principle of ultra precision positioning stage are pictured. By adopting merits of both coarse and fine stage, a desired motion control system having the capacity of large workspace with high resolution of motion is enabled. The automated synthesis procedure of parallel PID controller and robust controller are developed using loop shaping techniques. Force decoupling strategy is introduced under the consideration of reality in practice. The simulation result shows that the positioning precision achieves to 10 nanometers using dual stage.

Index Terms - Motion control; ultra precision; force decoupling; nanometer.

I. INTRODUCTION

Ever-increasing demands on higher productivity and better product quality in advanced industries, such as the semiconductors and precision engineering industries, has continued to motivated and stimulate the development of high speed and high precision motion control system [1]. For this account, a new concept of servo system appeared, named the dual-stage servo system, which is defined as a combination of coarse and fine stages for long stroke, fast and precise positioning. In dual-stage systems, the coarse stage is used for coarse and large range motion while the fine stage for fine and small range motion. Usually, the fine stage has the characteristic of low power and small stroke but higher precision. Whereas, the situation is reversed in the coarse stage. So, by adopting merits of both coarse and fine stage, a desired motion control system having the capacity of large workspace with high resolution of motion is enabled. There were lots of studies where the concept of dual-stage compound stage is applied. For example, we can find several works on macro/micro manipulators and dual-stage positioning tables [2–5]. However, in these days, the research on the dual-stage servo seems to be the most active in hard disk drives (HDD) [6–10], where the fine actuator fabricated using the micro machining technology enables high speed track following.

In this paper, we propose an ultra precision positioning stage and its motion control. The system configuration and the master/slaver control work principle in dual-stage are pictured in section 2. The automated synthesis procedure of parallel PID controller and robust controller using loop shaping techniques, and force decoupling strategy are developed in section 3. The simulation results are showed in section 4. Section 5 concludes this paper following with acknowledgment in section 6.

II. SYSTEM DESCRIPTION

Setup of the ultra-precision positioning stage is shown in Fig. 1 for high speed and high precision production in IC manufacturing. The system mainly consists of two perpendicular long stroke motors, three short stroke motors, a granite stone and E-chuck/airfoot assembly. The position is monitored by an HP laser interferometer system. The long stroke motors are also referred to as ‘LIMMS’ motors, which stand for ‘Linear Motion Motor System’, having a large range of travel of about 300mm. The short stroke motor, working according to the Lorentz motor principle, can be seen as a small ‘cage’ in which the E-chuck can move without any mechanical contact. The range of movement for the E-chuck is approximately 2mm. The short stroke coil assembly is fixed to the long stroke motor assembly by very strong clamping mechanism. Short stroke motors are devoted to fine stage, and long stroke motors to coarse stage. The E-chuck is positioned by the short stroke motors (X, Y1 and Y2), which further are carried by the long stroke motors. The short stroke motors can move a very limited range with high accuracy, while the long stroke motors move the short stroke assembly over a long range with relatively low accuracy. The relative motion between E-chuck and mover of long stroke motor is detected by difference sensors (which are not shown in Fig. 1). As soon as the E-chuck moves, as a result of short stroke motor force, the difference sensors will detect that movement and long stroke motion arises. The short stroke motors move under HP-control and the long stroke motors move under difference sensor control following E-chuck. So is called master/slaver principle.

Fig.1 Structure sketch of ultra-precision positioning stage
III. MOTION CONTROL SYSTEM

As shown in Fig. 2, the motion control system dealing with ultra-precision servo-control problem mainly consists of modeling ($G_p$), feedback ($G_c$), trajectory ($r^*$), feedforward ($F_i$ or $ff$) and other advanced compensation methods.

A. Mechanical Dynamics

An accurate system description is needed for the design of any of the aspects of the servo control system. Mechanical dynamic model of the studied positioning system can be built using the method introduced in reference [1] by dint of Matlab/Simulink/Simmechanics toolbox. When the system having a force as input, and measured position as output, the system will exhibit a double integrator behaviors in a large low-frequency field, and some addition high-frequency resonance as shown in Fig. 3, which is alike at each DOF direction of the coarse and fine stage. These are profited from the frictionless motion using air bearing. When ignoring any resonances of the system, a free moving mass can be used as a simple model, written as formula (1), therefore it is effective to design the feedforward signal $ff$ having an acceleration profile only.

$$G_p(s) = \frac{1}{ms^2} \tag{1}$$

where $m$ — a free moving mass, kg

The notch filter ($G_n$) in Fig. 2 located directly after the summing point prevents the mechanic to be excited on the frequency by feedforward. Transfer function of the notch filter ($G_n$) employs the following form:

$$G_n(s) = \frac{\omega_n^2}{s^2 + 2\zeta_n\omega_n s + \omega_n^2} \tag{2}$$

As a zero dB crossover frequency approximately equals to the desired closed-loop bandwidth ($\omega_c$), we can conclude $\omega_n$ based on formula (4).

$$k_d = \frac{k_i}{\omega_c G(s)} = \frac{m \cdot 1}{\omega_c} \tag{5}$$

B. Controller Synthesis

There are some approaches in controller design for dual-stage system [3, 4, 6–10]. Because the air bearing driven system, the axial motion of stages will produce almost no motion of the other DOFs. So, we devote to single DOF controller synthesis using loop shaping techniques. The goal of the loop shaping techniques used here is to obtain an open-loop frequency shape with the following characteristics:

a) High gain at low frequencies for good tracking and disturbance rejection;

b) A slope of -20dB/dec slope in regions around the desired closed-loop bandwidth for a good phase margin and stability;

c) Limited peak-value of sensitivity magnitude;

d) Limited control energy.

1) Parallel PID Controller Design

The transfer function of traditional parallel PID feedback controller is given in formula (3).

$$G_i(s) = k_p + \frac{k_i}{s} + k_d s = \frac{k_i(s + z_1)(s + z_2)}{s} \tag{3}$$

where $k_p$ — proportion gain;

$k_i$ — integrator constant;

$k_d$ — differential constant;

$z_1$ — first zero of controller;

$z_2$ — second zero of controller

As seen in equation (3), $G_i(s)$ can be factored into two zeros, a free integrator, and the derivative gain ($k_d$).

Since these zeros will increase the slope of the open-loop transfer function, it is important to push them out to as near the crossover frequency as possible, and place them before the crossover where the slope should be -20dB/dec. This is necessary to allow the phase lag to come up towards -90 degree.

For the sake of convenience, the open-loop transfer function (denoting as $G$) in Fig. 2 can be simply written as:

$$G(s) = G_i(s) \cdot G_c(s) \cdot G_p(s) = \frac{k_i(s + z_1)(s + z_2)}{s m s^2} \tag{4}$$

As a zero dB crossover frequency approximately equals to the desired closed-loop bandwidth ($\omega_c$), we can conclude $k_d$ based on formula (4).

$$k_d = \frac{m \cdot 1}{\omega_c G(j\omega)_{s=\omega_c}} \tag{5}$$

From formula (3) the proportion gain and integrator constant can be found that

$$k_p = k_i (z_1 + z_2) \tag{6}$$

$$k_i = k_d (z_1 \cdot z_2) \tag{7}$$
Furthermore, to limit the bandwidth of the controller, an extra high-frequency pole is added, which causes the control gain to roll-off at high-frequencies. In practice the pure differential item $k_s$ is commonly replaced with $k_s (s/N + 1)$ using N as approximately 10 times of second zero $z_2$. Hence, the entire parallel PID controller can be described as:

$$G_c(s) = \left( k_p + \frac{k_d}{s} + \frac{k_i}{N^2 s + 1} \right) \frac{1}{\tau_s s + 1}$$  \hspace{1cm} (8)

where the last item is roll-off transfer function which time constant is $\tau_h$.

2) Alternative Robust Controller Design

A robust feedback controller can be designed using $H_s$ mathematical framework alternatively. From the interconnection structure shown in Fig. 2, the robust controller design problem is to solve the so-called mixed sensitivity problem:

$$\begin{bmatrix} W_s S & W_s T \\ W_s G & W_s G_s \end{bmatrix}_{\infty} < 1$$  \hspace{1cm} (9)

with $S_s, T$ denoting the output sensitivity and complementary sensitivity transfer function. It is shown that $W_s$ can be used to reflect disturbances and low frequency performance requirements like desired values of the sensitivity at certain frequencies, whereas $W_s$ originates from additive uncertainty and can also be used to specify high frequency performance requirements, e.g., the roll-off of the controller dynamics. For achieving to the loop shaping characteristic, we followed the approach suggested in [1], which solving the following design problem:

$$\begin{bmatrix} W_s S G_p & W_s S_p \\ W_s G_s G_p & W_s G_s G_p \end{bmatrix}_{\infty} < 1$$  \hspace{1cm} (10)

Here, $G_p$ should use the identified model though experiment data. By weighting $S, G_p$ and $G_s, S_p$ instead of $S_s$ and $T$, respectively, use is made of the double integral action of $G_p$, and the double derivative action of $S$ at low frequencies, respectively. In order to improve the numerical conditioning of formula (10), the plant model is firstly scaled in magnitude such that it has magnitude 0 dB near the expected bandwidth in each direction.

It can be easily verified that the open-loop interconnection in Fig. 2 is given by:

$$z_i = \begin{bmatrix} \begin{bmatrix} W_s G_p & W_s G_p & r_1 \\ 0 & 0 & W_s G_p \\ G_p & I & -G_p \end{bmatrix} \end{bmatrix} \begin{bmatrix} r_1 \\ r_2 \\ u \end{bmatrix}$$  \hspace{1cm} (11)

results in the design problem (10). To keep the controller complexity as low as possible, one order weights for $W_i$ and $W_2$ are used:

$$W_i(s) = k_{i} \left( \frac{s + \omega_{i1}}{s + \omega_{i2}} \right) \hspace{1cm} (i = 1, 2)$$  \hspace{1cm} (12)

where $k_i, \omega_{i1}$ and $\omega_{i2}$ are design parameters. To achieve above loop shaping characteristics goal, there are three requirements which are used to determine three parameters for weighting function $W_i$: 

$$k_i = \frac{1}{s_p}$$  \hspace{1cm} (13)

with $s_p$ denoting the peak-value.

$$k_i \left( \frac{j \omega_0 + \omega_0}{j \omega_1 + \omega_1} \right) = s_i$$  \hspace{1cm} (14)

with $\omega_0$ denoting the low frequency and $s_i$ denoting the desired gain of the $1/(S, G_p)$ at frequency $\omega_0$.

$$k_i \left( \frac{j \omega_0 + \omega_0}{j \omega_0 + \omega_0} \right) = 0$$  \hspace{1cm} (15)

with $\omega_0$ denoting the desired bandwidth.

$\omega_{i1}$ and $\omega_{i2}$ of $W_i$ can directly be calculated through the (14) and (15) resulting in:

$$\omega_{i1} = \sqrt{\omega_0^2 (s_1^2 - k_1^2) + \omega_0^2 s_1^2 (k_1^2 - 1)}$$  \hspace{1cm} k_1 \left( 1 - s_1^2 \right)$$  \hspace{1cm} (16)

$$\omega_{i2} = \sqrt{\omega_0^2 (s_2^2 - k_2^2) + \omega_0^2 s_2^2 (k_2^2 - 1)}$$  \hspace{1cm} 1 - s_2^2 \right)$$  \hspace{1cm} (17)

A general rule of thumb is to choice $\omega_{i1}$ and $\omega_{i2}$ as 3 and 30 times of $\omega_0$ respectively, and $k_2$ is chosen equal to 200 times of $\omega_{i2}/\omega_{i1}$.

Then, the robust controller can be solved, and the reduced-order force controller for practicality can be achieved resorting to balanced and truncated method, using programs in Matlab/Robust Control Toolbox.

C. Force Decoupling

The short stroke motors control the accurate positioning of the E-chuck in the X, Y and Rz directions. As seen from Fig. 4, the action points of short stroke motors are not coincidence with the COG of the E-chuck. The output forces $[F_x, F_y, F_z]$ of short stroke motors and the force/torque $[f_x, f_y, f_z]$ at the COG of the E-chuck are related with the following mapped formula:

$$\begin{bmatrix} f_x \\ f_y \\ f_z \end{bmatrix} = \begin{bmatrix} 1 & 0 & 0 \\ 0 & 1 & 1 \\ -L_3 & -L_4 & L_3 \end{bmatrix} \begin{bmatrix} F_x \\ F_y \\ F_z \end{bmatrix}$$  \hspace{1cm} (18)

Fig.4 Schematic force mapping relation
Then the control signals denoting force/torque values, from the controllers designed in X, Y and Rz directions of fine stage, must be decoupled into the force signals used in short stroke motors. From equation (18) the relation between them can be easily rewritten as:

\[
\begin{bmatrix}
F_x \\
F_y \\
F_z \\
\end{bmatrix} = J^T \begin{bmatrix}
f_x \\
f_y \\
\tau_z \\
\end{bmatrix}
\]

with \( J^T \) given by

\[
J^T = \begin{bmatrix}
1 & 0 & 0 \\
L_1 + L_2 & L_2 & L_1 + L_2 \\
L_1 & L_1 & L_1 + L_2 \\
\end{bmatrix}
\]

IV. SIMULATION

The content described above has been validated using simulation. To push the performance of the electromechanical system to its limits, a smooth three order setting trajectory for point-to-point control is used, which can also limit the activation of mechanical resonances. The parallel PID controllers are designed for coarse stage and robust controllers for fine stage. An acceleration profile feedforward signal which further enhance the tracking performance, and some notch filters limiting high frequency resonances are used in simulation. The disturbance and noise are imitated using white noise. And the sample time is 250\( \mu \)s. Some important simulation parameters are listed in TABLE I and TABLE II.

<table>
<thead>
<tr>
<th>TABLE I</th>
<th>SIMULATION PARAMETERS OF FINE STAGE</th>
</tr>
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<tbody>
<tr>
<td>mass(kg)</td>
<td>m</td>
</tr>
<tr>
<td>bandwidth(Hz)</td>
<td>( \omega_n )</td>
</tr>
<tr>
<td>peak-value (dB)</td>
<td>( s_p )</td>
</tr>
<tr>
<td>low frequency point(Hz)</td>
<td>( \omega_l )</td>
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<tr>
<td>desired gain of low frequency point(dB)</td>
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<tr>
<td>parameter of weighting function No.1</td>
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<td>parameter of weighting function No.1</td>
<td>( \omega_{11} )</td>
</tr>
<tr>
<td>parameter of weighting function No.1</td>
<td>( \omega_{12} )</td>
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<tr>
<td>parameter of weighting function No.2</td>
<td>( k_2 )</td>
</tr>
<tr>
<td>parameter of weighting function No.2</td>
<td>( \omega_{21} )</td>
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<tr>
<td>parameter of weighting function No.2</td>
<td>( \omega_{22} )</td>
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<tr>
<td>scaled gain of plant model</td>
<td>( k_m )</td>
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</table>

<table>
<thead>
<tr>
<th>TABLE II</th>
<th>SIMULATION PARAMETERS OF COARSE STAGE</th>
</tr>
</thead>
<tbody>
<tr>
<td>mass(kg)</td>
<td>m</td>
</tr>
<tr>
<td>bandwidth(Hz)</td>
<td>( \omega_n )</td>
</tr>
<tr>
<td>no.1 zero(rad/sec)</td>
<td>( z_1 )</td>
</tr>
<tr>
<td>no.2 zero(rad/sec)</td>
<td>( z_2 )</td>
</tr>
<tr>
<td>proportion gain</td>
<td>( k_1 )</td>
</tr>
<tr>
<td>integrator constant</td>
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<tr>
<td>differential constant</td>
<td>( k_d )</td>
</tr>
<tr>
<td>differential time constant</td>
<td>N</td>
</tr>
<tr>
<td>roll-off time constant</td>
<td>( n_0 )</td>
</tr>
</tbody>
</table>

Fig.5 shows tracking trajectory of fine stage, and the tacking results in simulation are shown in Fig.6 along X direction. As seen, the short stroke motor moves in the region of 2\( \mu \)m, that means the short stroke motor moves in a very limited range with high accuracy, while the long stroke motor follows the short stroke to fulfill over a long range with relatively low accuracy.

V. CONCLUSION REMARKS

In this paper we present motion control system of an ultra precision positioning stage with application to IC production. Using loop shaping techniques, the automated synthesis procedure of parallel PID controller and robust controller are developed. Force decoupling strategy is pictured under the consideration of reality in practice. The simulation result with detailed simulation parameters shows that the positioning precision achieves to 10 nanometres using coarse/fine dual stage. By adopting merits of both coarse and fine stage according to master/slaver principle, a desired motion control...
system having the capacity of large workspace with ultra-precision motion is enabled.

For the sake of simple dealing, the effect of electric amplifier dynamics is not taken into account which contributing to the high frequency dynamics of the entirely system. More compensation signals can be used for disturbances originating from various sources in real system.

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